Heating Ventilating Air Conditioning Guide 1941

HEATING VENTILATING AIR CONDITIONING GUIDE

1941

AN INSTRUMENT OF SERVICE PREPARED FOR THE PROFESSION—CONTAINING A

Technical Data Section

OF REFERENCE MATERIAL ON THE DESIGN AND SPECIFICATION OF HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS BASED ON—THE TRANSACTIONS—THE INVESTIGATIONS OF THE RESEARCH LABORATORY AND CO-OPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY

TOGETHER WITH A

Manufacturers' Catalog Data Section

CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT

ALSO

The Roll of Membership of the Society

WITH

Complete Indexes

TO TECHNICAL AND CATALOG DATA SECTIONS

Vol. 19

$5.00 per Copy

(With thumb index $5.50)

PUBLISHED ANNUALLY BY

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

51 MADISON AVENUE ∴ NEW YORK, N. Y.

and by it
Dedicated
To the Advancement of
The Profession
and
Its Allied Industries

Printed and Bound by
The Horn-Shafer Company
BALTIMORE :-: MARYLAND

PREFACE TO THE 19th EDITION

THE nineteenth edition of the HEATING, VENTILATING, AIR CONDITIONING GUIDE has been prepared to provide readers with recent, authoritative and useful information. In accordance with established policy, a considerable part of the text material has been reviewed or rewritten by authors who were selected by the Guide Publication Committee because they were acknowledged authorities on the subjects covered. Much old material has been retained and that covering new developments has been added.

The number of chapters is the same as for the previous edition but there has been some rearrangement of material. Chapter 23, Cooling, Dehumidification and Dehydration, and Chapter 24, Refrigeration are new chapters containing material formerly in the old chapters on Refrigerants and Air Drying Agents, and Cooling and Dehumidification Methods, plus a considerable amount of new information. The chapters have also been rearranged and grouped into seven sections as will be noted on the Contents page. This has been done to make a more logical arrangement and to better correlate the Technical Data Section and the Catalog Data Section. It has required revision of the numbering by which some of the first chapters have been known for several years but the advantages of the change appeared to be justified.

In all, twenty-one chapters have been reviewed or rewritten this year and minor changes have been made in other chapters. Chapter 1, Thermodynamics of Air and Water Mixtures, is entirely new. It is based on the most recent information available on the subject. Since the treatment is new, it is given in greater detail than is customary in the GUIDE, for the benefit of readers who may wish to follow such a detailed discussion. A new chart, Mollier Diagram for Moist Air, is presented for use in analyzing air conditioning processes. It is printed with the Bulkeley Psychrometric Chart and will be found in the pocket on the inside back cover. The steam table in Chapter 1 has been enlarged to include a temperature table covering the range needed for most air conditioning problems.

The chapter on Heat Transmission Coefficients and Tables contains some new data developed by recent research investigations. The table of climatic conditions in Chapter 5, Heating Load, has been enlarged by adding a column of recommended design temperatures. In Chapter 6 on Cooling Load will be found new data on solar heat transmission through walls, roofs and glass blocks resulting from investigations made by the A.S.H.V.E. Research Laboratory. Curves showing measured heat flow into structures replace the theoretical analysis given in previous editions. The new data on walls and roofs result in a reduction in the calculated cooling load.

New material on cooling towers is included in Chapter 26, Spray Equipment. The chapters on Air Duct Design, Sound Control and

Automatic Control have been rewritten. The old chapter on Railway Air Conditioning has been rewritten and is now called Transportation Air Conditioning. Information on buses and automobiles has been added and references to air conditioning of ships and airplanes are included. In the chapter on Terminology, some of the definitions have been revised to make them more exact and a table of specific heats has been added. Other chapters which have been reviewed and revised are as follows: Chimneys and Draft Calculations, Automatic Fuel Burning Equipment, Heat and Fuel Utilization, Central Systems for Comfort Air Conditioning, Fans, Air Distribution, Air Conditioning in the Treatment of Disease, Industrial Air Conditioning and Electric Heating.

The Committee is appreciative of the response of the membership to the request which it made for suggestions for the GUIDE. Many valuable and useful ideas were received. All were carefully considered and as many as possible were incorporated in this edition.

The text is made up of the contributions of many individuals over a period of years. To those who have contributed to previous editions the material which makes up the greater part of the Technical Data Section, the Committee acknowledges its indebtedness. Thanks is particularly due to the following who have contributed the new material which appears in this edition:

F. R. BICHOWSKY	R. E. GOULD	W. G. SCHLICHTING
L. M. K. BOELTER	W. A. GRANT	E. L. SCHULZ
C. J. BRAATZ	F. C. HOUGHTEN	ERNEST SZEKELY
R. M. CONNER	A. D. MARSTON	H. F. TAPP
M. B. FERDERBER	F. C. MCINTOSH	WALLACE WATERFALL
PETER FRANCK	ARVIN PAGE	E. C. WEBB
M. C. GIANNINI	HALE J. SABINE	E. R. WOLFERT
JOHN A. GOFF	W. M. SAWDON	WILLIAM M. WALLACE, II

The Committee on Research, the A.S.H.V.E. Research Laboratory and several others have been particularly helpful. They have provided much new and needed information. Technical papers presented before the Society have also been the source of useful material.

To all of those who have reviewed material, contributed advice and suggestions and who have assisted in many ways in the preparation of this edition, the Guide Publication Committee extends its thanks.

An important part of the GUIDE is the Catalog Data Section. Much useful information has been supplied by the various manufacturers and readers will be well repaid by a careful examination of this section. Equipment has been grouped in sub-divisions for convenience in locating data about a particular type of equipment.

The most recent and reliable data available have been used in this edition and it covers the latest developments in the field of heating, ventilating and air conditioning. The Committee hopes that the many readers will find it informative and useful.

GUIDE PUBLICATION COMMITTEE

S. S. SANFORD, *Chairman*

P. D. CLOSE — A. J. OFFNER

C. M. HUMPHREYS — M. F. RATHER

JOHN JAMES, *Technical Secretary*

CONTENTS

CODE *of* ETHICS *for* ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

INDEX

●

HEATING VENTILATING
AIR CONDITIONING
GUIDE 1941

●

TECHNICAL DATA SECTION

CHAPTERS 1-46 and PAGES 1-808

●

Cross Reference to Subjects in
Chapters 1-46 Alphabetically Listed

●

19th EDITION

INDEX

HEATING VENTILATING AIR CONDITIONING GUIDE 1941

Technical Data Section
Chapters 1-46 and Pages 1-808

A

D

E

F

I

Q-R

S

U

Chapter I

THERMODYNAMICS OF AIR AND WATER MIXTURES

Dry Air, Specific Enthalpy, Water Vapor, Moist Air, Dalton's Law, Humidity Ratio, Relative Humidity, Dew-point, Enthalpy, Thermodynamic Wet-bulb Temperature, Mollier Diagram, Typical Air Conditioning Processes, Adiabatic Saturation, Psychrometric Chart, Steady Flow Energy Equation

THE working substance of the air conditioning engineer may be regarded, for the purpose of analysis, as a mixture of only two constitutents, dry air and water. The mixture may consist of two, and possibly three distinct phases, solid, liquid and vapor. The vapor phase is conveniently referred to as *moist air* and is regarded as a mixture of *dry air* and *water vapor*.

DRY AIR

Composition. Dry air is itself a mixture of several gases, but its composition is subject to such slight variation that it may be regarded as fixed. According to *International Critical Tables*, the mol-fraction composition of dry air is given by the first column of figures in Table 1. Molecular weights are given in the second column; the last figure in the third column is the *apparent* molecular weight; the fourth column of figures gives the ordinary weight-fraction composition.

It is well known that dry air contains other gases besides those listed in Table 1; but these are present in such minute amounts that they can be grouped together as argon. Values in the lower section of Table 1 give the approximate mol-fraction composition of what is called argon in the upper portion of the Table.

In physical and chemical thermodynamics, there is a distinct advantage in using a different unit of weight, the mol, for each different substance involved. A mol of oxygen weighs 32.000 lb as a matter of definition; a mol of any other substance is a weight, in pounds, equal to its molecular weight.

Specific Volume and Density

The ratio of total volume to total weight is called *specific volume*, *v*. In the English system, volume is expressed in cubic feet and weight in pounds; hence, specific volume is expressed in cubic feet per pound. The reciprocal of specific volume, that is, weight per unit volume is called *weight density*, *d*. The unit of density is the pound per cubic foot.

The earliest investigation into the relation between pressure, specific volume, and temperature for gases was made by Boyle (1661) who was able to confirm the hypothesis that the volume of a given weight of gas should vary inversely as the absolute pressure if temperature is maintained constant. Thus, within the limits of his experimental error Boyle found that at constant temperature the product pv, pressure times specific volume, has a constant value over a considerable range of pressures. These results are best visualized by plotting values of the product pv as ordinate against values of pressure p itself as abscissa. According to Boyle's experimental findings lines of constant temperature (isotherms) of a gas are *straight* and *horizontal* on this pv, p-plane.

The first rough experiments of Charles (1787) and the subsequent more refined experiments of Gay-Lussac (1802) suggested the possibility

TABLE 1. COMPOSITION OF DRY AIR

Gas	Mol per Mol Dry Air		Lb per Mol		Lb per Mol Dry Air	Lb per Lb Dry Air
Nitrogen	0.7803	×	28.016	=	21.861	0.7547
Oxygen	0.2099	×	32.000	=	6.717	0.2319
Carbon Dioxide	0.0003	×	44.003	=	0.013	0.0004
Hydrogen	0.0001	×	2.016	=	0.000	0.0000
Argon	0.0094	×	39.944	=	0.376	0.0130
	1.0000				28.967	1.0000

Composition of Argon	Mol per Mol Dry Air
Argon	0.00933
Neon	0.000018
Helium	0.000005
Krypton	0.000001
Xenon	....
	0.00935

of establishing a universal temperature scale such that the product pv *for any gas* is simply proportional to temperature measured on this scale in accordance with Equation 1.

$$pv = BT \tag{1}$$

where B is a constant characteristic of the given gas. Referring to the graphical representation previously described in which the product pv is plotted as ordinate against pressure p as abscissa, the vertical spacing of the isotherms should be such that the ordinates to any two isotherms are in the ratio of corresponding absolute temperatures and therefore in the same ratio for any gas.

Precise measurements by modern methods have shown that the experimental findings of Boyle, Charles and Gay-Lussac are only approximately correct. In the range of sufficiently low pressures the isotherms of gases are indeed *straight* on the pv, p-plane; but they are *not horizontal* in accordance with Boyle's Law, being inclined downward to the right at

relatively low temperatures, upward to the right at higher temperatures. Extrapolation of each isotherm to zero pressure has revealed the remarkable fact that the limiting value of the product pv thus obtained is strictly proportional to absolute temperature as suggested by Equation 1, this strict proportionality providing an accurate basis for the establishment of the absolute temperature scale.

The experimental facts of the preceding paragraph are expressed mathematically by Equation 2.

$$pv = BT - A(T)\, p \tag{2}$$

where

p = absolute pressure, pounds per square foot.
v = specific volume, cubic feet per pound.
B = a constant depending on the molecular weight of the gas.
T = absolute temperature, degrees Fahrenheit.
$A(T)$ = a temperature function called *second virial coefficient*, cubic feet per pound. The name undoubtedly originated from consideration of Clausius' Virial Theorem according to which the mean kinetic energy of a molecular aggregate is equal to the mean value of a quantity, which Clausius called the *virial* of the system, depending solely on the forces acting upon the molecules and not upon the motion of the molecules. This name is used extensively. For many gases, the magnitude of the second virial coefficient can be predicted from theory; but at present, direct experimental measurements are more reliable.

It will appear in what follows that the error committed in computing values of specific volume from Equation 1 instead of Equation 2 is extremely small. Thermodynamically, however, the former would deny the effect of pressure on the thermal properties of a gas which experiment shows to be appreciable. Therefore Equation 1 cannot be made the basis of an accurate analysis.

The numerical value of the constant B in Equation 2 is different for every different gas, but can be calculated if the molecular weight m, pounds per mol, is known; for the product mB is a universal gas constant R, namely,

$$R = 1545.4$$

Example 1. Find the value of B for dry air and water vapor.

Solution. $B_a = 1545.4 \div 28.967 = 53.351$
$B_w = 1545.4 \div 18.0154 = 85.782$

The temperature function $A(T)$, the so-called *second virial coefficient*, expresses the effect of intermolecular forces. It is positive at low temperatures where these forces are predominantly attractive, negative at higher temperatures where they are predominantly repulsive. It is known with satisfactory accuracy for both dry air and water vapor. Values of specific volume are listed in Table 2 for dry air at standard atmospheric pressure (29.921 in. Hg) as computed from Equation 2.

The fact that $A(T)$ is multiplied by pressure in Equation 2 means that intermolecular forces vanish at zero pressure and infinite volume where infinite distances separate the molecules. The finite value of the product pv at zero pressure is due entirely to the translational kinetic energy of the molecules. In ordinary calculations not requiring too

TABLE 4. SPECIFIC ENTHALPY OF DRY AIR[a] AT 29.921 IN. HG

TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a	SPECIFIC HEAT $\left[c_p\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a	SPECIFIC HEAT $\left[c_p\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a	SPECIFIC HEAT $\left[c_p\right]_o^t$
−96	−23.035	0.2399	32	7.680	0.2400	160	38.454	0.2403
−64	−15.356	0.2399	64	15.363	0.2400	192	46.172	0.2405
−32	−7.678	0.2399	96	23.053	0.2401	224	53.903	0.2406
0	0.000	0.2400	128	30.749	0.2402	256	61.649	0.2408

[a]Prepared by John A. Goff.

enthalpy of dry air at standard atmospheric pressure (29.921 in. Hg) as computed from Equation 6 are given in Table 4.

Reference Point. It is desired to give some prominence to the choice of reference point. As energy, and therefore enthalpy, is purely relative, any convenient state can be selected at which to assign the value *zero* to specific enthalpy. The state chosen is 0 F, 29.921 in. Hg. Perhaps the only really valid argument for this particular choice is that, for ordinary calculations at, or near, atmospheric pressure, a very simple equation can be used, namely,

$$h_a = 0.24t \tag{7}$$

WATER VAPOR

Saturation Pressure. It is common knowledge that a substance like water can exist in at least three distinct phases, solid (ordinary ice), liquid and vapor; and that under certain conditions two or more phases can *coexist* in stable equilibrium. For example, steam having a quality of 98 per cent is a mixture of two coexisting phases, vapor and liquid, 98 per cent by weight being vapor and 2 per cent by weight, liquid. When two phases can coexist in stable equilibrium, each is said to be *saturated* with respect to the other.

One of the important problems of thermodynamics is to formulate the conditions for *saturation* in mathematical terms. The answer to the problem can be stated quite generally as equality, between the several coexisting phases, of (a) pressure, (b) temperature, (c) each component *chemical potential.*

In the case of a pure substance like water, containing a single component, there is only one component chemical potential; and this becomes identical with a thermodynamic property called *specific free enthalpy* denoted by the letter g (Btu per pound) and defined by the equation:

$$g = h - Ts$$

where

h = specific enthalpy, Btu per pound.
T = absolute temperature, degrees Fahrenheit.
s = specific entropy, Btu per pound per degree Fahrenheit.

To illustrate, *liquid* water at 212 F, 14.696 lb per square inch has a specific free enthalpy of $180.07 - 671.70 \times 0.3120 = -25.90$ Btu per pound. At the same temperature and pressure, water *vapor* has a specific

free enthalpy of $1150.4 - 671.70 \times 1.7566 = -25.90$ Btu per pound. The numerical data used in these calculations are to be found in the steam tables[1]. Since the two specific free enthalpies are equal *at the same temperature and the same pressure*, the two phases can coexist in stable equilibrium to form a saturated mixture and are therefore saturated with respect to each other.

But suppose that a different pressure had been assumed, the temperature being 212 F as before; for example, assume a pressure of 14 lb per square inch. The specific free enthalpy of the liquid phase will be practically the same as before, but that of the vapor phase will change from -25.90 to -32.84 Btu per pound, most of this change being due to change of entropy which, in the case of a vapor, depends markedly upon the pressure. Since the specific free enthalpies of the two phases are no longer equal, they cannot coexist in stable equilibrium and neither is saturated. As a matter of fact the vapor is superheated while the liquid is supersaturated.

From this analysis it will be seen that *to a given temperature T there corresponds a definite saturation pressure p_s.* This is also called the *vapor pressure* of the liquid or solid as the case may be. It will also be seen that a working definition of saturation can only be arrived at by application of the fundamental laws of thermodynamics.

Referring specifically to the vapor phase, if the actual pressure is *less* than the saturation pressure corresponding to the actual temperature, the vapor is said to be *superheated*; if it is *greater*, as it may well be under proper circumstances, the vapor is said to be *supersaturated.* Values of the saturation pressure of pure water are given in Table 6[2].

Specific Volume

Accurate values of the specific volume of water vapor at pressures equal or near the saturation pressure (for the given temperature) can be computed from Equation 1 since the second virial coefficient $A(T)$ is known with satisfactory accuracy. Usually, however, the desired information can be read directly from the steam tables. Values for the specific volume of the *saturated* vapor, v_g, are also listed in Table 8.

Specific Enthalpy

The zero-pressure specific enthalpy, as calculated by A. R. Gordon from spectroscopic measurements, has recently been corrected for distortion of the water molecules due to centrifugal forces. Best values at present available are listed in Table 5.

From the numerical values of mean specific heat, it is clear that for ordinary calculations the following simple relation may be used:

$$h_w^o = 0.444t + 1061 \tag{8}$$

Reference Point. The reference point for water has been chosen as *saturated liquid at 32 F* in conformity with usual steam table practice. In order to refer the zero-pressure values of specific enthalpy to this

[1]Thermodynamic Properties of Steam, by J. H. Keenan and F. G. Keyes, published by John Wiley & Sons. Inc., 1936, of which Table 8 is an abridgment.

[2]Strictly speaking the values listed in Table 6 are not values of p_s as labeled, but of p'_s (Equation 13b) with the Dalton Factor (DF) taken to be unity.

TABLE 5. SPECIFIC ENTHALPY OF WATER VAPOR AT ZERO PRESSURE[a]

TEMP F t	SPECIFIC ENTHALPY BTU PER LB h^o_w	MEAN SPECIFIC HEAT $\left[c^o_p\right]^t_o$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h^o_w	MEAN SPECIFIC HEAT $\left[c^o_p\right]^t_o$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h^o_w	MEAN SPECIFIC HEAT $\left[c^o_p\right]^t_o$
−96	1018.49	0.4425	32	1075.16	0.4435	160	1132.26	0.4455
−64	1032.64	0.4427	64	1089.39	0.4440	192	1146.64	0.4462
−32	1046.80	0.4429	96	1103.64	0.4444	224	1161.08	0.4469
0	1060.97	0.4431	128	1117.93	0.4450	256	1175.58	0.4477

[a]Prepared by John A. Goff from published data computed from spectroscopic measurements.

datum, best available information regarding latent heat, saturation pressure and second virial coefficient at 32 F has been used. The values in Table 5 do not agree exactly with those in the steam tables, but do agree with later information from the National Bureau of Standards.

MOIST AIR

Dalton's Law. Having accurate information regarding the thermodynamic properties of dry air and water vapor *separately*, it is desired to predict the properties of moist air which is regarded as a mixture of these two constitutents. Statistical mechanics furnishes a starting point in the form of a prediction that, at not too high pressures,

$$Pv = RT - \left[A_{aa}x^2 + 2A_{aw}\,x\,(1 - x) + A_{ww}\,(1 - x)^2\right] P \qquad (9)$$

where

P = observed pressure, pounds per square foot.
v = specific volume, cubic feet per *mol.*
A_{aa} = second virial coefficient for the dry air expressing the effect of forces between air—air molecules, cubic feet per *mol.*
A_{ww} = second virial coefficient for the water vapor, expressing the effect of forces between water—water molecules, cubic feet per *mol.*
A_{aw} = *interaction constant* expressing the effect of forces between air—water molecules, cubic feet per *mol.*
x = mol-fraction of dry air in the mixture, mols dry air per mol mixture.

Equation 9 will be recognized as a generalization of Equation 2. Both A_{aa} and A_{aw} are known; but there is no reliable information at present available on the interaction constant A_{aw} though experiments are in progress[3] to measure it. Pending the results of these experiments, an accurate and thermodynamically consistent treatment is impossible and the simplest thing to do is to ignore the effect of intermolecular forces entirely.

But, in the absence of intermolecular forces, each constituent gas in a mixture such as moist air would behave exactly as if it alone occupied the volume V at the temperature T of the mixture and: (1) the observed pressure P would be the sum of individual *partial* pressures p; (2) the total enthalpy H would be the sum of the individual enthalpies. This is the essence of Dalton's Law of Partial Pressures.

[3]At the Towne Scientific School, University of Pennsylvania, in cooperation with the A.S.H.V.E. through the Research Technical Advisory Committee on Psychrometry.

Referring to dry air by the subscript *a* and, to water vapor by the subscript *w*, Dalton's Law would predict

$$V = \frac{n_a RT}{p_a} = \frac{n_w RT}{p_w} = \frac{(n_a + n_w)\, RT}{P} \tag{10a}$$

where

$$P = p_a + p_w \tag{10b}$$

From these equations are easily obtained,

$$\frac{n_w}{n_a} = \frac{p_w}{P - p_w} \quad \text{or} \quad \frac{p_w}{P} = \frac{n_w/n_a}{1 + n_w/n_a} \tag{10c}$$

in which,

p_a = partial pressure of the *dry air.*
p_w = partial pressure of the *water vapor.*
P = observed pressure of the mixture.
n_a = weight of *dry air* (mols).
n_w = weight of *water vapor* (mols).

Humidity Ratio

In Equation 10c the ratio by weight of water vapor to dry air, n_w/n_a, is expressed in mols per mol. Most engineers prefer to express it in pounds per pound which can easily be done, since the molecular weights of both water vapor (18.0154 lb per mol) and of dry air (28.967 lb per mol) are known. Thus Equation 10c becomes

$$W = 0.62193\,\frac{p_w}{P - p_w} \quad \text{or} \quad \frac{p_w}{P} = \frac{W}{0.62193 + W} \tag{11}$$

There is little doubt but that the weight ratio W is the most convenient parameter in terms of which to express the composition of moist air; but to choose a suitable name and one that would have general acceptance has always been a perplexing problem. In previous issues of the GUIDE, *specific humidity* was adopted even though it was recognized that the adjective *specific* should properly refer to weight of water vapor per pound of *mixture*, and not per pound of dry air. Various other names have been proposed from time to time including: mixing ratio, proportionate humidity, density ratio, absolute humidity. It is believed that the name *humidity ratio* is most suggestive of the meaning which it is desired to express, that it violates no well established usage as does the name specific humidity and that its adoption will avoid much confusion.

To repeat: in the case of moist air, the ratio by weight (pounds) of water vapor to dry air is called *humidity ratio* and denoted by the letter W.

Saturation

It is often stated that moist air is *saturated* when the water vapor in it is itself in the *dry saturated condition* at the given temperature. This statement would imply that the humidity ratio of saturated moist air is, in accordance with Equation 11,

$$W_s = 0.62193\,\frac{p_s}{P - p_s} \tag{12}$$

where p_s is the saturation pressure of pure water vapor.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds × 10⁵	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure $p_s \times 10^5$: In. of Hg	Lb per Sq In.	Temp Deg F
−60	2.108	0.14756	10.07	0.00	10.07	−14.48	0.02	−14.46	−185.4	101.4	49.808	−60
−59	2.262	.15834	10.09	.00	10.09	−14.23	.02	−14.21	−185.0	108.8	53.443	−59
−58	2.418	.16926	10.12	.00	10.12	−13.99	.02	−13.97	−184.6	116.3	57.127	−58
−57	2.595	.18165	10.14	.00	10.14	−13.75	.03	−13.72	−184.2	124.8	61.302	−57
−56	2.773	.19411	10.17	.00	10.17	−13.50	.03	−13.47	−183.8	133.4	65.526	−56
−55	2.973	0.20811	10.19	0.00	10.19	−13.26	0.03	−13.23	−183.4	143.0	70.242	−55
−54	3.181	.22267	10.22	.00	10.22	−13.02	.03	−12.99	−182.9	153.0	75.154	−54
−53	3.399	.23793	10.24	.00	10.24	−12.78	.04	−12.74	−182.5	163.5	80.311	−53
−52	3.636	.25452	10.27	.00	10.27	−12.53	.04	−12.49	−182.1	174.9	85.911	−52
−51	3.888	.27216	10.29	.00	10.29	−12.29	.04	−12.25	−181.7	187.0	91.854	−51
−50	4.156	0.29092	10.32	0.00	10.32	−12.05	0.04	−12.01	−181.3	199.9	98.191	−50
−49	4.428	.30996	10.34	.00	10.34	−11.81	.05	−11.76	−180.9	213.0	104.63	−49
−48	4.738	.33166	10.37	.00	10.37	−11.57	.05	−11.52	−180.4	227.9	111.94	−48
−47	5.054	.35378	10.40	.00	10.40	−11.32	.05	−11.27	−180.0	243.1	119.41	−47
−46	5.395	.37765	10.42	.00	10.42	−11.08	.06	−11.02	−179.6	259.5	127.47	−46
−45	5.753	0.40271	10.45	0.00	10.45	−10.84	0.06	−10.78	−179.2	276.7	135.92	−45
−44	6.133	.42931	10.47	.00	10.47	−10.60	.06	−10.54	−178.7	295.0	144.90	−44
−43	6.543	.45801	10.50	.00	10.50	−10.35	.07	−10.28	−178.3	314.7	154.58	−43
−42	6.971	.48797	10.52	.00	10.52	−10.11	.07	−10.04	−177.9	335.3	164.70	−42
−41	7.435	.52045	10.55	.00	10.55	−9.872	.077	−9.795	−177.4	357.6	175.65	−41
−40	7.907	0.55349	10.57	0.00	10.57	−9.629	0.082	−9.547	−177.0	380.3	186.80	−40
−39	8.431	.59017	10.60	.00	10.60	−9.388	.088	−9.300	−176.6	405.5	199.18	−39
−38	8.965	.62755	10.62	.00	10.62	−9.146	.093	−9.053	−176.1	431.2	211.81	−38
−37	9.548	.66836	10.65	.00	10.65	−8.905	.100	−8.805	−175.7	459.2	225.56	−37
−36	10.16	.71120	10.67	.00	10.67	−8.663	.106	−8.557	−174.8	488.4	239.90	−36
−35	10.80	0.75600	10.69	0.00	10.69	−8.422	0.113	−8.309	−174.4	519.5	255.18	−35
−34	11.49	.80430	10.72	.00	10.72	−8.180	.120	−8.060	−174.0	552.4	271.34	−34
−33	12.20	.85400	10.75	.00	10.75	−7.939	.127	−7.812	−173.5	586.5	288.09	−33
−32	12.97	.90790	10.77	.00	10.77	−7.698	.136	−7.562	−173.1	623.7	306.36	−32
−31	13.76	.96320	10.80	.00	10.80	−7.457	.144	−7.313	−172.6	661.8	325.08	−31
−30	14.58	1.0206	10.82	0.00	10.82	−7.216	0.152	−7.064	−172.2	701.0	344.33	−30
−29	15.43	1.0801	10.85	.00	10.85	−6.975	.161	−6.814	−171.7	742.2	364.57	−29
−28	16.45	1.1515	10.87	.00	10.87	−6.734	.172	−6.562	−171.3	791.2	388.64	−28
−27	17.49	1.2243	10.90	.00	10.90	−6.493	.183	−6.310	−170.9	841.0	413.10	−27
−26	18.55	1.2985	10.92	.00	10.92	−6.251	.194	−6.057	−170.4	892.1	438.20	−26

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds × 10^5	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure $p_s \times 10^5$: In. of Hg	Lb per Sq In.	Temp Deg F
−25	19.68	1.3776	10.95	0.00	10.95	−6.011	0.206	−5.805	−170.4	946.4	464.87	−25
−24	20.86	1.4602	10.97	.00	10.97	−5.770	.219	−5.551	−170.0	1003.	492.67	−24
−23	22.13	1.5491	11.00	.00	11.00	−5.529	.232	−5.297	−169.5	1064.	522.64	−23
−22	23.42	1.6394	11.02	.00	11.02	−5.288	.246	−5.042	−169.0	1126.	553.09	−22
−21	24.79	1.7353	11.05	.00	11.05	−5.047	.260	−4.787	−168.6	1192.	585.51	−21
−20	26.25	1.8375	11.07	0.00	11.07	−4.807	0.276	−4.531	−168.2	1262.0	619.89	−20
−19	27.81	1.9467	11.10	.00	11.10	−4.566	.292	−4.274	−167.7	1337.	656.73	−19
−18	29.45	2.0615	11.13	.00	11.13	−4.325	.310	−4.015	−167.2	1416.	695.54	−18
−17	31.12	2.1784	11.15	.00	11.15	−4.085	.327	−3.758	−166.8	1496.	734.84	−17
−16	32.95	2.3065	11.18	.00	11.18	−3.844	.347	−3.497	−166.3	1584.	778.06	−16
−15	34.84	2.4388	11.20	0.01	11.21	−3.604	0.367	−3.237	−165.9	1675.0	822.76	−15
−14	36.86	2.5802	11.23	.01	11.24	−3.363	.388	−2.975	−165.4	1772.	870.41	−14
−13	38.98	2.7286	11.25	.01	11.26	−3.123	.411	−2.712	−165.0	1874.	920.51	−13
−12	41.19	2.8833	11.28	.01	11.29	−2.883	.434	−2.449	−164.5	1980.	972.58	−12
−11	43.54	3.0478	11.30	.01	11.31	−2.642	.459	−2.183	−164.0	2093.	1028.1	−11
−10	45.98	3.2186	11.33	0.01	11.34	−2.402	0.485	−1.917	−163.6	2210.0	1085.6	−10
−9	48.58	3.4006	11.35	.01	11.36	−2.162	.513	−1.649	−163.1	2335.	1147.0	−9
−8	51.25	3.5875	11.38	.01	11.39	−1.921	.541	−1.380	−162.7	2463.	1209.8	−8
−7	52.06	3.6442	11.40	.01	11.41	−1.681	.570	−1.111	−162.2	2502.	1229.0	−7
−6	57.12	3.9984	11.43	.01	11.44	−1.441	.603	−0.838	−161.7	2745.	1348.3	−6
−5	60.30	4.2210	11.45	0.01	11.46	−1.201	0.637	−0.564	−161.3	2898.0	1423.5	−5
−4	63.57	4.4499	11.48	.01	11.49	−0.960	.672	−0.288	−160.8	3055.	1500.6	−4
−3	67.05	4.6935	11.50	.01	11.51	−0.720	.709	−0.011	−160.3	3222.	1582.6	−3
−2	70.69	4.9483	11.53	.01	11.54	−0.480	.748	+0.268	−159.9	3397.	1668.6	−2
−1	74.50	5.2150	11.55	.01	11.57	−0.240	.789	+0.549	−159.4	3580.	1758.5	−1
0	78.52	5.5000	11.58	0.01	11.59	0	0.832	+0.832	−158.9	3773.0	1853.3	0

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v'_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In. of Hg	Lb per Sq In.	
0	0.0007852	5.50	11.58	0.01	11.59	0.000	0.832	0.832	−158.9	0.03773	0.01853	0
1	.0008275	5.79	11.60	.02	11.62	.240	.877	1.117	−158.5	.03975	.01963	1
2	.0008714	6.10	11.63	.01	11.64	.480	.924	1.404	−158.0	.04186	.02056	2
3	.0009179	6.43	11.65	.02	11.67	.720	.974	1.694	−157.5	.04409	.02166	3
4	.0009671	6.77	11.68	.02	11.70	.960	1.026	1.986	−157.0	.04645	.02282	4
5	0.001017	7.12	11.70	0.02	11.72	1.200	1.080	2.280	−156.6	0.04886	0.02400	5
6	.001071	7.50	11.73	.02	11.75	1.440	1.137	2.577	−156.1	.05144	.02527	6
7	.001127	7.89	11.75	.02	11.77	1.680	1.197	2.877	−155.6	.05412	.02658	7
8	.001186	8.30	11.78	.02	11.80	1.920	1.260	3.180	−155.1	.05692	.02796	8
9	.001247	8.73	11.80	.03	11.83	2.160	1.336	3.486	−154.7	.05988	.02941	9
10	0.001311	9.18	11.83	0.02	11.85	2.400	1.395	3.795	−154.2	0.06295	0.03092	10
11	.001379	9.65	11.86	.02	11.88	2.640	1.468	4.108	−153.7	.06618	.03251	11
12	.001450	10.15	11.88	.03	11.91	2.880	1.544	4.424	−153.2	.06958	.03418	12
13	.001523	10.66	11.91	.02	11.93	3.120	1.622	4.742	−152.7	.07309	.03590	13
14	.001600	11.20	11.93	.03	11.96	3.359	1.705	5.064	−152.2	.07677	.03771	14
15	0.001682	11.77	11.96	0.03	11.99	3.599	1.793	5.392	−151.8	0.08067	0.03963	15
16	.001766	12.36	11.98	.03	12.01	3.839	1.883	5.722	−151.3	.08469	.04160	16
17	.001855	12.99	12.00	.04	12.04	4.079	1.979	6.058	−150.8	.08895	.04369	17
18	.001947	13.63	12.03	.04	12.07	4.319	2.078	6.397	−150.3	.09337	.04586	18
19	.002043	14.30	12.06	.03	12.09	4.559	2.182	6.741	−149.8	.09797	.04812	19
20	0.002144	15.01	12.08	0.04	12.12	4.798	2.290	7.088	−149.3	0.1028	0.05050	20
21	.002250	15.75	12.11	.04	12.15	5.038	2.405	7.443	−148.8	.1078	.05295	21
22	.002361	16.53	12.13	.05	12.18	5.278	2.524	7.802	−148.3	.1132	.05560	22
23	.002476	17.33	12.16	.04	12.20	5.518	2.648	8.166	−147.8	.1186	.05826	23
24	.002596	18.17	12.18	.05	12.23	5.758	2.778	8.536	−147.3	.1244	.06111	24
25	0.002722	19.05	12.21	0.05	12.26	5.998	2.914	8.912	−146.8	0.1304	0.06405	25
26	.002853	19.97	12.23	.06	12.29	6.237	3.055	9.292	−146.4	.1366	.06710	26
27	.002991	20.94	12.26	.06	12.32	6.477	3.205	9.682	−145.9	.1432	.07034	27
28	.003133	21.93	12.28	.06	12.34	6.717	3.358	10.075	−145.4	.1500	.07368	28
29	.003283	22.99	12.31	.06	12.37	6.957	3.520	10.477	−144.9	.1571	.07717	29
30	0.003439	24.07	12.33	0.07	12.40	7.197	3.689	10.886	−144.4	0.1645	0.08080	30
31	.003601	25.21	12.36	.07	12.43	7.437	3.865	11.302	−143.9	.1722	.08458	31
32	.003771	26.40	12.38	.08	12.46	7.677	4.049	11.726	−143.4	.1803	.08856	32
33	.003931	27.52	12.41	.08	12.49	7.917	4.222	12.139	+ 1.0	.1879	.09230	33
34	.004094	28.66	12.43	.08	12.51	8.157	4.399	12.556	+ 2.0	.1957	.09610	34

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
35	0.004262	29.83	12.46	0.08	12.54	8.397	4.582	12.979	3.0	0.20360	0.1000	35
36	.004438	31.07	12.48	.09	12.57	8.636	4.773	13.409	4.0	.21195	.1041	36
37	.004618	32.33	12.51	.09	12.60	8.876	4.969	13.845	5.0	.22050	.1083	37
38	.004803	33.62	12.53	.10	12.63	9.116	5.169	14.285	6.0	.22925	.1126	38
39	.004996	34.97	12.56	.10	12.66	9.356	5.380	14.736	7.0	.23842	.1171	39
40	0.005194	36.36	12.59	0.10	12.69	9.596	5.595	15.191	8.0	0.24778	0.1217	40
41	.005401	37.80	12.61	.11	12.72	9.836	5.821	15.657	9.1	.25755	.1265	41
42	.005616	39.31	12.64	.11	12.75	10.08	6.05	16.13	10.1	.26773	.1315	42
43	.005840	40.88	12.66	.12	12.78	10.32	6.30	16.62	11.1	.27832	.1367	43
44	.006069	42.48	12.69	.12	12.81	10.56	6.55	17.11	12.1	.28911	.1420	44
45	0.006306	44.14	12.71	0.13	12.84	10.80	6.81	17.61	13.1	0.30031	0.1475	45
46	.006553	45.87	12.74	.13	12.87	11.04	7.08	18.12	14.1	.31191	.1532	46
47	.006808	47.66	12.76	.14	12.90	11.28	7.36	18.64	15.1	.32393	.1591	47
48	.007072	49.50	12.79	.14	12.93	11.52	7.64	19.16	16.1	.33635	.1652	48
49	.007345	51.42	12.81	.15	12.96	11.76	7.94	19.70	17.1	.34917	.1715	49
50	0.007626	53.38	12.84	0.15	12.99	12.00	8.25	20.25	18.1	0.36241	0.1780	50
51	.007921	55.45	12.86	.16	13.02	12.23	8.57	20.80	19.1	.37625	.1848	51
52	.008226	57.58	12.89	.17	13.06	12.47	8.91	21.38	20.1	.39051	.1918	52
53	.008534	59.74	12.91	.18	13.09	12.71	9.24	21.95	21.1	.40496	.1989	53
54	.008856	61.99	12.94	.18	13.12	12.95	9.60	22.55	22.1	.42003	.2063	54
55	0.009192	64.34	12.96	0.19	13.15	13.19	9.96	23.15	23.1	0.43570	0.2140	55
56	.009536	66.75	12.99	.20	13.19	13.43	10.34	23.77	24.1	.45179	.2219	56
57	.009890	69.23	13.01	.21	13.22	13.67	10.73	24.40	25.1	.46828	.2300	57
58	.01026	71.82	13.04	.21	13.25	13.91	11.14	25.05	26.1	.48538	.2384	58
59	.01064	74.48	13.06	.23	13.29	14.15	11.55	25.70	27.1	.50310	.2471	59
60	0.01103	77.21	13.09	0.23	13.32	14.39	11.98	26.37	28.1	0.52142	0.2561	60
61	.01144	80.08	13.11	.24	13.35	14.63	12.43	27.06	29.1	.54035	.2654	61
62	.01186	83.02	13.14	.25	13.39	14.87	12.89	27.76	30.1	.55970	.2749	62
63	.01229	86.03	13.16	.26	13.42	15.11	13.37	28.48	31.1	.57985	.2848	63
64	.01274	89.18	13.19	.27	13.46	15.35	13.86	29.21	32.1	.60042	.2949	64
65	0.01320	92.40	13.21	0.28	13.49	15.59	14.37	29.96	33.1	0.62179	0.3054	65
66	.01368	95.76	13.24	.29	13.53	15.83	14.90	30.73	34.1	.64378	.3162	66
67	.01417	99.19	13.26	.31	13.57	16.07	15.44	31.51	35.1	.66638	.3273	67
68	.01468	102.8	13.29	.31	13.60	16.31	16.00	32.31	36.0	.68980	.3388	68
69	.01520	106.4	13.31	.33	13.64	16.55	16.57	33,12	37.0	.71382	3506	69

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

Table 6. Thermodynamic Properties of Moist Air[a], 29.921 In. Hg (Continued)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In. of Hg	Lb per Sq In.	
70	0.01574	110.2	13.34	0.34	13.68	16.79	17.17	33.96	38.0	0.73866	0.3628	70
71	.01631	114.2	13.37	.34	13.71	17.03	17.80	34.83	39.0	.76431	.3754	71
72	.01688	118.2	13.40	.35	13.75	17.27	18.43	35.70	40.0	.79058	.3883	72
73	.01748	122.4	13.42	.37	13.79	17.51	19.09	36.60	41.0	.81766	.4016	73
74	.01809	126.6	13.44	.39	13.83	17.75	19.76	37.51	42.0	.84555	.4153	74
75	0.01873	131.1	13.47	0.40	13.87	17.99	20.47	38.46	43.0	0.87448	0.4295	75
76	.01938	135.7	13.49	.42	13.91	18.23	21.19	39.42	44.0	.90398	.4440	76
77	.02005	140.4	13.52	.43	13.95	18.47	21.93	40.40	45.0	.93452	.4590	77
78	.02075	145.3	13.54	.45	13.99	18.71	22.71	41.42	46.0	.96588	.4744	78
79	.02147	150.3	13.57	.46	14.03	18.95	23.51	42.46	47.0	.99825	.4903	79
80	0.02221	155.5	13.59	0.49	14.08	19.19	24.32	43.51	48.0	1.0316	0.5067	80
81	.02298	160.9	13.62	.50	14.12	19.43	25.18	44.61	49.0	1.0661	.5236	81
82	.02377	166.4	13.64	.52	14.16	19.67	26.05	45.72	50.0	1.1013	.5409	82
83	.02459	172.1	13.67	.54	14.21	19.91	26.97	46.88	51.0	1.1377	.5588	83
84	.02543	178.0	13.69	.57	14.26	20.15	27.90	48.05	52.0	1.1752	.5772	84
85	0.02629	184.0	13.72	0.58	14.30	20.39	28.85	49.24	53.0	1.2135	0.5960	85
86	.02718	190.3	13.74	.60	14.34	20.63	29.84	50.47	54.0	1.2527	.6153	86
87	.02810	196.7	13.77	.62	14.39	20.87	30.87	51.74	55.0	1.2933	.6352	87
88	.02904	203.3	13.79	.65	14.44	21.11	31.91	53.02	56.0	1.3346	.6555	88
89	.03002	210.1	13.82	.66	14.48	21.35	33.00	54.35	57.0	1.3774	.6765	89
90	0.03102	217.1	13.84	0.69	14.53	21.59	34.11	55.70	58.0	1.4211	0.6980	90
91	.03205	224.4	13.87	.71	14.58	21.83	35.26	57.09	59.0	1.4661	.7201	91
92	.03312	231.8	13.89	.74	14.63	22.07	36.45	58.52	60.0	1.5125	.7429	92
93	.03421	239.5	13.92	.77	14.69	22.32	37.67	59.99	61.0	1.5600	.7662	93
94	.03535	247.5	13.94	.79	14.73	22.56	38.94	61.50	62.0	1.6088	.7902	94
95	0.03652	255.6	13.97	0.82	14.79	22.80	40.25	63.05	63.0	1.6591	0.8149	95
96	.03772	264.0	13.99	.85	14.84	23.04	41.58	64.62	64.0	1.7108	.8403	96
97	.03896	272.7	14.02	.88	14.90	23.28	42.97	66.25	65.0	1.7638	.8663	97
98	.04024	281.7	14.04	.91	14.95	23.52	44.40	67.92	66.0	1.8181	.8930	98
99	.04156	290.9	14.07	.94	15.01	23.76	45.87	69.63	67.0	1.8741	.9205	99
100	0.04293	300.5	14.10	0.97	15.07	24.00	47.00	71.40	68.0	1.9316	0.9487	100
101	.04433	310.3	14.12	1.00	15.12	24.24	48.97	73.21	69.0	1.9904	.9776	101
102	.04577	320.4	14.15	1.03	15.18	24.48	50.58	75.06	70.0	2.0507	1.0072	102
103	.04726	330.8	14.17	1.08	15.25	24.72	52.25	76.97	71.0	2.1128	1.0377	103
104	.04879	341.5	14.20	1.11	15.31	24.96	53.96	78.92	72.0	2.1763	1.0689	104

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In. of Hg	Lb per Sq In.	
105	0.05037	352.6	14.22	1.15	15.37	25.20	55.73	80.93	73.0	2.2414	1.1009	**105**
106	.05200	364.0	14.25	1.19	15.44	25.44	57.56	83.00	74.0	2.3084	1.1338	106
107	.05368	375.8	14.27	1.23	15.50	25.68	59.45	85.13	75.0	2.3770	1.1675	107
108	.05541	387.9	14.30	1.27	15.57	25.92	61.38	87.30	76.0	2.4473	1.2020	108
109	.05719	400.3	14.32	1.32	15.64	26.16	63.38	89.54	76.9	2.5196	1.2375	109
110	0.05904	413.3	14.35	1.36	15.71	26.40	65.46	91.86	77.9	2.5939	1.274	**110**
111	.06092	426.4	14.37	1.41	15.78	26.64	67.57	94.21	78.9	2.6692	1.311	111
112	.06292	440.4	14.39	1.46	15.85	26.88	69.82	96.70	79.9	2.7486	1.350	112
113	.06493	454.5	14.42	1.51	15.93	27.12	72.08	99.20	80.9	2.8280	1.389	113
114	.06700	469.0	14.45	1.55	16.00	27.36	74.40	101.76	81.9	2.9094	1.429	114
115	0.06913	483.9	14.47	1.61	16.08	27.60	76.80	104.40	82.9	2.9929	1.470	**115**
116	.07134	499.4	14.50	1.66	16.16	27.84	79.29	107.13	83.9	3.0784	1.512	116
117	.07361	515.3	14.52	1.72	16.24	28.08	81.84	109.92	84.9	3.1660	1.555	117
118	.07600	532.0	14.55	1.77	16.32	28.32	84.53	112.85	85.9	3.2576	1.600	118
119	.07840	548.8	14.57	1.84	16.41	28.56	87.24	115.80	86.9	3.3492	1.645	119
120	0.08093	566.5	14.60	1.90	16.50	28.80	90.09	118.89	87.9	3.4449	1.692	**120**
121	.08348	584.4	14.62	1.96	16.58	29.04	92.97	122.01	88.9	3.5406	1.739	121
122	.08616	603.1	14.65	2.03	16.68	29.28	95.99	125.27	89.9	3.6404	1.788	122
123	.08892	622.4	14.67	2.10	16.77	29.52	99.11	128.63	90.9	3.7422	1.838	123
124	.09175	642.3	14.70	2.17	16.87	29.76	102.30	132.06	91.9	3.8460	1.889	124
125	0.09466	662.6	14.72	2.24	16.96	30.00	105.59	135.59	92.9	3.9519	1.941	**125**
126	.09770	683.9	14.75	2.31	17.06	30.24	109.02	139.26	93.9	4.0618	1.995	126
127	.1008	705.6	14.77	2.40	17.17	30.48	112.53	143.01	94.9	4.1718	2.049	127
128	.1040	728.0	14.80	2.47	17.27	30.72	116.15	146.87	95.9	4.2858	2.105	128
129	.1074	751.8	14.83	2.55	17.38	30.96	120.00	150.96	96.9	4.4039	2.163	129
130	0.1107	774.9	14.85	2.64	17.49	31.20	123.73	154.93	97.9	4.5220	2.221	**130**
131	.1143	800.1	14.88	2.73	17.61	31.45	128.81	159.26	98.9	4.6441	2.281	131
132	.1180	826.0	14.90	2.83	17.73	31.69	131.99	163.68	99.9	4.7703	2.343	132
133	.1218	852.6	14.93	2.92	17.85	31.93	136.31	168.24	100.9	4.8986	2.406	133
134	.1257	879.9	14.95	3.02	17.97	32.17	140.72	172.89	101.9	5.0289	2.470	134
135	0.1297	907.9	14.98	3.12	18.10	32.41	145.26	177.67	102.9	5.1633	2.536	**135**
136	.1339	937.3	15.00	3.23	18.23	32.65	150.02	182.67	103.9	5.2997	2.603	136
137	.1382	967.4	15.03	3.33	18.36	32.89	154.91	187.80	104.9	5.4402	2.672	137
138	.1427	998.9	15.05	3.45	18.50	33.13	160.01	193.14	105.9	5.5827	2.742	138
139	.1473	1031.1	15.08	3.57	18.65	33.37	165.24	198.61	106.9	5.7293	2.814	139

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In. of Hg	Lb per Sq In.	
140	0.1521	1064.7	15.10	3.69	18.79	33.61	170.69	204.30	107.9	5.8779	2.887	**140**
141	.1570	1099.0	15.13	3.81	18.94	33.85	176.26	210.11	108.9	6.0306	2.962	141
142	.1622	1135.4	15.15	3.95	19.10	34.09	182.17	216.26	109.9	6.1874	3.039	142
143	.1675	1172.5	15.18	4.08	19.26	34.33	188.20	222.53	110.9	6.3482	3.118	143
144	.1730	1211.0	15.20	4.23	19.43	34.57	194.45	229.02	111.9	6.5111	3.198	144
145	0.1787	1250.9	15.23	4.37	19.60	34.81	200.95	235.76	112.9	6.6781	3.280	**145**
146	.1846	1292.2	15.25	4.53	19.78	35.05	207.66	242.71	113.9	6.8471	3.363	146
147	.1908	1335.6	15.28	4.68	19.96	35.29	214.73	250.02	114.9	7.0222	3.449	147
148	.1971	1379.7	15.30	4.85	20.15	35.53	221.90	257.43	115.9	7.1993	3.536	148
149	.2037	1425.9	15.33	5.02	20.35	35.77	229.43	265.20	116.9	7.3805	3.625	149
150	0.2105	1473.5	15.35	5.20	20.55	36.02	237.17	273.19	117.9	7.5658	3.716	**150**
151	.2176	1523.2	15.38	5.38	20.76	36.26	245.28	281.54	118.9	7.7551	3.809	151
152	.2250	1575.0	15.40	5.57	20.97	36.50	253.71	290.21	119.9	7.9485	3.904	152
153	.2327	1628.9	15.43	5.77	21.20	36.74	262.51	299.25	120.9	8.1460	4.001	153
154	.2407	1684.9	15.45	5.98	21.43	36.98	271.63	308.61	121.9	8.3476	4.100	154
155	0.2490	1743.0	15.48	6.19	21.67	37.22	281.12	318.34	122.9	8.5532	4.201	**155**
156	.2577	1803.9	15.50	6.43	21.93	37.46	291.05	328.51	123.9	8.7650	4.305	156
157	.2667	1866.9	15.53	6.66	22.19	37.70	301.24	339.04	124.9	8.9788	4.410	157
158	.2761	1932.7	15.56	6.90	22.46	37.94	312.08	350.02	125.9	9.1986	4.518	158
159	.2858	2000.6	15.58	7.16	22.74	38.18	323.18	361.36	126.9	9.4206	4.627	159
160	0.2961	2072.7	15.61	7.42	23.03	38.43	334.95	373.38	127.9	9.6486	4.739	**160**
161	.3067	2146.9	15.63	7.70	23.33	38.67	347.09	385.76	128.9	9.8807	4.853	161
162	.3179	2225.3	15.66	7.99	23.65	38.91	359.89	398.80	129.9	10.119	4.970	162
163	.3295	2306.5	15.68	8.30	23.98	39.15	373.19	412.34	130.9	10.361	5.089	163
164	.3416	2391.2	15.71	8.62	24.33	39.39	387.03	426.42	131.9	10.608	5.210	164
165	0.3544	2480.8	15.73	8.96	24.69	39.63	401.71	441.34	132.9	10.860	5.334	**165**
166	.3677	2573.9	15.76	9.31	25.07	39.87	416.94	456.81	133.9	11.117	5.460	166
167	.3817	2671.9	15.78	9.68	25.46	40.11	433.00	473.11	134.9	11.379	5.589	167
168	.3964	2774.8	15.81	10.07	25.88	40.35	449.83	490.18	135.9	11.646	5.720	168
169	.4118	2882.6	15.83	10.48	26.31	40.59	467.52	508.11	136.9	11.919	5.854	169
170	0.4280	2996.0	15.86	10.91	26.77	40.83	486.08	526.91	137.9	12.196	5.990	**170**
171	.4451	3115.7	15.88	11.36	27.24	41.07	505.72	546.79	138.9	12.480	6.130	171
172	.4631	3241.7	15.91	11.83	27.74	41.32	526.36	567.68	139.9	12.770	6.272	172
173	.4821	3374.7	15.93	12.35	28.28	41.56	548.20	589.76	140.9	13.065	6.417	173
174	.5022	3515.4	15.96	12.88	28.84	41.80	571.25	613.05	141.9	13.366	6.565	174

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONCLUDED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} (v_s-v_a)	Saturated Mixture v_s	Dry Air h_a	h_{as} (h_s-h_a)	Saturated Mixture h_s		In. of Hg	Lb per Sq In.	
175	0.5235	3664.5	15.98	13.45	29.43	42.04	595.74	637.78	142.9	13.674	6.716	**175**
176	.5459	3821.3	16.01	14.04	30.05	42.28	621.45	663.73	143.9	13.985	6.869	176
177	.5697	3987.9	16.03	14.68	30.71	42.52	648.83	691.35	144.9	14.303	7.025	177
178	.5949	4164.3	16.06	15.35	31.41	42.76	677.77	720.53	145.9	14.627	7.184	178
179	.6215	4350.5	16.08	16.07	32.15	43.00	708.39	751.39	146.9	14.954	7.345	179
180	0.6501	4550.7	16.11	16.83	32.94	43.24	741.24	784.48	147.9	15.290	7.510	**180**
181	.6805	4763.5	16.13	17.65	33.78	43.49	776.25	819.74	148.9	15.632	7.678	181
182	.7131	4991.7	16.16	18.52	34.68	43.73	813.72	857.45	149.9	15.981	7.849	182
183	.7481	5236.7	16.18	19.47	35.65	43.97	854.03	898.00	150.9	16.337	8.024	183
184	.7854	5497.8	16.21	20.46	36.67	44.21	896.93	941.14	151.9	16.697	8.201	184
185	0.8258	5780.6	16.23	21.55	37.78	44.45	943.48	987.93	152.9	17.066	8.382	**185**
186	.8693	6085.1	16.26	22.72	38.98	44.69	993.52	1038.21	153.9	17.440	8.566	186
187	.9162	6413.4	16.28	23.99	40.27	44.93	1047.58	1092.51	154.9	17.821	8.753	187
188	.9673	6771.1	16.31	25.36	41.67	45.18	1106.40	1151.58	155.9	18.210	8.944	188
189	1.0227	7158.9	16.34	26.70	43.04	45.42	1170.62	1216.04	157.0	18.605	9.138	189
190	1.083	7581.0	16.36	28.49	44.85	45.66	1239.71	1285.37	158.0	19.008	9.336	**190**
191	1.150	8050.0	16.39	30.29	46.68	45.90	1316.98	1362.88	159.0	19.419	9.538	191
192	1.224	8568.0	16.41	32.29	48.70	46.14	1402.21	1448.35	160.0	19.839	9.744	192
193	1.306	9142.0	16.44	34.49	50.93	46.38	1496.81	1543.19	161.0	20.266	9.954	193
194	1.397	9779.0	16.46	36.96	53.42	46.62	1601.66	1648.28	162.0	20.702	10.168	194
195	1.499	10493.0	16.49	39.71	56.20	46.86	1719.35	1766.21	163.0	21.144	10.385	**195**
196	1.613	11291.0	16.51	42.80	59.31	47.10	1850.76	1897.86	164.0	21.592	10.605	196
197	1.742	12194.0	16.54	46.31	62.85	47.34	1999.64	2046.98	165.0	22.048	10.829	197
198	1.890	13230.0	16.56	50.32	66.88	47.59	2170.29	2217.88	166.0	22.512	11.057	198
199	2.061	14427.0	16.59	54.95	71.54	47.83	2367.68	2415.51	167.0	22.984	11.289	199
200	2.261	15827.0	16.61	60.38	76.99	48.07	2598.34	2646.41	168.0	23.465	11.525	**200**

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

This statement lacks thermodynamic soundness due to actual departures from Dalton's Law, but has real practical merit as an approximation.

Example 3. Calculate the humidity ratio of saturated moist air at 68 F, 30 in. Hg.

Solution. The saturation pressure of pure water at 68 F from Table 6 is 0.68980 in. Hg; hence,

$$W_s = \frac{0.62193 \times 0.68980}{29.3102} = 0.01464 \text{ (pound per pound of dry air).}$$

It is also frequently stated that moist air is saturated when the space (volume) occupied by it contains the *maximum weight of water vapor* at the given temperature. This means that any additional water would have to be in the liquid or solid phase. But under proper circumstances the water vapor can be *supersaturated*, in which case the space occupied by the mixture can contain more than the *maximum possible* water vapor. The statement is therefore meaningless as a definition of saturation.

A precise definition must necessarily refer to the coexistence of at least two distinct phases, say, liquid and vapor. These can only coexist in stable equilibrium if evaporation of the liquid or condensation of the vapor under conditions of constant total volume and constant total internal energy would have to involve a decrease of total entropy. This would be the situation if, and only if, the pressure, the temperature, and each component chemical potential has the same value in each phase.

In the case of moist air, the general conditions for saturation previously stated can be deduced from Equation 9 together with available data on the solubility of air in the liquid. They can be reduced to the form,

$$W_s = 0.62193 \frac{p_s'}{P - p_s'} \qquad (13a)$$

where

$$p_s' = \frac{(PF)\ (DF)}{(RF)} p_s \qquad (13b)$$

The liquid (or solid) phase will contain a small amount of dissolved air and the Raoult factor (*RF*) expresses the effect of this dissolved air in lowering the vapor pressure in accordance with Raoult's Law. The Poynting factor (*PF*) accounts for the fact that the very presence of dry air requires the liquid (or solid) to support a higher pressure at saturation than it would if no dry air were present. The Dalton factor (*DF*) expresses the effect of intermolecular forces in the vapor phase. All three factors depend more or less on pressure as well as on temperature.

The Dalton factor is the only one of the three factors listed here which cannot at present be calculated with reasonable certainty due to ignorance regarding the interaction constant A_{aw}. Its order of magnitude can be *guessed*, however, by assuming a simple combination rule which has received some confirmation on mixtures similar to moist air, namely,

$$A_{aw} = \frac{A_{aa} + A_{ww}}{2}$$

At 68 F, 30 in. Hg, for example,

$$p_s' = \frac{1.00073 \times 1.05863}{1.00002} p_s$$

These figures suggest that Dalton's Law may not be the close approximation it is generally assumed to be. However, until the Dalton factor can be measured, it is better to ignore (call it unity) than guess it. This procedure has been followed in computing the values in Table 6.

Relative Humidity

The ratio of actual humidity ratio W to the saturation humidity ratio W_s corresponding to the actual temperature and the observed pressure is denoted by the symbol μ and may be called alternatively *degree of saturation* or *percent saturation;* thus,

$$W = \mu \cdot W_s \tag{14}$$

Example 4. Air is to be maintained at 70 F, 40 per cent saturation when outside air is at 0 F, 70 per cent. The observed pressure may be taken to be 29.921 in. Hg. Find the weight of water to be added to each pound of dry air using Table 6.

Solution. The desired humidity ratio is $0.40 \times 0.01574 = 0.006296$ while that of outside air is $0.70 \times 0.0007852 = 0.000550$. Hence the weight of water to be added is $0.006296 - 0.000550 = 0.005746$ lb per pound dry air.

Under Dalton's Law the water vapor exerts a partial pressure p_w which may be calculated from the given humidity ratio W and the observed pressure P by means of Equation 11. The ratio of this partial pressure p_w to the saturation pressure of pure water p_s corresponding to the actual temperature is called *relative humidity* and may be denoted by the symbol Φ; thus,

$$\Phi = \frac{p_w}{p_s} \tag{15}$$

The relation between μ and Φ is obtained directly from Equation 11 and 12 and is

$$\mu = \left(\frac{P - p_s}{P - p_w}\right) \Phi \tag{15a}$$

whence it is clear that for ordinary temperatures where p_s and therefore p_w are small compared with P, the two are approximately equal.

As an aid in quickly translating degree of saturation μ into relative humidity Φ or vice versa, the following empirical equation may be substituted for Equation 15a:

$$\Phi - \mu = \alpha \cdot \mu (1 - \mu) \tag{15b}$$

where α depends upon temperature *for standard atmospheric pressure,* as shown by the values in Table 7.

TABLE 7. PERCENTAGE DIFFERENCES CORRESPONDING TO TEMPERATURE FOR EQUATION 15b

TEMP F t	PER CENT α	TEMP F t	PER CENT α	TEMP F t	PER CENT α	TEMP F t	PER CENT α
5	0.16	30	0.55	55	1.47	80	3.51
10	0.21	35	0.68	60	1.76	85	4.14
15	0.27	40	0.83	65	2.10	90	4.86
20	0.34	45	1.01	70	2.50	95	5.70
25	0.44	50	1.22	75	2.97	100	6.67

For example, corresponding to a degree of saturation of 40 per cent at 100 F the difference between relative humidity and degree of saturation is $6.67 \times 0.40 \times 0.60 = 1.6$ per cent; hence the relative humidity itself is 41.6 per cent.

Dew-point

If moist air is cooled at constant humidity ratio W and constant observed pressure P, a temperature will be reached at which the air just becomes saturated and formation of a liquid (or solid) phase just commences. This temperature is called the *dew-point* corresponding to the given humidity ratio and observed pressure.

Example 5. Find the dew-point of the humidified air of Example 4.

Solution. The given humidity ratio is 0.006296 which is the saturation value at 44.96 F (Table 6, assuming the total pressure to be 29.921 in. Hg). This is therefore the dew-point of the humidified air.

Example 6. Find the degree of saturation of air having a temperature of 90 F, a dew-point of 60 F.

Solution. Assuming the total pressure to be 29.921 in. Hg, the humidity ratio is given in Table 6 as 0.01103 lb per pound dry air. The saturation humidity ratio at 90 F is 0.03102 lb per pound dry air; hence the degree of saturation is $0.01103 \div 0.03102 = 0.355$ or 35.5 per cent.

Volume

The volume of moist air *per pound of dry air* contained in it is a very useful quantity. It should not be called specific volume; for the adjective *specific* should properly refer to volume per pound of *mixture*. Using Equations 10a and 14 an expression for the volume per pound of dry air is obtained, namely,

$$v = \frac{B_a T}{P} + \mu \left(\frac{W_s B_w T}{P}\right) \tag{16}$$

Example 7. Find the volume (per pound of dry air) of the humidified air of Example 4.

Solution. $v = \left(\frac{53.35 \times 529.7}{29.92 \times 0.49115 \times 144}\right) + 0.40 \left(\frac{0.01574 \times 85.78 \times 529.7}{29.92 \times 0.49115 \times 144}\right)$
$= 13.354 + 0.40 \times 0.338 = 13.489$ cu ft per pound dry air.

Equation 16 is linear in degree of saturation μ and of the form

$$v = v_a + \mu\, v_{as} \tag{17}$$

where v_a denotes specific volume of dry air at temperature T and pressure P; and v_{as} denotes the difference between this and the volume of the saturated mixture per pound of dry air v_s. Strict linearity is, of course, a result of the use of Dalton's Law; but it is expected that it can be retained as a very close approximation even when the abandonment of Dalton's Law becomes possible.

Example 8. Work Example 7 using Table 6.

Solution. $v = 13.34 + (0.40 \times 0.34) = 13.48$ cu ft per pound dry air.

By putting $\mu = 1$ (100 per cent saturation) in Equation 16 an expression for v_s, the volume of saturated air per pound of dry air, is obtained. Values for standard atmospheric pressure (29.921 in. Hg) are listed in Table 6. Often it is preferred to express this information in terms of *density*, that

is, weight of saturated air per unit volume. This can easily be done by dividing v_s (volume of saturated air per pound of dry air) *into* $(1 + w_s)$ (*weight* of saturated air per pound of dry air). Thus, at 100 F, 29.921 in. Hg, the density of saturated air is, from Table 6, 1.04293 ÷ 15.07 = 0.06921 lb per cubic foot.

Values in Table 9 are intended to aid in determining the density of saturated air at different pressures. Values for temperatures and pressures other than those listed can be obtained by linear interpolation which is aided by the next to last column of figures. Thus, at 100 F, 29.921 in. Hg, the density of saturated air is, from Table 9, 0.06818 + (4.21 × 0.00024) = 0.06919 lb per cubic foot, in approximate agreement with Table 6.

A column of figures is included in Table 9 giving the approximate average increase in density per degree wet-bulb depression. This makes it easy to calculate a value for the density of moist air taking into account its moisture content as well as its temperature and pressure.

Enthalpy

Thermodynamically, Equation 10a implies that the specific enthalpies of dry air and water vapor are independent of pressure and that the enthalpy of moist air (per pound of dry air) is the sum of separate contributions from the dry air and water vapor according to the simple equation

$$h = h_a + \mu\,(W_s h_w) \tag{18}$$

Equation 18 is also linear in degree of saturation μ and of the form

$$h = h_a + \mu\, h_{as} \tag{19}$$

where h_a denotes the specific enthalpy of dry air at the given temperature and total pressure; and h_{as} denotes the difference between this and the enthalpy of the saturated mixture per pound of dry air h_s. Provisional values are listed in Table 6.

Example 9. Find the enthalpy (per pound of dry air) of air at 96 F, 60 per cent saturation and 29.921 in. Hg.

Solution. Using Table 6, h = 23.04 + (0.60 × 41.58) = 47.99 Btu per pound dry air.

Thermodynamic Wet-bulb Temperature

If liquid (or solid) water be injected into an air stream it will evaporate and thus increase the humidity ratio of the air. Enough water may be injected to saturate the air. If the process is one of steady flow with *observed pressure constant*; if it is *adiabatic*; and if the temperature at which the air reaches saturation *coincides* with the temperature of the liquid (or solid) as added; then the common temperature is called *thermodynamic wet-bulb temperature.* This lengthy definition is easily visualized by referring to Fig. 1 in which h'_w denotes the specific enthalpy of the liquid (or solid) as injected.

The process being adiabatic, weight and energy accountings give

$$h_1 + (W_s - W_1)\, h'_w = h_s \tag{20}$$

If the temperature of the saturated air at the leaving section *coincides*

with that of the injected liquid (or solid), then W_s, h'_w and h_s are functions of a single temperature t' which can therefore be determined by solving (20). This is the thermodynamic wet-bulb temperature corresponding to conditions at the entering section.

Example 10. Find the thermodynamic wet-bulb temperature of dry air at 80 F and 29.921 in. Hg.

Solution. Using Table 6, the equation to be solved is $19.19 + (W_s - 0)\, h'_w = h_s$.

A trial value is obtained by ignoring the small quantity $(W_s - 0)\, h'_w$; it is 48 F corresponding to $h_s = 19.19$ Btu per pound dry air. A final value of 48.26 F is then obtained from $h_s = 19.19 + (0.007072 \times 16.1) = 19.30$ Btu per pound dry air.

Example 11. Find the degree of saturation of moist air at 90 F dry-bulb, 70 F wet-bulb and 29.921 in. Hg.

Solution. Using Table 6, the equation to be solved is $(21.59 + 34.11\,\mu) + (0.01574 - 0.03102\mu) \times 38.0 = 33.96$ from which

$$\mu = 11.77 \div 32.93 = 0.357 \text{ or } 35.7 \text{ per cent.}$$

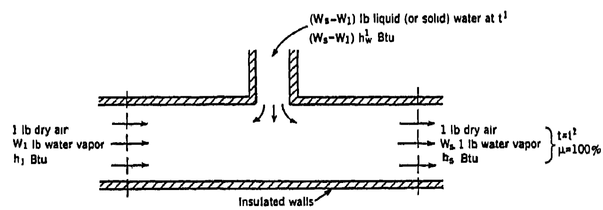


FIG 1. DIAGRAM ILLUSTRATING THERMODYNAMIC WET-BULB TEMPERATURE

It is important to note in connection with Equation 20 that the enthalpy per pound dry air is not constant along a line of constant thermodynamic wet-bulb temperature on account of the term $(W_s - W_1)\, h'_w$. In rough calculations, however, it is usually legitimate to ignore this term.

Thermodynamic wet-bulb is an important property of moist air because it is approximately the temperature indicated by the wet-bulb psychrometer. This instrument consists of a thermometer with its bulb covered with gauze moistened with clean liquid water. It is whirled through the air until the thermometer reads a steady temperature. At this point, the temperature of the liquid evaporating from the wetted surface has adjusted itself so that the air immediately in contact with the liquid is brought to saturation at the same temperature. Unfortunately, the mixing taking place beyond the liquid surface is not adiabatic; for one reason because the wet-bulb *sees* objects at dry-bulb temperature and considerable heat is transferred by radiation. Also there are other reasons why the readings of the psychrometer depend upon the design of the instrument, the velocity of the air stream in which it is placed, and other factors. Therefore wet-bulb temperature as indicated by the psychrometer cannot be regarded as a thermodynamic property; in fact, the approximate agreement with thermodynamic wet-bulb temperature in the case of moist air has been shown to be largely fortuitous.

Mollier Diagram

A thermodynamic analysis of any air conditioning process consists in writing: (1) a weight balance for the dry air; (2) a weight balance for the water; (3) an energy balance. The first is reduced to its simplest form by basing all quantities on one pound of dry air. The second is the most simply expressed in terms of humidity ratio, or weight of water per pound of dry air. Since most air conditioning processes are of the steady flow type in which the thermal energy convected with the fluid is its enthalpy, the third is most simply expressed in terms of enthalpy per pound of dry air. It is clear, therefore, that humidity ratio W and enthalpy per pound of dry air h are fundamental coordinates. Their use for the purpose of graphical representation is due to Mollier. A convenient modification of the Mollier diagram devised by Goff is obtained by taking humidity ratio W as ordinate and *reduced enthalpy* $(h - 1000W)$ as abscissa, as shown in the chart enclosed in the envelope attached to the inside back cover of this book.

The reasons for the use of the difference $(h - 1000W)$ as abscissa instead of h itself in the Mollier Diagram for Moist Air are the following: (1) it amounts to plotting on oblique coordinates and thus reduces to convenient proportions a diagram which would otherwise take the form of a scroll; (2) by the choice of the factor 1000 the necessary multiplication reduces to shifting the decimal point; (3) the ease with which the ordinate W can be multiplied by 1000 and added to the abscissa to obtain the enthalpy h makes it unnecessary to complicate the chart by a family of isenthalpic lines.

In the Mollier diagram, the lines inclined upward and slightly to the right are lines of constant (dry-bulb) temperature. They are straight under Dalton's Law but actually have slight curvature. The lines inclined upward to the left are lines of constant thermodynamic wet-bulb and are straight by definition. The dry-bulb and wet-bulb lines meet at the saturation curve and coincide in the region to the left of this curve. This region is divided into three subregions by the narrow *wedge* with apex at the junction of the 32 F wet-bulb and dry-bulb lines. Above the wedge, the mixture consists of two distinct phases, saturated vapor and saturated liquid. At point A, for example, the temperature is 60 F and the vapor phase contains 0.01103 pounds of water vapor per pound of dry air from Table 6. From the Mollier diagram, $W = 0.016$ lb, leaving 0.00497 lb per pound dry air in the liquid phase. The total enthalpy of the mixture is $10.51 + (1000 \times 0.016) = 26.51$ Btu per pound dry air of which $15.34 + (1000 \times 0.01103) = 26.37$ Btu per pound dry air is contributed by the vapor phase.

Within the wedge, the mixture consists of three distinct phases, saturated vapor, saturated solid and saturated liquid. The temperature is 32 F; and the relative proportions of the three phases depend upon the location of the state point within the wedge. Below the wedge, the mixture consists of saturated vapor and saturated solid.

The curved lines in the single vapor-phase region to the right of the saturation curve are lines of constant per cent saturation. Lines of constant dew-point are, of course, horizontal straight lines of constant humidity ratio. At point B, for example, the dry-bulb temperature is

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a]

Abs. Press. In. Hg. p	Temp F t	Specific Volume: Sat. Liquid v_f	Specific Volume: Sat. Vapor v_g	Enthalpy: Sat. Liquid h_f	Enthalpy: Evap. h_{fg}	Enthalpy: Sat. Vapor h_g	Entropy: Sat. Liquid s_f	Entropy: Evap. s_{fg}	Entropy: Sat. Vapor s_g	Abs. Press. In. Hg. p
0.25	40.23	0.01602	2423.7	8.28	1071.1	1079.4	0.0166	2.1423	2.1589	0.25
0.50	58.80	0.01604	1256.4	26.86	1060.6	1087.5	0.0532	2.0453	2.0985	0.50
0.75	70.43	0.01606	856.1	38.47	1054.0	1092.5	0.0754	1.9881	2.0635	0.75
1.00	79.03	0.01608	652.3	47.05	1049.2	1096.3	0.0914	1.9473	2.0387	1.00
1.5	91.72	0.01611	444.9	59.71	1042.0	1101.7	0.1147	1.8894	2.0041	1.5
2	101.14	0.01614	339.2	69.10	1036.6	1105.7	0.1316	1.8481	1.9797	2
4	125.43	0.01622	176.7	93.34	1022.7	1116.0	0.1738	1.7476	1.9214	4
6	140.78	0.01630	120.72	108.67	1013.6	1122.3	0 1996	1.6881	1.8877	6
8	152.24	0.01635	92.16	120.13	1006.9	1127.0	0.2186	1.6454	1.8640	8
10	161.49	0.01640	74.76	129.38	1001.4	1130.8	0.2335	1.6121	1.8456	10
12	169.28	0.01644	63.03	137.18	996.7	1133.9	0.2460	1.5847	1.8307	12
14	176.05	0.01648	54.55	143.96	992.6	1136.6	0.2568	1.5613	1.8181	14
16	182.05	0.01652	48.14	149.98	988.9	1138.9	0.2662	1.5410	1.8072	16
18	187.45	0.01655	43.11	155.39	985.7	1141.1	0.2746	1.5231	1.7977	18
20	192.37	0.01658	39.07	160.33	982.7	1143.0	0.2822	1.5069	1.7891	20
22	196.90	0.01661	35.73	164.87	979.8	1144.7	0.2891	1.4923	1.7814	22
24	201.09	0.01664	32.94	169.09	977.2	1146.3	0.2955	1.4789	1.7744	24
26	205.00	0.01667	30.56	173.02	974.8	1147.8	0.3014	1.4665	1.7679	26
28	208.67	0.01669	28.52	176.72	972.5	1149.2	0.3069	1.4550	1.7619	28
30	212.13	0.01672	26.74	180.19	970.3	1150.5	0.3122	1.4442	1.7564	30
Lb/Sq In.										Lb/Sq In.
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	14.696
16	216.32	0.01674	24.75	184.42	967.6	1152.0	0.3184	1.4313	1.7497	16
18	222.41	0.01679	22.17	190.56	963.6	1154.2	0.3275	1.4128	1.7403	18
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	20
22	233.07	0.01687	18.375	201.33	956.8	1158.1	0.3431	1.3811	1.7242	22
24	237.82	0.01691	16.938	206.14	953.7	1159.8	0.3500	1.3672	1.7172	24
26	242.25	0.01694	15.715	210.62	950.7	1161.3	0.3564	1.3544	1.7108	26
28	246.41	0.01698	14.663	214.83	947.9	1162.7	0.3623	1.3425	1.7048	28
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	30
32	254.05	0.01704	12.940	222.59	942.8	1165.4	0.3733	1.3209	1.6941	32
34	257.58	0.01707	12.226	226.18	940.3	1166.5	0.3783	1.3110	1.6893	34
36	260.95	0.01709	11.588	229.60	938.0	1167.6	0.3831	1.3017	1.6848	36
38	264.16	0.01712	11.015	232.89	935.8	1168.7	0.3876	1.2929	1.6805	38
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	40
42	270.21	0.01717	10.029	239.04	931.6	1170.7	0.3960	1.2764	1.6724	42
44	273.05	0.01720	9.601	241.95	929.6	1171.6	0.4000	1.2687	1.6687	44
46	275.80	0.01722	9.209	244.75	927.7	1172.4	0.4038	1.2613	1.6652	46
48	278.45	0.01725	8.848	247.47	925.8	1173.3	0.4075	1.2542	1.6617	48
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	50
52	283.49	0.01729	8.208	252.63	922.2	1174.8	0.4144	1.2409	1.6553	52
54	285.90	0.01731	7.922	255.09	920.5	1175.6	0.4177	1.2346	1.6523	54
56	288.23	0.01733	7.656	257.50	918.8	1176.3	0.4209	1.2285	1.6494	56
58	290.50	0.01736	7.407	259.82	917.1	1176.9	0.4240	1.2226	1.6466	58
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	60
62	294.85	0.01740	6.957	264.30	913.9	1178.2	0.4300	1.2112	1.6412	62
64	296.94	0.01742	6.752	266.45	912.3	1178.8	0.4328	1.2059	1.6387	64
66	298.99	0.01744	6.560	268.55	910.8	1179.4	0.4356	1.2006	1.6362	66
68	300.98	0.01746	6.378	270.60	909.4	1180.0	0.4383	1.1955	1.6338	68
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	70
72	304.83	0.01750	6.044	274.57	906.5	1181.1	0.4435	1.1857	1.6292	72
74	306.68	0.01752	5.890	276.49	905.1	1181.6	0.4460	1.1810	1.6270	74
76	308.50	0.01754	5.743	278.37	903.7	1182.1	0.4484	1.1764	1.6248	76
78	310.29	0.01755	5.604	280.21	902.4	1182.6	0.4508	1.1720	1.6228	78
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	80
82	313.74	0.01759	5.346	283.79	899.7	1183.5	0.4554	1.1633	1.6187	82
84	315.42	0.01761	5.226	285.53	898.5	1184.0	0.4576	1.1592	1.6168	84
86	317.07	0.01762	5.111	287.24	897.2	1184.4	0.4598	1.1551	1.6149	86
88	318.68	0.01764	5.001	288.91	895.9	1184.8	0.4620	1.1510	1.6130	88
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	90
92	321.83	0.01768	4.796	292.18	893.5	1185.7	0.4661	1.1433	1.6094	92
94	323.36	0.01769	4.699	293.78	892.3	1186.1	0.4682	1.1394	1.6076	94
96	324.87	0.01771	4.606	295.34	891.1	1186.4	0.4702	1.1358	1.6060	96
98	326.35	0.01772	4.517	296.89	889.9	1186.8	0.4721	1.1322	1.6043	98
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	100
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694	150
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	200
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	300
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	400
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	500

[a]Reprinted by permission from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, published by John Wiley and Sons, Inc., 1936 edition.

TABLE 8. PROPERTIES OF SATURATED STEAM: TEMPERATURE TABLE[a]

Temp F t	Abs. Pressure		Specific Volume			Enthalpy			Entropy			Temp F t
	Lb per Sq In. p	In. Hg p	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid S_f	Evap. S_{fg}	Sat. Vapor S_g	
32	0.08854	0.1803	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	32
33	0.09223	0.1878	0.01602	3180	3180	1.01	1075.2	1076.2	0.0020	2.1821	2.1841	33
34	0.09603	0.1955	0.01602	3061	3061	2.02	1074.7	1076.7	0.0041	2.1764	2.1805	34
35	0.09995	0.2035	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770	35
36	0.10401	0.2118	0.01602	2837	2837	4.03	1073.6	1077.6	0.0081	2.1654	2.1735	36
37	0.10821	0.2203	0.01602	2732	2732	5.04	1073.0	1078.0	0.0102	2.1598	2.1700	37
38	0.11256	0.2292	0.01602	2632	2632	6.04	1072.4	1078.4	0.0122	2.1544	2.1666	38
39	0.11705	0.2383	0.01602	2536	2536	7.04	1071.9	1078.9	0.0142	2.1489	2.1631	39
40	0.12170	0.2478	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597	40
41	0.12652	0.2576	0.01602	2356	2356	9.05	1070.7	1079.7	0.0182	2.1381	2.1563	41
42	0.13150	0.2677	0.01602	2271	2271	10.05	1070.1	1080.2	0.0202	2.1327	2.1529	42
43	0.13665	0.2782	0.01602	2190	2190	11.06	1069.5	1080.6	0.0222	2.1274	2.1496	43
44	0.14199	0.2891	0.01602	2112	2112	12.06	1068.9	1081.0	0.0242	2.1220	2.1462	44
45	0.14752	0.3004	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	45
46	0.15323	0.3120	0.01602	1964.3	1964.3	14.06	1067.8	1081.9	0.0282	2.1113	2.1395	46
47	0.15914	0.3240	0.01603	1895.1	1895.1	15.07	1067.3	1082.4	0.0302	2.1060	2.1362	47
48	0.16525	0.3364	0.01603	1828.6	1828.6	16.07	1066.7	1082.8	0.0321	2.1008	2.1329	48
49	0.17157	0.3493	0.01603	1764.7	1764.7	17.07	1066.1	1083.2	0.0341	2.0956	2.1297	49
50	0.17811	0.3626	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264	50
51	0.18486	0.3764	0.01603	1644.2	1644.2	19.07	1065.0	1084.1	0.0380	2.0852	2.1232	51
52	0.19182	0.3906	0.01603	1587.6	1587.6	20.07	1064.4	1084.5	0.0400	2.0799	2.1199	52
53	0.19900	0.4052	0.01603	1533.3	1533.3	21.07	1063.9	1085.0	0.0420	2.0747	2.1167	53
54	0.20642	0.4203	0.01603	1481.0	1481.0	22.07	1063.3	1085.4	0.0439	2.0697	2.1136	54
55	0.2141	0.4359	0.01603	1430.7	1430.7	23.07	1062.7	1085.8	0.0459	2.0645	2.1104	55
56	0.2220	0.4520	0.01603	1382.4	1382.4	24.06	1062.2	1086.3	0.0478	2.0594	2.1072	56
57	0.2302	0.4686	0.01603	1335.9	1335.9	25.06	1061.6	1086.7	2.0497	2.0544	2.1041	57
58	0.2386	0.4858	0.01604	1291.1	1291.1	26.06	1061.0	1087.1	0.0517	2.0493	2.1010	58
59	0.2473	0.5035	0.01604	1248.1	1248.1	27.06	1060.5	1087.6	0.0536	2.0443	2.0979	59
60	0.2563	0.5218	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948	60
61	0.2655	0.5407	0.01604	1166.8	1166.8	29.06	1059.3	1088.4	0.0574	2.0343	2.0917	61
62	0.2751	0.5601	0.01604	1128.4	1128.4	30.05	1058.8	1088.9	0.0593	2.0293	2.0886	62
63	0.2850	0.5802	0.01604	1091.4	1091.4	31.05	1058.2	1089.3	0.0613	2.0243	2.0856	63
64	0.2951	0.6009	0.01605	1055.7	1055.7	32.05	1057.6	1089.7	0.0632	2.0194	2.0826	64
65	0.3056	0.6222	0.01605	1021.4	1021.4	33.05	1057.1	1090.2	0.0651	2.0145	2.0796	65
66	0.3164	0.6442	0.01605	988.4	988.4	34.05	1056.5	1090.6	0.0670	2.0096	2.0766	66
67	0.3276	0.6669	0.01605	956.6	956.6	35.05	1056.0	1091.0	0.0689	2.0047	2.0736	67
68	0.3390	0.6903	0.01605	925.9	925.9	36.04	1055.5	1091.5	0.0708	1.9998	2.0706	68
69	0.3509	0.7144	0.01605	896.3	896.3	37.04	1054.9	1091.9	0.0726	1.9950	2.0676	69
70	0.3631	0.7392	0.01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647	70
71	0.3756	0.7648	0.01606	840.4	840.4	39.04	1053.8	1092.8	0.0764	1.9854	2.0618	71
72	0.3886	0.7912	0.01606	813.9	813.9	40.04	1053.2	1093.2	0.0783	1.9805	2.0588	72
73	0.4019	0.8183	0.01606	788.3	788.4	41.03	1052.6	1093.6	0.0802	1.9757	2.0559	73
74	0.4156	0.8462	0.01606	763.7	763.8	42.03	1052.1	1094.1	0.0820	1.9710	2.0530	74
75	0.4298	0.8750	0.01607	740.0	740.0	43.03	1051.5	1094.5	0.0839	1.9663	2.0502	75
76	0.4443	0.9046	0.01607	717.1	717.1	44.03	1050.9	1094.9	0.0858	1.9615	2.0473	76
77	0.4593	0.9352	0.01607	694.9	694.9	45.02	1050.4	1095.4	0.0876	1.9569	2.0445	77
78	0.4747	0.9666	0.01607	673.6	673.6	46.02	1049.8	1095.8	0.0895	1.9521	2.0416	78
79	0.4906	0.9989	0.01608	653.0	653.0	47.02	1049.2	1096.2	0.0913	1.9475	2.0388	79
80	0.5069	1.0321	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360	80
81	0.5237	1.0664	0.01608	613.9	613.9	49.02	1048.1	1097.1	0.0950	1.9382	2.0332	81
82	0.5410	1.1016	0.01608	595.3	595.3	50.01	1047.5	1097.5	0.0969	1.9335	2.0304	82
83	0.5588	1.1378	0.01609	577.4	577.4	51.01	1046.9	1097.9	0.0987	1.9290	2.0277	83
84	0.5771	1.1750	0.01609	560.1	560.2	52.01	1046.4	1098.4	0.1005	1.9244	2.0249	84
85	0.5959	1.2133	0.01609	543.4	543.5	53.00	1045.8	1098.8	0.1024	1.9198	2.0222	85
86	0.6152	1.2527	0.01609	527.3	527.3	54.00	1045.2	1099.2	0.1042	1.9153	2.0195	86
87	0.6351	1.2931	0.01610	511.7	511.7	55.00	1044.7	1099.7	0.1060	1.9108	2.0168	87
88	0.6556	1.3347	0.01610	496.6	496.7	56.00	1044.1	1100.1	0.1079	1.9062	2.0141	88
89	0.6766	1.3775	0.01610	482.1	482.1	56.99	1043.5	1100.5	0.1097	1.9017	2.0114	89
90	0.6982	1.4215	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087	90
91	0.7204	1.4667	0.01611	454.4	454.4	58.99	1042.4	1101.4	0.1133	1.8927	2.0060	91
92	0.7432	1.5131	0.01611	441.2	441.3	59.99	1041.8	1101.8	0.1151	1.8883	2.0034	92
93	0.7666	1.5608	0.01611	428.5	428.5	60.98	1041.2	1102.2	0.1169	1.8838	2.0007	93
94	0.7906	1.6097	0.01612	416.2	416.2	61.98	1040.7	1102.6	0.1187	1.8794	1.9981	94
95	0.8153	1.6600	0.01612	404.3	404.3	62.98	1040.1	1103.1	0.1205	1.8750	1.9955	95
96	0.8407	1.7117	0.01612	392.8	392.8	63.98	1039.5	1103.5	0.1223	1.8706	1.9929	96
97	0.8668	1.7647	0.01612	381.7	381.7	64.97	1038.9	1103.9	0.1241	1.8662	1.9903	97
98	0.8935	1.8192	0.01613	370.9	370.9	65.97	1038.4	1104.4	0.1259	1.8618	1.9877	98
99	0.9210	1.8751	0.01613	360.4	360.5	66.97	1037.8	1104.8	0.1277	1.8575	1.9852	99
100	0.9492	1.9325	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826	100
101	0.9781	1.9915	0.01614	340.6	340.6	68.96	1036.6	1105.6	0.1313	1.8488	1.9801	101

[a]Reprinted by permission from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, published by John Wiley and Sons, Inc., 1936 edition.

TABLE 9. WEIGHT OF SATURATED AND PARTLY SATURATED AIR[a]

Dry-Bulb Temp Deg F	Weight of Saturated Air for Various Barometric and Hygrometric Conditions—Pounds per Cubic Foot							Approx. Average Increase in Weight per Deg Wet-Bulb Depression
	Barometric Pressure Inches of Mercury						Increase In Weight Per 0.1 in. Rise in Barometer	
	28.5	29.0	29.5	30.0	30.5	31.0		
30	0.07703	0.07839	0.07974	0.08110	0.08245	0.08381	0.00027	0.000017
32	0.07671	0.07806	0.07940	0.08075	0.08210	0.08345	0.00027	0.000017
34	0.07638	0.07772	0.07907	0.08041	0.08175	0.08310	0.00027	0.000018
36	0.07605	0.07739	0.07873	0.08007	0.08141	0.08274	0.00027	0.000018
38	0.07573	0.07706	0.07840	0.07973	0.08106	0.08239	0.00027	0.000019
40	0.07541	0.07674	0.07806	0.07939	0.08072	0.08205	0.00027	0.000019
42	0.07509	0.07641	0.07773	0.07905	0.08038	0.08170	0.00026	0.000020
44	0.07477	0.07609	0.07740	0.07872	0.08004	0.08135	0.00026	0.000020
46	0.07445	0.07576	0.07707	0.07838	0.07970	0.08101	0.00026	0.000021
48	0.07413	0.07544	0.07674	0.07805	0.07936	0.08066	0.00026	0.000021
50	0.07381	0.07512	0.07642	0.07772	0.07902	0.08032	0.00026	0.000022
52	0.07350	0.07479	0.07609	0.07739	0.07868	0.07998	0.00026	0.000023
54	0.07318	0.07447	0.07576	0.07706	0.07835	0.07964	0.00026	0.000023
56	0.07287	0.07415	0.07544	0.07673	0.07801	0.07930	0.00026	0.000024
58	0.07255	0.07383	0.07512	0.07640	0.07768	0.07896	0.00026	0.000025
60	0.07224	0.07352	0.07479	0.07607	0.07734	0.07862	0.00026	0.000026
62	0.07193	0.07320	0.07447	0.07574	0.07701	0.07828	0.00026	0.000027
64	0.07161	0.07288	0.07414	0.07541	0.07668	0.07794	0.00026	0.000028
66	0.07130	0.07256	0.07382	0.07508	0.07634	0.07760	0.00026	0.000029
68	0.07098	0.07224	0.07350	0.07475	0.07601	0.07727	0.00026	0.000030
70	0.07067	0.07192	0.07317	0.07442	0.07568	0.07693	0.00026	0.000031
72	0.07035	0.07160	0.07285	0.07410	0.07534	0.07659	0.00025	0.000032
74	0.07004	0.07128	0.07252	0.07377	0.07501	0.07625	0.00025	0.000033
76	0.06972	0.07096	0.07220	0.07343	0.07467	0.07591	0.00025	0.000034
78	0.06940	0.07064	0.07187	0.07310	0.07434	0.07557	0.00025	0.000036
80	0.06909	0.07032	0.07155	0.07277	0.07400	0.07523	0.00025	0.000037
82	0.06877	0.07000	0.07122	0.07244	0.07366	0.07489	0.00024	0.000039
84	0.06845	0.06967	0.07089	0.07211	0.07333	0.07454	0.00024	0.000040
86	0.06812	0.06934	0.07056	0.07177	0.07299	0.07420	0.00024	0.000042
88	0.06780	0.06901	0.07022	0.07143	0.07264	0.07385	0.00024	0.000043
90	0.06748	0.06868	0.06989	0.07109	0.07230	0.07351	0.00024	0.000045
92	0.06715	0.06835	0.06955	0.07075	0.07195	0.07316	0.00024	0.000047
94	0.06682	0.06801	0.06921	0.07041	0.07161	0.07280	0.00024	0.000049
96	0.06648	0.06768	0.06887	0.07006	0.07126	0.07245	0.00024	0.000051
98	0.06615	0.06734	0.06853	0.06972	0.07091	0.07209	0.00024	0.000053
100	0.06581	0.06700	0.06818	0.06937	0.07055	0.07174	0.00024	0.000055

[a]Approximate average decrease in weight per 0.1 F rise in dry-bulb temperature equals 0.000017 lb per cubic foot.

60 F, the thermodynamic wet-bulb is 50 F, the dew-point is 40.8 F, the degree of saturation is 48.6 per cent, the humidity ratio is 0.00536 lb per pound dry air, and the enthalpy is 14.85 + (1000 × 0.00536) = 20.21 Btu per pound dry air.

With the aid of the Mollier diagram, it is easy to throw the definition of thermodynamic wet-bulb, Equation 20, into a more familiar form. Consider the three points 1, 2, 3, Fig. 2. Point 3 is located with respect to points 1 and 2 so that $W_3 = W_1$ and $t_3 = t_2$. Points 1 and 2, being on a line of constant thermodynamic wet-bulb, satisfy Equation 20; thus,

$$h_1 - h_3 + (W_2 - W_1)\, h'_{w,2} = h_2 - h_3$$

where h_3 has been subtracted from both sides. Under Dalton's Law, $h_2 - h_3 = (W_2 - W_1)\, h_{w,2}$; moreover, $h_1 - h_3$ may be replaced by $\bar{s}_1 (t_1 - t_2)$ where $\bar{s}_1$ is often referred to as *mean humid heat* and may be calculated with good approximation from

$$\bar{s}_1 = 0.240 + 0.444\, W_1 \tag{21}$$

Finally, introducing latent heat of vaporization at the wet-bulb tempera-

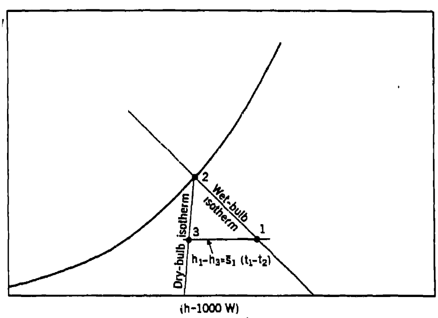


FIG. 2. DIAGRAM ILLUSTRATING THERMODYNAMIC WET-BULB TEMPERATURE

ture, namely, $(h_{fg})_2 = (h_w - h'_w)_2$, Equation 20 becomes, after omitting the subscript 2,

$$\frac{t_1 - t'}{W_s - W_1} = \frac{h_{fg}}{\bar{s}_1} \tag{22}$$

it being understood that W_s is the saturation humidity ratio and h_{fg}, the latent heat, at the wet-bulb temperature t'. Equation 22 was derived by Carrier.

Example 12. Work Example 10 using Equation 22.

Solution. A trial-by-error method is involved. Taking 48 F as a trial value of t', (80 − 48) ÷ (0.007072 − 0) = 4520; but 1066.7 ÷ 0.240 = 4440. The trial value must, therefore, be revised upward, the final solution being 48.26 F as in Example 10.

TYPICAL AIR CONDITIONING PROCESSES

Illustrative Examples. The use of Table 6 and the Mollier diagram in analyzing typical air conditioning processes is best explained by the use of illustrative examples. In each of these examples, the observed pressure is assumed to be standard atmospheric pressure (29.921 in. Hg).

Example 13. Heating. Air at 0 F and 80 per cent saturation is to be heated to 120 F. Analyze the process as illustrated in Fig. 3.

Solution. The initial humidity ratio is 0.80 × 0.0007852 = 0.000628 lb per pound dry air (table). This same value is read directly on the chart. The initial enthalpy is 0.00 + (0.80 × 0.832) = 0.666 Btu per pound dry air (table) or 0.04 + (1000 × 0.00063) = 0.67 (chart).

The final degree of saturation is 0.000628 ÷ 0.08093 = 0.0078 (table); hence the final enthalpy is 28.80 + (0.0078 × 90.09) = 29.50 Btu per pound (table) or 28.87 + (1000 × 0.00063) = 29.50 (chart).

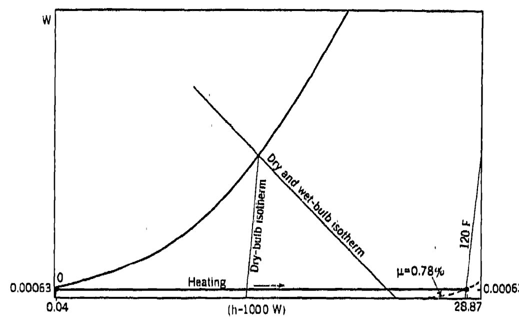


FIG. 3. DIAGRAM ILLUSTRATING EXAMPLE 13

The increase in enthalpy is the quantity of heat to be supplied, namely, 29.50 − 0.666 = 28.83 Btu per pound dry air (table). Since humidity ratio W and therefore $1000W$ is constant, this is also simply the horizontal distance between the representative points on the chart; thus, the heat to be supplied is also 28.87 − 0.04 = 28.83 Btu per pound dry air (chart).

The final volume is 14.60 + (0.0078 × 1.90) = 14.61 cu ft per pound (table). Therefore, if 20,000 cfm of heated air is to be supplied, the quantity of heat required is (20,000 ÷ 14.61) × 28.83 = 39,460 Btu per minute.

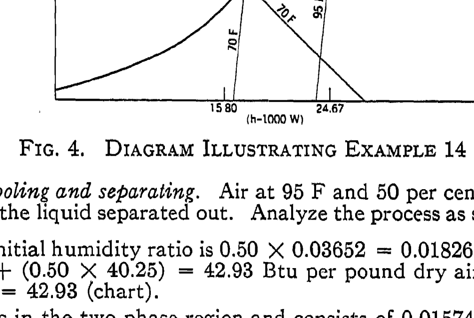


FIG. 4. DIAGRAM ILLUSTRATING EXAMPLE 14

Example 14. Cooling and separating. Air at 95 F and 50 per cent saturation is to be cooled to 70 F and the liquid separated out. Analyze the process as shown in Fig. 4.

Solution. The initial humidity ratio is 0.50 × 0.03652 = 0.01826 (table). The initial enthalpy is 22.80 + (0.50 × 40.25) = 42.93 Btu per pound dry air (table) or 24.67 + (1000 × 0.01826) = 42.93 (chart).

The final state is in the two-phase region and consists of 0.01574 lb water per pound dry air in the vapor phase, and 0.00252 lb water per pound dry air in the liquid phase.

The final enthalpy is therefor $33.96 + (0.00252 \times 38.04) = 34.06$ Btu per pound dry air (table) or $15.80 + (1000 \times 0.01826) = 34.06$ (chart).

The decrease of enthalpy is the refrigeration to be supplied and is $42.93 - 34.06 = 8.87$ Btu per pound dry air (table). Since the weight of water per pound of dry air is constant, this is also the horizontal distance between the representative points on the chart, namely, $24.67 - 15.80 = 8.87$ Btu per pound dry air (chart).

The initial volume is $13.97 + (0.50 \times 0.82) = 14.38$ cu ft per pound (table). Therefore, if 20,000 cfm of initial air is to be processed, the refrigeration required is $(20{,}000 \times 8.87) \div (14.38 \times 200) = 61.7$ tons. The weight of water to be removed is $(20{,}000 \times 0.00252) \div 14.38 = 3.51$ lb per minute.

Example 15. Adiabatic Saturation with Recirculated Spray Water. Air at 75 F and 60 per cent saturation is saturated adiabatically with spray water which is recirculated. Find the resulting temperature and the weight of water added per pound of dry air as outlined in Fig. 5.

Solution. The recirculated water will assume the thermodynamic wet-bulb temperature of the entering air which will also be the temperature of the resulting saturated mixture. The humidity ratio of the entering air is $0.60 \times 0.01873 = 0.01124$ lb water per pound dry air (table); its enthalpy is $17.99 + (0.60 \times 20.47) = 30.27$ Btu per pound

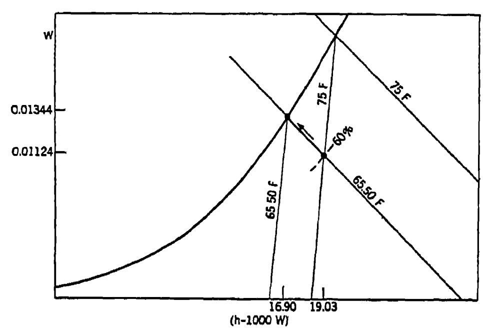


FIG. 5. DIAGRAM ILLUSTRATING EXAMPLE 15

dry air (table) or $19.03 + (1000 \times 0.01124) = 30.27$ Btu per pound dry air (chart). To determine the resulting temperature, the following equation must be solved.

$$30.27 + (W_s - 0.01124)\, h_w' = h_s$$

A trial value is 65 F corresponding to $h_s = 30.27$. The final value is 65.50 F corresponding to $h_s = 30.27 + (0.01320 - 0.01124) \times 33.05 = 30.34$ Btu per pound dry air. The weight of water to be added is $0.01344 - 0.01124 = 0.00200$ lb per pound dry air.

The volume of the entering air is $13.47 + (0.60 \times 0.40) = 13.71$ cu ft per pound. If 20,000 cfm of entering air is to be saturated, the weight of water to be added per minute is $(20{,}000 \times 0.00200) \div 13.71 = 2.92$ lb per minute.

Adiabatic Mixing of Two Air Streams

A typical process requiring special discussion is the adiabatic mixing of two air streams. Let stream 1 contain M_1 pounds of dry air per minute and let its enthalpy be h_1 and its humidity ratio W_1. Using subscripts 2 and 3 in a similar manner to designate stream 2 and the resulting mixture respectively, write:

$$M_1 + M_2 = M_3 \qquad \text{(weight balance for the dry air)}$$
$$M_1W_1 + M_2W_2 = M_3W_3 \qquad \text{(weight balance for the water)}$$
$$M_1h_1 + M_2h_2 = M_3h_3 \qquad \text{(energy balance, no heat absorbed)}$$

Eliminating M_3,

$$\frac{W_2 - W_3}{W_3 - W_1} = \frac{h_2 - h_3}{h_3 - h_1} = \frac{M_1}{M_2} \tag{23}$$

according to which: *on the Mollier Chart the representative point of the resulting mixture lies on the straight line connecting the representative points of the two streams being mixed, and divides the line into two segments which are in the same ratio as the weights of dry air in the two streams.* It must not be forgotten that this analysis assumes *adiabatic* mixing.

Example 16. Outside air at 0 F and 80 per cent saturation is to be mixed adiabatically with recirculated air at 70 F and 20 per cent saturation in the ratio, one pound of dry air in the former to seven in the latter. Find the temperature and degree of saturation of the resulting mixture as shown in Fig. 6.

Solution. The humidity ratio and enthalpy of the resulting mixture satisfy

$$\frac{0.003148 - W_3}{W_3 - 0.000628} = \frac{20.23 - h_3}{h_3 - 0.666} = \frac{1}{7}$$

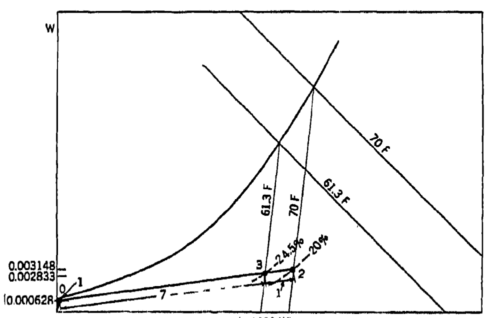


FIG. 6. DIAGRAM ILLUSTRATING EXAMPLE 16

whence,

$W_3 = 0.002833$ lb water per pound dry air.

$h_3 = 17.78$ Btu per pound dry air.

The corresponding temperature and degree of saturation are 61.3 F and 24.5 per cent as is easily verified by use of Table 6. The numerical solution is somewhat tedious, but the graphical solution is easy.

Adiabatic Mixing with Injected Water

Another typical process is that of injecting water (solid, liquid or vapor) into an air stream to mix adiabatically with it. Let the subscripts 1 and 2 refer to the initial and final conditions, respectively; then write

$$\begin{array}{lll} 1 + 0 & = 1 & \text{(weight balance for the dry air)} \\ W_1 + (W_2 - W_1) & = W_2 & \text{(weight balance for the water)} \\ h_1 + (W_2 - W_1)\, h_w & = h_2 & \text{(energy balance, no heat absorbed)} \end{array}$$

The first two are identities and are incorporated in the third. This may be rewritten as follows:

$$\frac{h_2 - h_1}{W_2 - W_1} = h_w \tag{24}$$

and shows that *the process is represented by a straight line on the Mollier*

diagram, the slope of the line being determined by the specific enthalpy of the injected water. It must not be forgotten that the analysis assumes adiabatic mixing. *Energy convected with a fluid is not heat.*

Example 17. It is desired to increase the humidity ratio of air at 70 F without changing its temperature. Under what conditions may water be injected in order to accomplish the desired result.

Solution. Under Dalton's Law a line of constant (dry-bulb) temperature is straight on the Mollier diagram and its slope is determined by the specific enthalpy of water *vapor* at the given temperature. At 70 F, $h_W = 1092.3$ Btu per pound; hence injection of steam having this specific enthalpy will cause the representative point to move in a direction parallel to the 70 F isotherm. Saturated steam at 70 F may not be used because its pressure is only 0.7392 in. Hg and it cannot therefore be injected into air at atmospheric pressure. Saturated steam at 667.4 F, 2488 lb per sq inch has the right specific enthalpy and can be throttled into a room at 70 F without altering the room temperature.

Border Scale

On the Mollier diagram is placed a border scale to facilitate the graphical solution of problems in which given quantities of energy and water are added (or withdrawn) simultaneously as in the case of adiabatic mixing

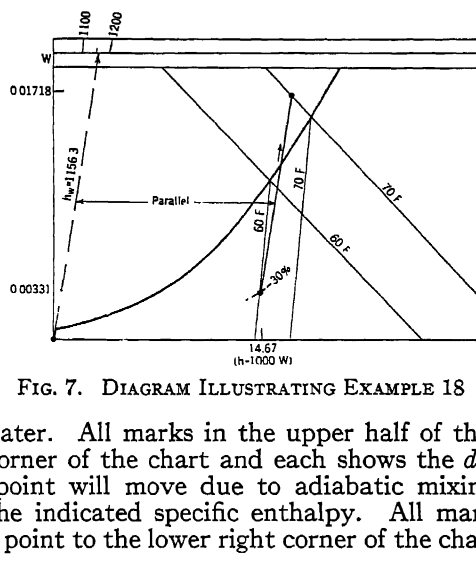


FIG. 7. DIAGRAM ILLUSTRATING EXAMPLE 18

with injected water. All marks in the upper half of this scale point to the lower left corner of the chart and each shows the *direction* that the representative point will move due to adiabatic mixing with injected water having the indicated specific enthalpy. All marks in the lower half of the scale point to the lower right corner of the chart.

Example 18. If dry saturated steam at 20 lb per square inch absolute is injected into air initially at 60 F and 30 per cent relative humidity to raise the temperature to 70 F, what is the final relative humidity and how much water is added per pound dry air?

Solution. The initial humidity ratio is $0.30 \times 0.01103 = 0.00331$ lb water per pound dry air (or direct from chart). The initial enthalpy is $14.39 + (0.30 \times 11.98) = 17.98$ Btu per pound dry air (table) or $14.67 + (1000 \times 0.00331) = 17.98$ (chart). A preliminary calcuation shows that the final mixture contains liquid. The final weight of water per pound of dry air is determined from

$$\frac{33.96 + (W - 0.01574) \times 38.0 - 17.98}{W - 0.00331} = 1156.3$$

where the specific enthalpy of the injected water is 1156.3 Btu per pound. The answer is

$$W = 0.01718 \text{ lb water per pound dry air.}$$

Therefore, the weight of water added is $0.01718 - 0.00331 = 0.01387$ lb per pound dry air as shown in Fig. 7.

Cooling Load

In the calculation of the cooling load for an air conditioned space, the problem usually reduces to determining the quantity of inside air that must be withdrawn and the condition to which it must be brought by cooling, separating and possibly reheating so that return of the conditioned air will have the net effect of removing given amounts of energy and water from the air conditioned space.

Let m denote the weight of dry air withdrawn per hour. With it will be withdrawn *energy* of amount mh_1 Btu per hour and *water* of amount mW_1 pounds per hour, where h_1 and W_1 denote enthalpy and humidity ratio, respectively, of inside air. The weight of dry air returned per hour will be the same as that withdrawn but with it must be returned a smaller amount of *energy*, mh Btu per hour, and a smaller quantity of *water*, mW pounds per hour, where h and W denote enthalpy and humidity ratio of conditioned air.

With this understanding, the requirements of the cooling load problem are,

$$mh = mh_1 - \Delta Q$$
$$mW = mW - \Delta W$$

where ΔQ and ΔW are the *given* amounts of energy and water, respectively, to be removed. Eliminating m from these two equations,

$$\frac{h - h_1}{W - W_1} = \frac{\Delta Q}{\Delta W}$$

which says that all possible states for the conditioned air lie on a straight line, on the Mollier Chart, which passes through the state point of the inside air with a slope determined by the ratio of the quantity of energy to be removed to the quantity of water to be removed. This straight line is called the *condition line* for the given problem. The border scale facilitates the graphical solution of this problem.

In practice the point at which the condition line crosses the saturation curve may dictate an excessive number of air changes for the particular space to be conditioned. If so it might be necessary to cool to a lower temperature; but if the requirements of the problem are to be exactly met both as regards the removal of water and the removal of energy, the mixture returned to the conditioned space must contain a certain amount of liquid. In other words, its state point must lie on the condition line; otherwise excessive dehumidification will result.

Example 19. In order to maintain a condition of 80 F dry-bulb, 67 F wet-bulb in a certain store, it is found necessary to remove 115,060 Btu of energy per hour and 15.97 lb of water per hour. Analyze the problem.

Solution. The state point of the inside air is easily located on the Mollier Chart. Through it draw a line having the slope 115,060 ÷ 15.97 = 7205 Btu per pound *water* as determined from the border scale. This line crosses the saturation curve at 58.02 F. Hence a possible conditioning process is to cool some of the inside air to 58.02 F, separate the liquid thus formed, and return the resulting saturated mixture to the store.

The thermodynamic properties entering the calculation are:

	Inside Air	*After Cooling*	*After Separating*
t	80.0 F	58.02 F	58.02 F
W	0.01115	0.01115	0.01027
h	31.41	25.09	25.07

The weight of dry air to be withdrawn is 115,060 ÷ (31.41 − 25.07) = 18,130 lb per hour.

Adiabatic Saturation

Any case of adiabatic mixing in which the resulting mixture is saturated may properly be called *adiabatic saturation.* For example, if enough water at 352 F be sprayed into dry air at 80 F to produce a saturated mixture, the resulting enthalpy will be $h_s = 19.19 + (W_s - 0)\ 324$; and since h_s and W_s are functions of the same temperature, this temperature is determined by the equation to be 53.0 F. Thus, adiabatic saturation of dry air at 80 F by injecting liquid water at 352 F results in a temperature of 53.0 F when saturation is reached.

But in practice, much more is usually read into the term adiabatic saturation, it being generally understood that saturation is to be produced by injecting liquid water at such a temperature as will *coincide* with that at which the saturation curve is reached. *With this understanding it may be said that thermodynamic wet-bulb temperature is the*

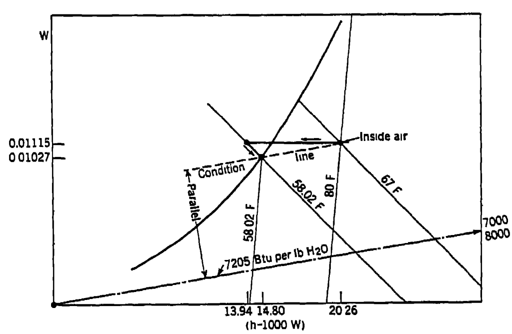


FIG. 8. DIAGRAM ILLUSTRATING EXAMPLE 19

result of adiabatic saturation. Thus, if liquid water at 48.26 F instead of 352 F be injected into dry air at 80 F a saturated mixture at 48.26 F instead of 53.0 F will be produced. Therefore, 48.26 F is the thermodynamic wet-bulb temperature of dry air at 80 F.

It is possible to produce adiabatic saturation, interpreting the term literally, by mixing two air streams neither of which is itself saturated. In order for this to be possible, the straight line connecting the representative points on the Mollier diagram must cut the saturation curve twice.

PSYCHROMETRIC CHART

Many types of charts which give graphical solutions of the psychrometric equations, and other useful data, have been devised. One of these, the Revised Bulkeley Psychrometric Chart[4], will be found in the envelope attached to the inside back cover of this book. Detailed in-

[4]The original Bulkeley Psychrometric Chart was presented to the Society in 1926. (See A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 163). Single copy of the revised chart can be furnished at a cost of $ .25.

structions in the use of this chart will be found thereon. The lines of constant relative humidity appearing on this chart have been drawn in accordance with the definition as given by Equations 11 and 15.

STEADY FLOW ENERGY EQUATION

It was previously stated that, in steady flow, the energy convected by the fluid at any section is the sum of (*a*) kinetic energy due to velocity; (*b*) gravitational energy due to elevation; (*c*) enthalpy due to the condition of pressure, temperature and composition of the fluid. A more detailed discussion of item (*a*) is in order.

Kinetic Energy

There are reasons to believe that the so-called *velocity pressure* h_V read by a Pitot tube is simply the kinetic energy per unit volume of the fluid immediately upstream from the tube, as application of Bernoulli's Equation suggests. Thus (see Equation 3, Chapter 34).

$$V = 1096.2 \sqrt{\frac{h_V}{d}} \tag{25}$$

where

V = velocity, feet per minute.
h_V = velocity pressure, inches of water.
d = density of fluid, pounds per cubic foot.

In the case of flow through a duct, the velocity pressure is found to vary considerably over the section and a traverse has to be made. The cross-sectional area of the duct is divided into a number of equal concentric areas, and measuring stations are located at centroidal points in each area along two perpendicular diameters. Usually the ultimate object is to determine an *average* velocity $\overline{V}$ from which the weight of fluid crossing the section per unit time can be obtained on multiplying by the cross-sectional area of the duct and by the density of the fluid. This is obtained by simply averaging the square roots of all measured velocity pressures as follows:

$$\overline{V} = \frac{1096.2}{\sqrt{d}} \left(h_V^{1/2} \right)_{av} \tag{26}$$

where

$\overline{V}$ = average velocity, feet per minute.
$(h_V^{1/2})_{av}$ = arithmetic average of the square roots of all measured velocity pressures, inches of water.

But the item of present importance is the *average kinetic energy* convected with each pound of fluid. Consistently with the previous discussion, this can be shown to be

$$\overline{KE} = 0.006678\, v \frac{\left(h_V^{3/2} \right)_{av}}{\left(h_V^{1/2} \right)_{av}} \tag{27}$$

where

$\overline{KE}$ = average kinetic energy, Btu per pound.
v = specific volume, cubic feet per pound.
$(h_V^{3/2})_{av}$ = arithmetic average of the 3/2-powers of all measured velocity pressures, inches of water.

If the velocity pressure were uniform over the section, Equations 26 and 27 could be combined to give

$$\overline{KE} = \left(\frac{\overline{V}}{13{,}430}\right)^2 \tag{28}$$

But, it is interesting to note that if the velocity varies *parabolically* from zero at the walls to maximum at the center as it does in the case of purely viscous flow in a circular duct, *then the average kinetic energy is twice that given by* Equation 28.

Example 20. If 2000 cfm of air flows through an 8 in. diameter circular duct, find the average kinetic energy per pound of air.

Solution. The cross-sectional area of the duct is 0.349 sq ft; hence the average flow velocity is 5730 fpm. If the velocity were uniform over the section, the average kinetic energy would be $(5730 \div 13{,}430)^2 = 0.182$ Btu per pound. But it is more likely that the actual distribution of velocity would approximate that characteristic of viscous flow; hence the average kinetic energy would be more nearly $2 \times 0.182 = 0.364$ Btu per pound.

Gravitational Energy

The potential energy due to elevation Z (feet) above any convenient datum is simply $Z \div 778.3$ Btu per pound of fluid. In the case of moist air,

$$\overline{PE} = \frac{\overline{Z}\,(1 + W)}{778.3} \tag{29}$$

where

$\overline{PE}$ = average potential energy, Btu per pound dry air.

$\overline{Z}$ = average elevation, feet.

W = humidity ratio, pound water per pound dry air.

Enthalpy

No further discussion of enthalpy is required. It may be well to emphasize, however, that enthalpies have been figured on the basis of one pound of dry air.

Heat and Shaft Work

Between any two sections 1 and 2 in an apparatus through which steady flow occurs, there may be heat *absorbed* from outside, ${}_1q_2$, Btu per pound of dry air, and shaft work *removed* to outside, ${}_1l_2$, Btu per pound of dry air. If heat is actually rejected to outside, ${}_1q_2$ is intrinsically negative; and if shaft work is actually put in from outside ${}_1l_2$, is intrinsically negative.

Steady-flow Energy Equation

A complete energy accounting takes the form of Equation 30 which is usually referred to as the steady-flow energy equation.

$${}_1q_2 = (h_2 + \overline{KE}_2 + \overline{PE}_2) - (h_1 + \overline{KE}_1 + \overline{PE}_1) + {}_1l_2 \tag{30}$$

where

${}_1q_2$ = heat *added* from outside *between* sections 1 and 2, Btu per pound dry air.

h_2 = enthalpy of the mixture *at* section 2, Btu per pound dry air.

$\overline{KE}_2$ = average kinetic energy *at* section 2, Btu per pound dry air.
$\overline{PE}_2$ = average potential energy *at* section 2, Btu per pound dry air.
h_1 = enthalpy *at* section 1, Btu per pound dry air.
$\overline{KE}_1$ = average kinetic energy *at* section 1, Btu per pound dry air.
$\overline{PE}_1$ = average potential energy *at* section 1, Btu per pound dry air.
${}_1l_2$ = shaft work withdrawn *between* sections 1 and 2, Btu per pound dry air.

In Equation 30 all quantities are *per pound of dry air*. If Equation 27 is used in computing average kinetic energy, the result will be in Btu per pound of dry air if v is taken as volume per pound of dry air. If Equation 28 is used, multiplication by $(1 + W)$ as in Equation 29 is required though this is a refinement seldom justified.

Properties of saturated steam are given in Table 8 and for additional definitions refer to Chapter 46.

REFERENCES

Rational Psychrometric Formulae, by W. H. Carrier (*A.S.M.E. Transactions*, Vol. 33 1911, p. 1005).

Heat and Thermodynamics, by J. K. Roberts (Chapter IV), (Blackie and Son, London, 1928).

Das i, x-Diagramm für Dampfluftgemische, by R. Mollier (ZVDI, 1929).

The Theory of the Psychrometer, by J. H. Arnold (*Physics*, Vol. 4, 1933).

Thermodynamic Properties of Moist Air, by John A. Goff (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 199).

Thermodynamic Properties of Steam, by J. H. Keenan and F. G. Keyes (John Wiley and Sons, Inc., New York, 1936).

Notes on Thermodynamics, by John A. Goff (John S. Swift Company, St. Louis, 1939).

Thermal Properties of Saturated Water and Steam, by N. S. Osborne, H. F. Stimson and D. C. Ginnings (*National Bureau of Standards*, Research Paper No. RP 1229, 1939).

Psychrometric Chart, Its Application and Theory, by William Goodman (*Heating, Piping and Air Conditioning*, June, 1939 to August, 1940).

Chapter 2

PHYSIOLOGICAL PRINCIPLES

Chemical Vitiation of Air, Physical Impurities in Air, Thermal Changes Between the Body and Its Environment, Adaptation to Hot Conditions, Adaptation to Cold Conditions, Relation of Air Conditioning Needs to Metabolism, Acclimatization, Effective Temperature Index, Physiological Objectives of Heating and Ventilation, Relation of Air and Wall Temperatures, Influence of Humidity, Influence of Air Movement, The Four Vital Factors

VENTILATION is defined in part as *the process of supplying or removing air by natural or mechanical means to or from any space.* (see Chapter 46). The word in itself implies quantity but not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying a given quantity.

The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. More particularly, it is often used to include the simultaneous control of temperature, humidity, movement and quality of air. The term is broad enough to embrace whatever factors may be found desirable, in a given case, for maintaining the atmosphere of occupied spaces at a condition best suited to the physiological requirements of the human body.

CHEMICAL VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain physical and chemical changes which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs.

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are too small to produce appreciable effects even under the worst conditions of normal human occupancy. The amount of carbon dioxide in air is often used in ventilation work as an index of odors of human origin, but the information it affords rarely justifies the labor involved in making the observation[1,2].

---

[1]A.S.H.V.E. RESEARCH REPORT No. 959—Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

[2]A.S.H.V.E. RESEARCH REPORT No. 1031—Ventilation Requirements, by C. P. Yaglou, E. C. Riley and D. J. Coggins (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 133).

Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a substance has not been demonstrated. The only certain fact is that expired and transpired air may be odorous and offensive, and it is capable of producing loss of appetite and a disinclination for physical activity. These reasons, whether esthetic or physiological, call for the introduction of a certain minimum amount of clean outdoor air to dilute the odoriferous matter to a concentration which is not objectionable.

In certain industrial processes toxic fumes and gases may be produced, whose removal by local exhaust ventilation is essential for the protection of human health; and in the ordinary occupied space chemical impurities may be contributed by the fumes from certain types of cooking and heating appliances. Odors of cooking should not be minimized since it has been shown that odors have an important indirect effect on health in diminishing the appetite for food; and carbon monoxide from imperfect combustion may be a serious hazard to life and health.

When the only source of contamination is the human occupant, the minimum quantity of outdoor air needed appears to be that necessary to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends upon a number of factors, including socio-economic status of occupants, outdoor air supply, air space allowed per person, odor adsorbing capacity of air conditioning processes, temperature, and other factors of secondary importance. With any given group of occupants and type of air conditioner the intensity of body odor perceived upon entering a room from relatively clean air has been found to vary inversely with the logarithm of outdoor air supply and the logarithm of the air space allowed per person.

The minimum outdoor air supply necessary to remove objectionable body odors under various conditions, as determined experimentally at the *Harvard School of Public Health*[3], is given in Table 1. Outdoor air requirements for the removal of objectionable tobacco smoke odors have yet to be determined. Practical values in the field vary from 10 to 15 cfm per person; this air quantity may and should be a part of that necessary for other requirements, *e.g.*, removal of body odors, heat, moisture, etc.

The total quantity of air to be circulated through an enclosure is governed largely by the needs for controlling temperature and air distribution when either heating or cooling is required. The factors which determine total air quantity include the type and nature of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and last but not least, the method of distribution.

It will be noted that, with adequate air space, the rate of air change indicated in Table 1 is from 10 to 30 cfm per person. In rooms occupied by only a few persons such a rate of air change will be automatically attained in cold weather by normal leakage around doors and windows while it can easily be secured in warm weather by the opening of windows. With a space allotment of 400 cu ft per person, only 1½ air changes per

[3]Loc. Cit. Note 2.

hour are necessary to provide an air change of 10 cfm per person. This space allotment is essential for other reasons. It is indicated by the space requirements of ordinary furniture with room to move about between, and it is about the space needed in a two-bed sleeping room to avoid mouth spray infection.

Therefore, in the ordinary dwelling with adequate cubic space allotment, no special provision for controlling chemical purity of the air is necessary (aside from removal of fumes from heating appliances). With such conditions, the control of heat loss from the body is the major

TABLE 1. MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS

(Provisional values subject to revision upon completion of work)

TYPE OF OCCUPANTS	AIR SPACE PER PERSON CU FT	OUTDOOR AIR SUPPLY CFM PER PERSON
Heating season with or without recirculation. Air not conditioned.		
Sedentary adults of average socio-economic status	100	25
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary Adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person.		
Sedentary Adults	200	<4

factor to be considered. The air breathed in the dwelling rarely contains chemical impurities; but the air which bathes the skin may often be too hot or too cold.

In more crowded rooms (large offices, large workrooms, auditoriums), the whole picture changes. Cubic space per person is less and the size of the room makes it impossible to admit untempered outside air without drafts. Here, mechanical ventilation is essential, but as will be noted in a later paragraph, it is even more essential for thermal than for chemical reasons. It is the removal of the heat produced by human bodies, rather than dilution of chemical poisons, which must govern practice.

In spite of the rapid advances in the field of air conditioning during the past few years, the secret of reproducing indoor atmospheres of as stimulating qualities as those existing outdoors under ideal weather conditions, has not as yet been found. Extensive studies have failed to elucidate the cause of the stimulating qualities of country air, qualities which are lost when such air is brought indoors and particularly when it is handled by mechanical means. Ultra-violet light and ionization have been suggested but the evidence so far is inconclusive or negative[4].

Ozone has been used with success for the destruction of micro-organisms (molds) in meat packing establishments and the like; and where considerable amounts of organic effluvia are present this substance may be useful as a deodorant. For ordinary ventilation practice, however, neither of these purposes can be usefully attained, since the concentration of ozone necessary for effectiveness would be likely to transcend the limit of comfort in ordinary occupied rooms. While ozone has been used in the treatment of certain diseases, there is no evidence that it has a tendency to increase comfort or to benefit health under conditions of normal human occupancy. The allowable concentrations in the breathing zone are very small, between 0.01 to 0.05 ppm parts of air. These are much too small to influence bacteria. Higher concentrations are associated with a pungent unpleasant odor and considerable discomfort to the occupants. One part per million causes respiratory discomfort in man, headaches and depression, lowers the metabolism, and may even lead to coma[5].

PHYSICAL IMPURITIES IN AIR

Dust particles of various types, when present in considerable concentrations, produce an irritant effect upon the mucous membranes of nose and throat and may be associated with high prevalence of acute respiratory diseases such as bronchitis and pneumonia. Dust which contains free silica has special harmful effects, causing a primary disease of the lungs (silicosis) and predisposing the victim in a high degree to tuberculosis. These, however, are special problems of industrial hygiene which will not be discussed in detail in this chapter.

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic organisms from infected persons. Droplets sprayed into the air in talking, coughing, sneezing, etc., do not all fall immediately to the ground within a few feet from the source, as was formerly believed. The large droplets do, of course, but minute droplets less than 0.1 mm in diameter evaporate to dryness before they fall the height of a man. Nuclear residues from such sources, which may contain

[4]A.S.H.V.E. Research Report No. 921—Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 191). A.S.H.V.E. Research Report No. 965—Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt and L. C. Benjamin (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 357). A.S.H.V.E. Research Report No. 985—Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 271). The Nature of Ions in Air and Their Possible Physiological Effects, by L. B. Loeb (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 101). The Influence of Ionized Air upon Normal Subjects, by L. P. Herrington (*Journal Clinical Investigation*, 14, January, 1935). The Effect of High Concentrations of Light Negative Atmospheric Ions on the Growth and Activity of the Albino Rat, by L. P. Herrington and Karl L. Smith (*Journal Ind. Hygiene*, 17, November, 1935). Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions, by C.-E. A. Winslow and L. P. Herrington (A.S.H.V.E. Transactions, Vol. 42, 1936, p. 119).

[5]*The British Medical Journal*, Editorial, June 25, 1932, p. 1182. See also Loc. Cit. Note 4.

infective organisms drift long distances with the air currents and the virus may remain alive long enough to be transmitted to other persons in the same room or building. Droplet nuclei have been recovered from cultures of resistant micro-organisms a week after innoculation into a tight chamber of 3000 cu ft capacity, although the majority of disease germs died out within a few hours[6]. Practical epidemiological evidence indicates that the danger of such atmospheric transmission is slight with the bacterial diseases but may be appreciable with the diseases caused by the much smaller and lighter viruses. Avoidance of overcrowding is a major factor in avoiding such dangers. The microbic content of the atmosphere may also be reduced by air change but local drafts carrying minute droplets from person to person will increase the hazard. Practical possibilities in sterilizing air supplies by the use of ultra-violet light are now being studied[7].

THERMAL INTERCHANGES BETWEEN THE BODY AND ITS ENVIRONMENT

The importance of the thermal factors arises from the profound influence which they exert upon body temperature, comfort and health. Body temperature depends upon the balance between heat production and heat loss. The heat resulting from the combustion of food within the body (metabolism) maintains the body temperature well above that of the surrounding air. At the same time, heat is constantly lost from the body by radiation, convection and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss.

In conditioning air for comfort and health it is necessary to know the rate of sensible and latent heat liberation from the human body, which in conjunction with other heat loads (see Chapters 3, 5 and 6) determine the capacity required for proper conditioning. The data in common use are those of the A.S.H.V.E. Research Laboratory[8].

The fundamental thermodynamic processes concerned in heat interchanges between the body and its environment may be described by the equation:

$$M \pm S = E \pm R \pm C \tag{1}$$

where

M = rate of metabolism.
S = rate of storage.
E = rate of evaporative heat loss.
R = rate of radiative heat loss or gain.
C = rate of convective heat loss or gain.

Units may be expressed in kilogram-calories or Btu per hour, and storage is considered positive when the body cools; negative, when the

[6]Air-Borne Infection and Sanitary Air Control, by W. F. Wells (*Journal Industrial Hygiene*, November, 1935).

[7]Sanitary Ventilation in Wards, by W. F. Wells (*Heating and Ventilating*, April, 1939, p. 26). Measurement of Sanitary Ventilation, by W. F. Wells (*American Journal of Public Health*, Vol. 28, 1938, p. 343).

[8]Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (*American Journal of Hygiene*, Vol. XIII No. 2, March, 1931, p. 415).

body becomes warmer. M is always positive, and E is always negative. R and C are positive when the surface of the body is above that of walls and air, respectively, and negative when the surface of the body is cooler than walls or air.

The human body possesses remarkable powers of adaptation to a range of atmospheric conditions around an ideal optimum where storage is zero, and metabolism and skin and tissue temperature are at optimum values. As skin temperature and body-tissue temperature rise or fall above or below an optimum, complex adaptive mechanisms come into play, chiefly associated with redistribution of blood supply between the skin and deeper tissues (in a cold environment) and with sweat secretion (in a hot environment). Under cold conditions, shivering or other muscular movements increase metabolism, which is, again, a reaction favorable to temperature regulation; but under very hot conditions metabolism also rises and this reaction is obviously harmful and indicates failure of the entire regulative process. These reactions are governed by nervous or chemical stimuli from both skin and internal tissues. Nerves from the skin, for example, carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The mechanisms of adjustment are complex and the reactions involved in a cold and in a hot environment are radically different in nature. Therefore, any attempt to formulate simple engineering relationships covering the entire thermal scale are obviously doomed to failure.

In a certain middle range, normal and easy physiological regulation occurs by slight changes in the distribution of blood between the skin and the inner organs; here, heat loss and heat gain balance and a sensation of comfort is experienced. Above this range, the blood capillaries in the skin become dilated, allowing more blood to flow into the skin, and thus increase its temperature and consequently its heat loss. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin, where it is evaporated. This method of cooling is the most effective of all, as long as the humidity of the air is sufficiently low to allow for evaporation. In high humidities, where the difference between the dew-point temperature of the air and body temperature is not sufficient to allow rapid evaporation, equally good results may be obtained by increasing air movement. The body, under hot conditions, is in the *zone of evaporative regulation*, and for moderately extreme conditions perfect balance between heat production and heat loss may be attained, although at the cost of considerable discomfort.

In a cold environment, where environmental conditions are such as to remove heat too rapidly, the organism adapts in some degree by constricting the blood vessels of the skin, increasing the insulation of the body. The lowered surface temperature of the skin decreases heat loss which obviously depends on the differential between the temperature of

the skin and of the environment. This adaptation is, however, partial and incomplete, and the temperature of the body tissues falls, with accompanying discomfort and ultimate danger of serious chill. The process may go on for many hours. This is known as the *zone of body cooling*. The normal tendency of the individual is to move about and increase metabolism through muscular activity and thus balance the excessive heat demand of the environment.

These phenomena are important and a graph indicating the reactions of lightly-clothed human subjects in a semi-reclining position is shown in

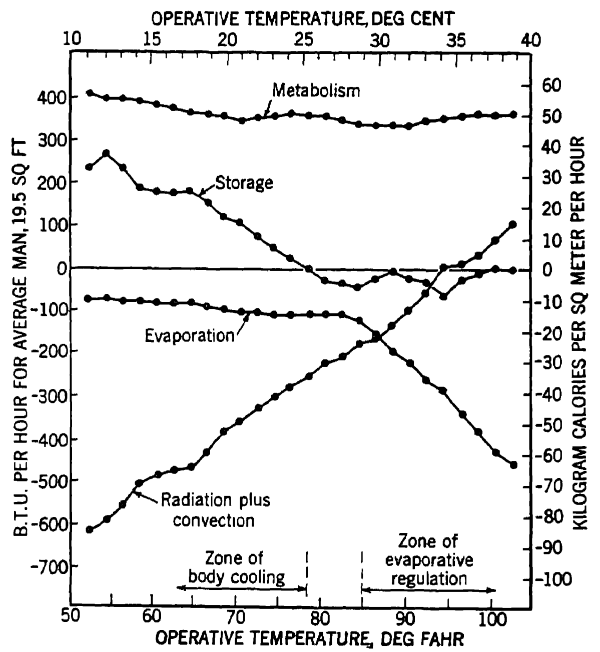


FIG. 1. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND OPERATIVE TEMPERATURE FOR THE CLOTHED SUBJECT

Fig. 1. The air movement was minimal and relative humidity between 40 and 50 per cent. The abscissae are *operative temperatures* which represent the combined effect of air and wall temperatures[9], the ordinates, heat loss per unit of body surface. The following phenomena are obvious:

1. Metabolism (for a given subject) remains approximately constant within the range of operative temperatures employed, and rises slightly below 70 F.

2. At a critical temperature of 85 F the heat produced by metabolism (roughly 50 kilogram-calories per square meter per hour) is balanced by the heat loss due to evaporation and to radiation plus convection; these two major components accounting for about 25 kilogram-calories each. Storage is zero; that is, the body tissues show no change in temperature.

---

[9]A.S.H.V.E. RESEARCH REPORT No. 1107—Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice, by C.-E. A. Winslow, T. Bedford, E. F. DuBois, R. W. Keeton, A. Missenard, R. R. Sayers, and C. Tasker (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 111).

3. As one proceeds to higher operative temperatures, the heat received from the environment due to radiation plus convection increases progressively but is exactly balanced by a similarly progressive increase in evaporative heat loss, so that no further negative storage (warming of the body) takes place. At an operative temperature of 100 F the body is gaining 10 kilogram-calories from the combined influence of walls and air, and this heat gain plus the metabolic heat produced is balanced by a heat loss of 60 kilogram-calories due to evaporation.

4. Below the critical temperature of 85 F the phenomena are wholly different. Evaporative heat loss here changes but slightly, falling from 20 to 10 kilogram-calories per hour as the operative temperature decreases, as a result chiefly of the purely physical factor of decreasing vapor pressure difference between skin and air. In this zone, heat loss due to the combined influence of radiation plus convection increases progressively (although the slope of the line is less rapid than above 85 F). Since this progressively increasing heat loss is not balanced, there is a parallel increase in storage (cooling of the body tissues).

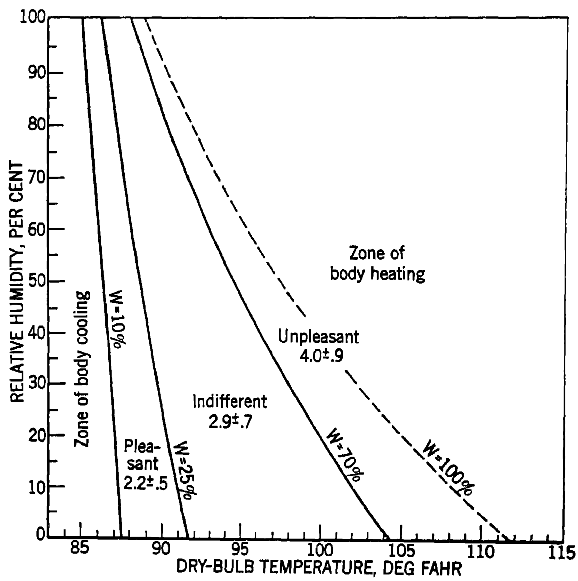


FIG. 2. CONTOUR CHART INDICATING LIMITING WETTED AREAS ASSOCIATED WITH CERTAIN SENSATIONS OF PLEASANTNESS (IN THE ZONE OF EVAPORATIVE REGULATION) IN RELATION TO AIR TEMPERATURE AND RELATIVE HUMIDITY. OBSERVED MEAN COMFORT VOTES ARE INDICATED IN EACH REGION. (UNCLOTHED SUBJECTS.)

Localized drafts are important since differential cooling of one area of the body may produce surprisingly unpleasant reactions in quite different parts of the body. In a recent experiment[10] it was shown that the application of an ice pack to an area of 60 sq cm on the back of the neck for 15 min caused a drop of 17 F in the skin temperature of the fingers and that this low temperature of the fingers persisted for one hour after the ice pack was removed.

[10]The Relative Influence of Radiation and Convection Upon the Temperature Regulation of the Clothed Body, by C.-E. A. Winslow, L. P. Herrington, and A. P. Gagge (*American Journal of Physiology*, Vol. 124, October, 1938, p. 51).

ADAPTATION TO HOT CONDITIONS

Within a certain thermal region on the hot side of the optimum, the body can balance increases in air or wall temperature by increasing secretion of sweat. The body can balance changes in relative humidity within this zone just as precisely as it can balance changes in temperatures, up to the point where the sweat no longer evaporates but runs off without exerting its cooling power. Until the latter point is reached, evaporative heat loss, at a given air temperature, is exactly the same whether humidity be high or low.

It is important to note, however, that even in the range where this type of adaptation is thermally adequate, the process of sweat secretion is associated with sensations of marked discomfort. It has been demonstrated that *wetted area*, the physiological index of sweat secretion, shows a correlation of 0.65 ± 0.006 with expressions of discomfort[11]. Furthermore, there develops, even under moderate conditions of overheating, a definite disinclination for physical activity. The *New York State Commission* found that experimental subjects performed 28 per cent less work at 86 F with 80 per cent relative humidity than at 68 F with 50 per cent relative humidity[12].

TABLE 2. UPPER LIMITS OF EVAPORATIVE REGULATION
(*Clothed Subject*)

Environmental Temperature		Relative Humidity Per Cent
Deg F	Deg C	
126.9	52.7	0
122.0	50.0	5
113.0	45.0	18
108.5	42.5	26
104.5	40.0	38
99.5	37.5	51
95.0	35.0	69
90.5	32.5	89
87.8	31.0	100

The upper limit of this *zone of evaporative regulation* is, of course, established by the combination of air and wall temperature, relative humidity and air movement beyond which the sweat runs off without evaporation and cooling of the body surface. Limits for unclothed, semi-reclining subjects, with minimal air movement are shown in Fig. 2. Studies at the *John B. Pierce Laboratory of Hygiene* have established the upper limits of evaporative regulation for the clothed, semi-reclining body at an air movement of 17 fpm as shown in Table 2[13].

Since 1922 notable progress has been made by the A.S.H.V.E. Research Laboratory in fixing corresponding upper limits for subjects engaged in

[11]Relations Between Atmospheric Conditions, Physiological Reactions, and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington, and A. P. Gagge (*American Journal of Hygiene*, Vol. 26, July, 1937, p. 102).

[12]Ventilation Report of the New York State Commission on Ventilation (E. P. Dutton Co., N. Y., 1923).

[13]The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington, and A. P. Gagge (*American Journal of Physiology*, Vol. 124, December, 1938, p. 692).

active work[14], limits which are, of course, much lower than those cited for subjects at rest. The upper limit of effective temperature to which the human organism is capable of adapting itself without serious discomfort or injury to health is 90 deg ET (Effective Temperature) for men at rest and between 80 and 90 deg ET for men at work depending upon the rate of work. Within these limits a new equilibrium is established at a higher body temperature level through a chain of physiological adjustments. The heat regulating center fails when the external temperature is so abnormally high that bodily heat cannot be eliminated as fast as it is produced. Part of it is retained in the body, causing a rise in skin and deep tissue temperature, an increase in the heart rate, and accelerated respiration. (See Table 3.) In extreme heat the metabolic rate is markedly increased owing to the excessive rise in body temperature, and a vicious cycle results which may eventually lead to serious physiologic damage or heat exhaustion[15].

TABLE 3. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Cheek Temp (Deg Fahr)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	....	....	....	....	225,000	0.0	6	0.5
70	....	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0	103[b]	2.7
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a]Data by A.S.H.V.E. Research Laboratory.
[b]Computed value from exposures lasting less than one hour.

Examples of this are met with in unusually hot summer weather and in hot industries where heat loss from the body by radiation and convection is impossible. Consequently, the workers depend entirely on evaporation for the elimination of body heat. They stream with perspiration and drink liquids abundantly to replace the loss.

One of the deleterious effects of high temperatures is that the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is suggested that the feeling of lassitude and discomfort experienced is due in part to the anaemic condition of the

[14]A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten, and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59).

[15]A.S.H.V.E. RESEARCH REPORT No. 719—Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by W. J. McConnell, C. P. Yaglou and W. B. Fulton (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 123). A.S.H.V.E. Research Paper—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion (The Role of Posture), by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1940, p. 122).

brain. The stomach loses some of its power to act upon the food, owing to a diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract[16]. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, it is recommended that 6 g of sodium chloride and 4 g of potassium chloride be added to a gallon of water[17].

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

The control of hot conditions by chilling the walls of an occupied space has only limited application in practice. Either air-cooling, increase in air movement, or dehumidification, or any combination of these procedures may be used in practice to keep a space cool in summer.

ADAPTATION TO COLD CONDITIONS

When the heat demand of the environment exceeds the metabolic output, the chief changes which occur, as external temperature decreases, are (1) increased heat loss due to radiation plus convection, and (2) increased positive storage or cooling of the body tissues. It will be noted in Fig. 1 that the slope of the line representing heat loss due to radiation plus convection changes as one passes from the *zone of evaporative regulation* to the *zone of body cooling*. The less abrupt slope in the latter zone is due to a progressive fall in skin temperature which is the only mechanism the body calls into play in this region to adapt to a cool environment. Under colder conditions, or after longer periods of time, a second mechanism, increased metabolism, may become operative but this does not appear in the experiments here reviewed.

For a fall in operative temperature from 88 to 68 F the mean skin temperature decreases from 94 to 84 F. The temperature of the lower extremities falls most rapidly while that of the head or trunk may decrease less. This type of regulation is, however, as pointed out previously, incomplete; and positive storage (cooling of the body) increases progressively as external temperature falls. Chilling, then, imposes an extra load upon the heat-producing organs to maintain body temperature. The strain falls largely upon digestion, metabolism, blood circulation, and the kidneys, and indirectly upon the nervous system[18]. In extremely

[16]Influence of Effective Temperature upon Bactericidal Action of Gastro-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp. Biol. Med.*, Vol. 24, 1927, p. 832).

[17]Some Effects of High Air Temperatures Upon the Miner, by K. N. Moss (*Transactions Institute of Mining Engineers*, Vol. 66, 1924, p. 284).

[18]Preventive Medicine and Hygiene, by M. J. Rosenau (6th Edition, p. 909).

cold atmospheres compensation by increased metabolism becomes inadequate. The body temperature falls and the reflex irritability of the spinal cord is markedly affected. The organism may finally pass into an unconscious state which ends in death.

A moderate amount of variability in temperature is known to be beneficial to health, comfort, and the performance of physical and mental work. On the other hand, extreme changes in temperature, such as those experienced by passing from a warm room to the cold air out-of-doors, appear to be harmful to the tissues of the nose and throat which are the portals for the entry of respiratory diseases.

Experiments show that chilling causes a constriction of the blood vessels of the palate, tonsils, throat, and nasal mucosa, which is accompanied by a fall in the temperature of the tissues. On re-warming, the palate and throat do not always regain their normal temperature and blood supply. This anaemic condition favors bacterial activity and it probably plays a part in favoring infection with common colds and other respiratory diseases. Lowered resistance may also be related to a diminution in the number and phagocytic activity of the leucocytes (white blood cells) brought about by exposure to cold and by changes in temperature.

Sickness records in industries seem to strengthen this belief. The *Industrial Fatigue Research Board* of England[19] found that with the workers exposed to high temperatures and to changes in temperature, namely, steel smelters, puddlers, and general laborers, there is an excess of all sickness, the excess among the puddlers being due chiefly to respiratory diseases and rheumatism. The causative factor was not the heat itself but the sudden changes in temperature to which the workers were exposed. The tin-plate millmen who were not exposed to chills, since they work almost continuously throughout the shift, had no excess of rheumatism and respiratory diseases. On the other hand, the blast-furnacemen, who work mostly in the open, showed more respiratory sickness than the steel workers. This experience in British factories is well in accord with the findings in American industries[20,21]. According to these data the highest pneumonia death rate is associated with dust, extreme heat, exposure to cold, and to sudden changes in temperature.

In the ordinary dwelling not equipped with a fan system during the winter season, air movement is likely to be minimal and the only air movements affecting comfort are those due to cold walls or convective heating. Below an air temperature of 77 F, for the clothed, semi-reclining body, sweat secretion is at a minimum and, even with high atmospheric humidity, all the moisture available is evaporated. Under conditions of low air movement, and with a relative humidity of 30-35 per cent, the evaporative heat loss is only 1.5 Btu per square foot of body surface greater than with a relative humidity of 75-80 per cent. This is the equivalent of decreasing the air temperature by less than 1 F.

---

[19]Fatigue and Efficiency in the Iron and Steel Industry, by H. M. Vernon (*Industrial Fatigue Research Board* Report No. 5, 1920, London).

[20]Iron Foundry Workers Show Highest Percentage of Deaths from Pneumonia (*Statistical Bulletin*, Metropolitan Life Insurance Co., New York, N. Y., 1928).

[21]The Pneumonia Problem in the Steel Industry, by D. K. Brundage and J. J. Bloomfield (*Journal of Industrial Hygiene*, 14, December, 1932).

RELATION OF AIR CONDITIONING NEEDS TO METABOLISM

The major objective of heating and ventilation is to balance heat losses from the human body. The basic factor is human metabolism. The desirable environment, from the standpoint of heat loss, depends directly on the heat produced in the body and this heat may be five times as great when a man is exercising violently as when he is reclining and at rest.

TABLE 4. RELATION BETWEEN METABOLIC RATE AND ACTIVITY[a]

Activity	Hourly Metabolic Rate for Avg Person or Total Heat Dissipated, Btu per hour	Hourly Sensible Heat Dissipated, at 79 F, Btu per Hour	Hourly Latent Heat Dissipated, at 79 F, Btu per Hour	Moisture Dissipated per Hour	
				Grains	Pounds
Basal	291	145	145	979	0.140
Seated at Rest	384	225	159	1070	0.153
Reading Aloud (Seated)	420	225	195	1316	0.880
Standing at Rest	431	225	206	1390	0.199
Hand Sewing (Seated)	441	225	216	1458	0.208
Knitting 23 stitches per minute on Sweater	462	225	237	1600	0.229
Dressing and Undressing	468	225	243	1640	0.234
Tailor	482	225	257	1740	0.248
Singing	486	225	261	1762	0.252
Office Worker Moderately Active	490	225	265	1790	0.256
Light Work Standing	549	225	324	2187	0.312
Typewriting Rapidly	558	225	333	2248	0.321
Ironing with 5 lb iron	570	225	345	2329	0.333
Dishwashing—Plates, Bowls, Cups and Saucers	600	225	375	2530	0.362
Clerk Moderately Active Standing at Counter	600	225	375	2530	0.362
Book Binder	626	225	401	2710	0.387
Shoemaker	661	225	436	2940	0.420
Sweeping Bare Floor 38 Strokes per Minute	672	229	443	2990	0.427
Pool Player	680	230	450	3040	0.434
Walking 2 mph, Light Dancing	761	250	511	3450	0.493
Light Metal Worker (at Bench)	862	277	585	3950	0.564
Painter of Furniture (at Bench)	876	280	596	4020	0.575
Carpenter	954	307	647	4367	0.624
Restaurant Serving	1000	325	675	4556	0.651
Pulling Weight	1041	335	708	4745	0.677
Walking 3 mph	1050	339	711	4750	0.679
Walking 4 mph, Active Dancing, Roller Skating	1390	452	938	6330	0.904
Walking Down Stairs	1444	467	977	6595	0.942
Stone Mason	1490	490	1000	6750	0.964
Bowling	1500	490	1010	6820	0.974
Man Sawing Wood	1800	590	1210	8170	1.167
Swimming	1986	.....	.....	.....	.....
Running 5.3 mph	2268	.....	.....	.....	.....
Walking 5 mph	2330	.....	.....	.....	.....
Walking Very Fast 5.3 mph	2580	.....	.....	.....	.....
Walking Up Stairs	4365	.....	.....	.....	.....
Maximum Exertion Different People	3000-4800	.....	.....	.....	.....

[a]These metabolic rates were compiled by the A.S.H.V.E. Research Laboratory from actual tests, from other authoritative sources, and from estimates based upon various considerations. Division of the total heat dissipation into latent and sensible rates is based on actual test data and on various considerations for metabolic rates up to 1250 Btu per hour, and extrapolated for higher rates. Values for total heat dissipation for a person at rest apply for a dry-bulb temperature range from approximately 60 to 90 F; for other than rest conditions the values apply for a similar but lower temperature range. Below these temperature ranges metabolic rates and total rates of heat dissipation increase, while above these ranges metabolic rates increase slightly and total heat dissipation rates decrease rapidly. Division of total dissipation rates into sensible and latent heat holds only for a dry-bulb temperature of 79 F. For lower temperatures, sensible heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true.

Therefore, there is no absolute optimum of air temperature or other environmental conditions, which will meet all cases. With moderate relative humidity and minimum air movement, an air temperature of 80 F has been found ideal for the lightly clothed subject at rest in a semi-reclining position. In factories where light work is performed in summer time, the ideal has been found to be about 76 F. For children

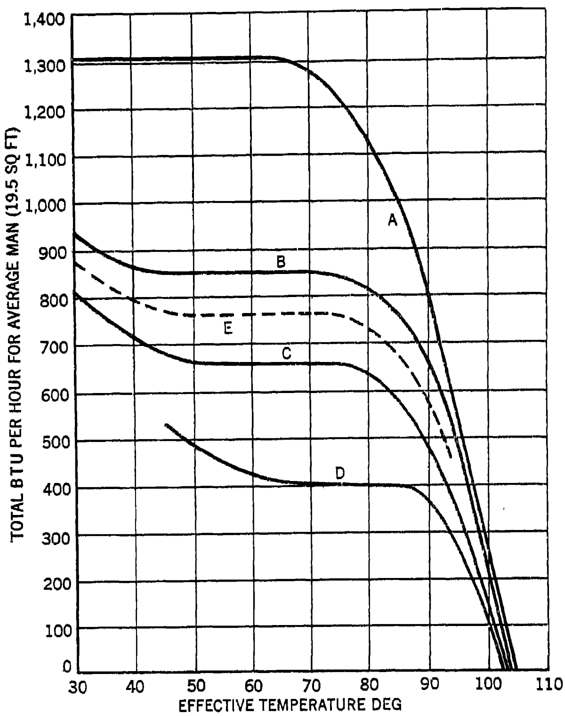


FIG. 3. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Persons working so as to have a metabolic rate of 1310 Btu per hour. Curve *B*—Persons working so as to have a metabolic rate of 850 Btu per hour. Curve *C*—Persons working so as to have a metabolic rate of 660 Btu per hour. Curve *D*—Persons seated at rest, or with a metabolic rate of 400 Btu per hour. Curves *B* and *D* based on test data covering a wide temperature range. Curves *A* and *C* based on test data at an Effective Temperature of 70 deg and extrapolation of Curves *B* and *D*. All curves are averages of values for high and low relative humidities which apply with satisfactory accuracy for most considerations. For special problems requiring a higher degree of accuracy see more detailed A.S.H.V.E. Research Laboratory reports.

(who have a high metabolism) at school, in winter clothing, 70 F has been considered correct; while in a gymnasium, 55 F is a desirable temperature.

The wide variations in metabolic activities with which the engineer must be prepared to cope and the influence of such variations in metabolism on the heat load contributed by the human body to the environment are given in Table 4 and in Figs. 3, 4 and 5. The curves in the figures are based on certain averages of test results with different humidities, and

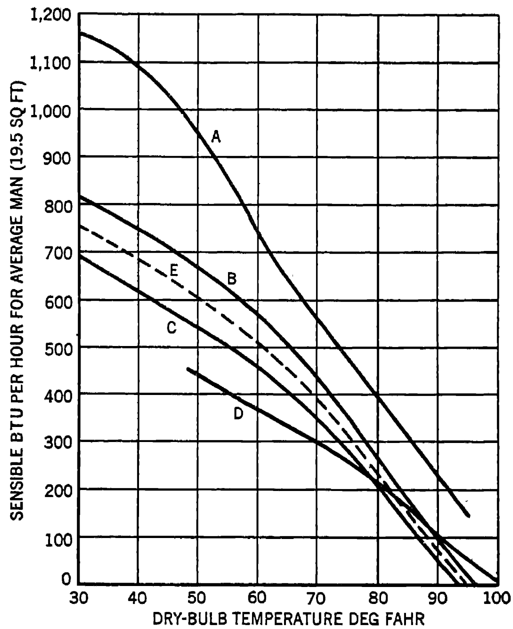


FIG. 4. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

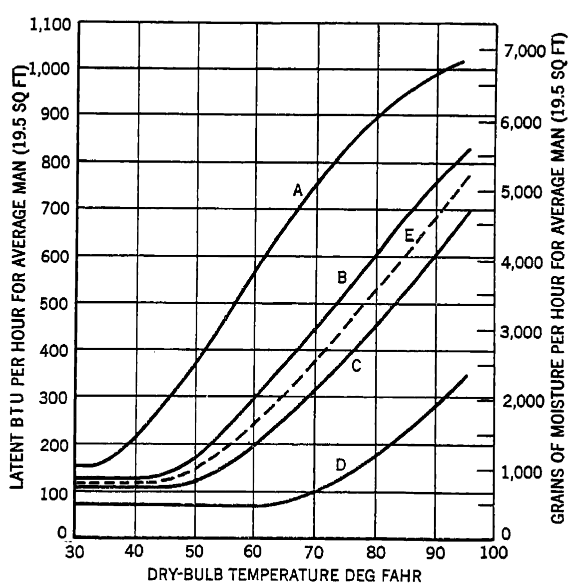


FIG. 5. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Loc. Cit. See footnote *a*, Fig. 3.

are sufficiently accurate for most practical applications. Where greater precision in the applications of the results is required, or for extreme variations in temperature and humidity, the reports[22] covering the A.S.H.V.E. Laboratory work may be consulted.

The curves in Figs. 3, 4 and 5, and proper interpolation between these curves make it possible to apply the data to persons engaged in any type of work or physical activity, providing the resulting metabolic rate is known. As an example, if it is found that a certain type of work results in a metabolic rate of approximately 760 Btu per hour for an average person working in an atmosphere of 70 ET, then his total rate of heat dissipation to atmospheres of various temperature will be approximately as given by the broken-line curve in Fig. 3. The broken line curves in Figs. 4 and 5 give the rate of sensible and latent heat dissipation of the person for different dry-bulb temperatures.

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue. They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature averaging 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. An environment averaging 64 F for the 24-hour period is associated with minimal mortality[23].

Within limits, however, there does occur a definite adaptation to external temperature level. People and animals raised under conditions of tropical moist heat stand chilling poorly as they are unable quickly to increase internal combustion to keep up the body temperature. For this reason they have trouble standing the cold, stormy weather of the temperate zones, and when exposed to it are very susceptible to respiratory infections. Likewise, people living in cool climates suffer greatly in the moist heat of the tropics until their adaptive mechanism has been trained. Within a couple of years, however, they find themselves standing the heat much better and disliking the cold.

[22]A.S.H.V.E. RESEARCH REPORT NO. 830—Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245). Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (*American Journal of Physiology*, Vol. 88, 1929, p. 386). A.S.H.V.E. RESEARCH REPORT No. 908—Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 541). Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (*American Journal of Hygiene*, Vol. XIII, 1931, No. 2, p. 415). A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten, and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59).

[23]Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1928.

The adaptive level changes somewhat with the season[24]. There are also marked differences between the sexes. In the cold zone the thickness of the thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone, the threshold of sweating and skin temperature levels are both higher for women.

Finally, the thickness and insulating value of the clothing worn is an important factor in the determination of the comfort level.

EFFECTIVE TEMPERATURE INDEX

Sensations of warmth or cold depend, not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer, upon air movement and upon radiation effects. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler. Radiation from cold or warm surfaces is another important factor under certain conditions.

Combinations of temperature, humidity, and air movement which induce the same feeling of warmth are called thermo-equivalent conditions. A series of tests[25,26,27,28] at the A.S.H.V.E. Research Laboratory, Pittsburgh, established the equivalent conditions met with in general air conditioning work. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

The numerical value of the effective temperature index for any given air conditions is fixed by the temperature of calm (15 to 25 fpm air movement) saturated air which induces a like sensation of warmth or cold. Thus, any air condition has an effective temperature of 60 deg, for instance, when it induces a sensation of warmth like that experienced in calm air at 60 deg saturated with moisture. The effective temperature index cannot be measured directly but is computed from the dry- and wet-bulb temperature for a given air velocity or using charts (see Figs. 6, 7 and 8) or tables. The relation of winter and summer sensations of

[24]The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology*, Vol. 124, December, 1938, p. 692).

[25]A.S.H.V.E. Research Report No. 673—Determination of the Comfort Zone, by F. C. Houghten and C. P. Yagloglou (A.S.H.V.E. Transactions, Vol. 29, 1923, p. 361).

[26]A.S.H.V.E. Research Report No. 691—Cooling Effect on Human Beings by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. Transactions, Vol. 30, 1924, p. 193).

[27]A.S.H.V.E. Research Report No. 717—Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller (A.S.H.V.E. Transactions, Vol. 31, 1925, p. 89).

[28]A.S.H.V.E. Research Report No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague, and W. E. Miller (A.S.H.V.E. Transactions, Vol. 32, 1926, p. 315).

comfort to wet- and dry-bulb temperature at low air movement is shown in Fig. 6. Relations between moisture content and various dry-bulb temperatures to wet-bulb readings and effective temperatures are depicted in Fig. 7. Effective temperatures for various combinations of wet- and

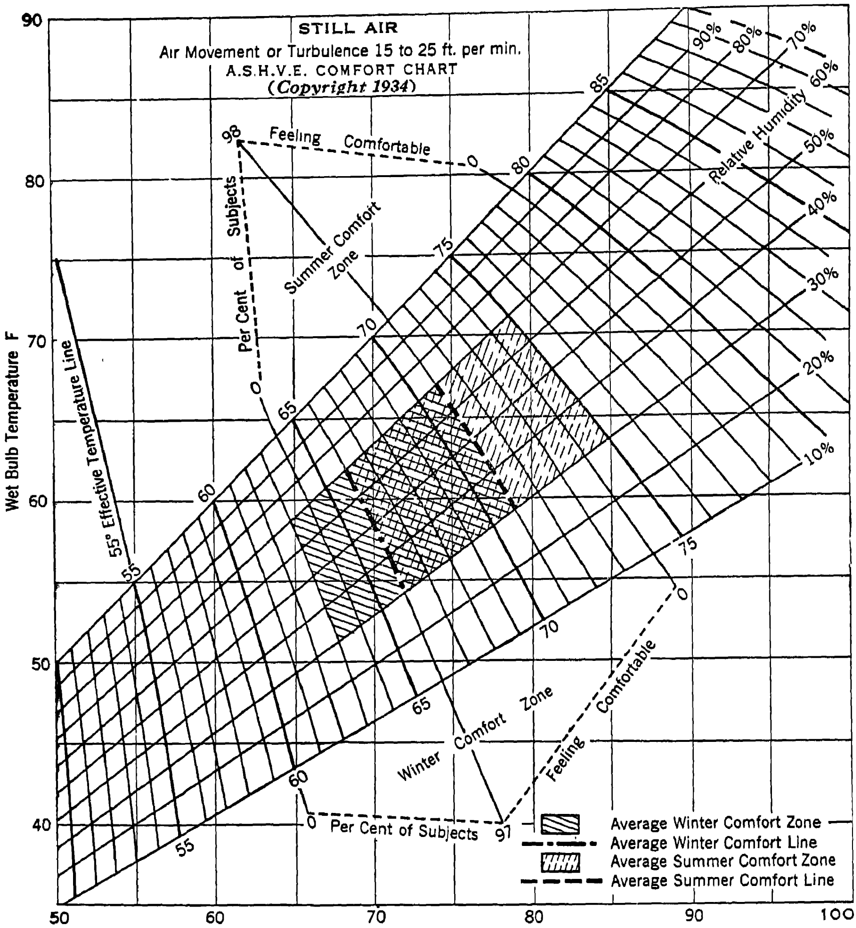


FIG. 6. A.S.H.V.E. COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours.

dry-bulb temperature readings are given in Fig. 8 accompanied by various air movements.

The Comfort Chart shown applies to average normal and healthy persons adapted to American living and working conditions. Application is limited to sedentary or light muscular activity. It will probably not

apply to inhabitants of other countries where the living conditions, climate, heating methods, and clothing are materially different than those of the subjects employed in experiments at the Research Laboratory.

In rooms in which the average wall surface temperature is considerably below or above air temperature, a correction must be applied to the readings of the dry-bulb thermometer to allow for such negative or posi-

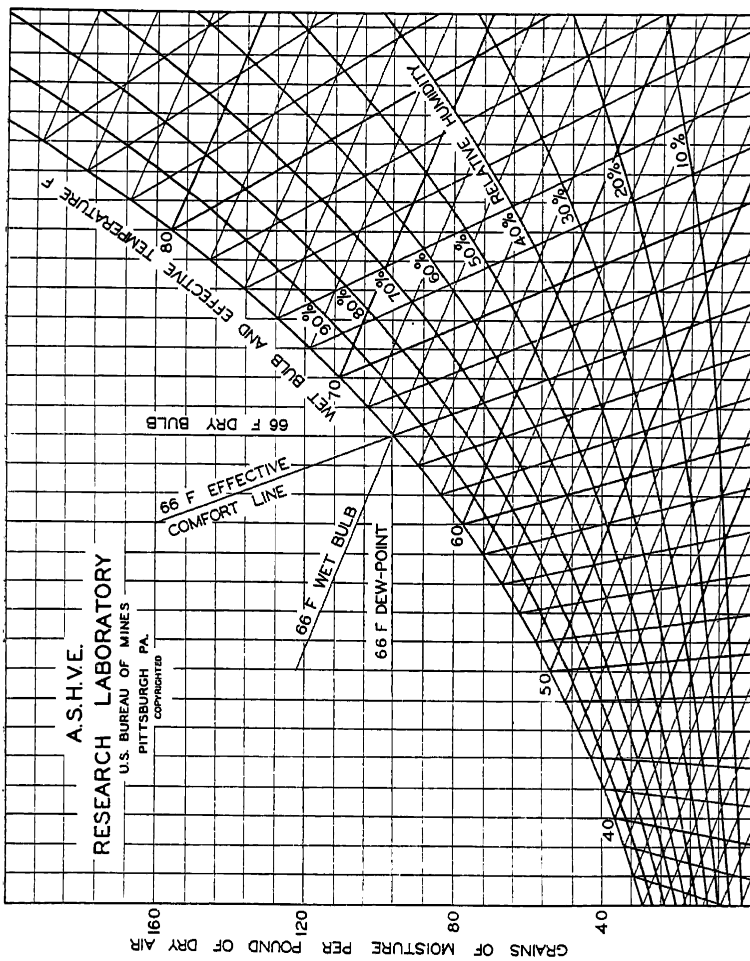


FIG. 7. PSYCHROMETRIC CHART, PERSONS AT REST, NORMALLY CLOTHED, IN STILL AIR

tive radiation. In Fig. 9 is given the cooling effect of cold walls as determined at the A.S.H.V.E. Research Laboratory[29] by trained subjects passing back and forth from a small experimental room having three cold walls, to a control room with walls and air at the same temperature.

It can be seen in Fig. 9 that in comparison with air and walls at 70 F in the control (warm wall) room, the cooling effect of three cold walls, at 55 F of the experimental room was 4 F. Therefore, for the same feeling

[29]Loc. Cit. Note 28.

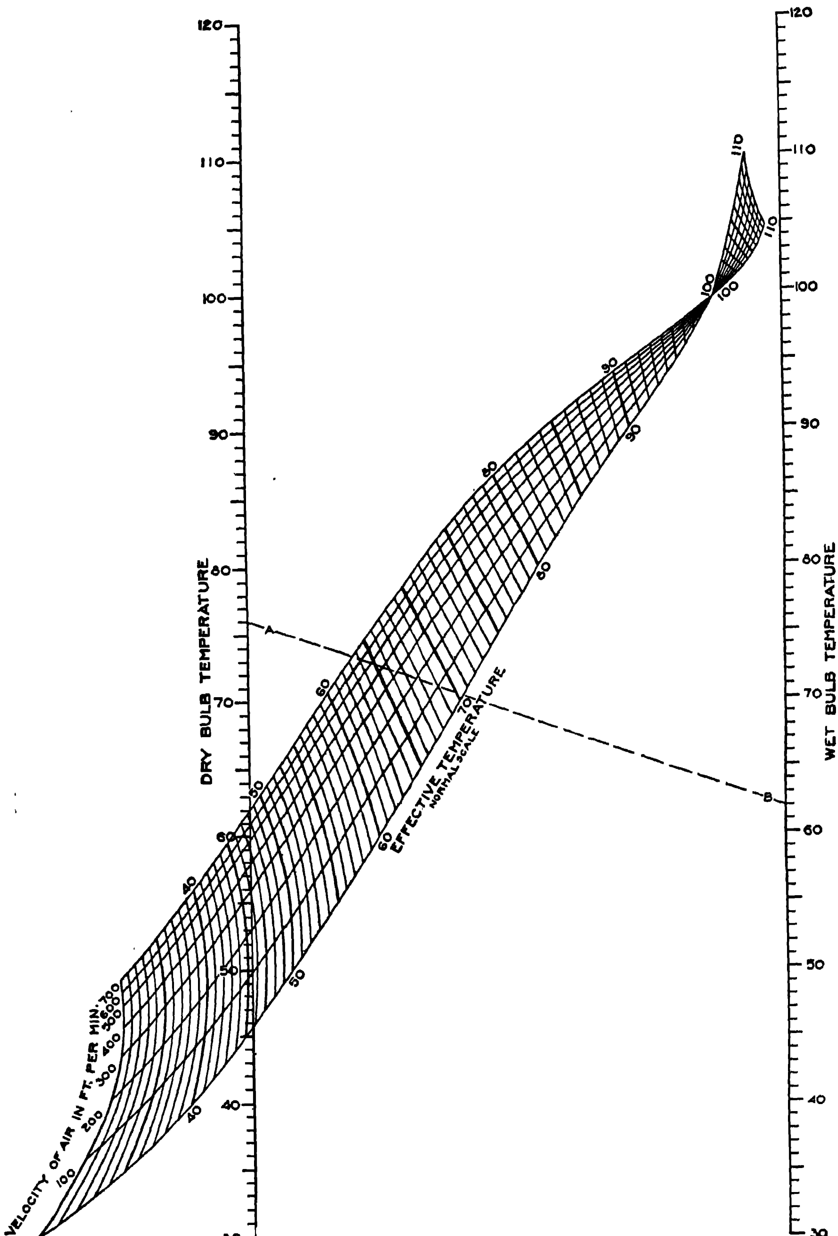


FIG. 8. EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

A. Clothing: Customary indoor clothing. *B. Activity:* Sedentary or light muscular work. *C. Heating Methods:* Convection type, *i.e.* warm air, direct steam or hot water radiators, plenum systems.

of warmth, the temperature in the experimental room should be increased to 74°F. The reverse would hold in rooms with high wall surface temperature; a lower air temperature would be required to compensate for positive radiations to the occupants.

In Fig. 6 is shown the A.S.H.V.E. winter comfort zone which was determined experimentally with large groups of men and women subjects wearing customary indoor winter clothing[30,31]. The extreme comfort zone includes conditions between 60 and 74 deg ET in which one or more of the experimental subjects was comfortable. The average comfort zone includes conditions between 63 and 71 deg ET conducive to comfort

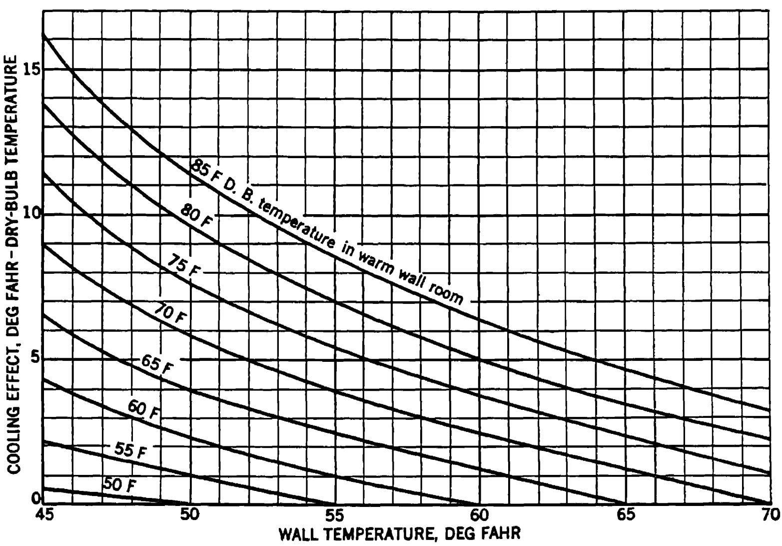


Fig. 9. Cooling Effect of Three Cold Walls in a Small Experimental Room, as Determined by Comparison with Sensations in a Room of Uniform Wall and Air Temperature

in 50 per cent or more of the experimental subjects. The most popular effective temperature was found to be 66 deg, and was adopted by the Society[32] as the *winter comfort* line for individuals at rest wearing customary winter clothing.

The comfort line separates the cool air conditions to its left from the warm air conditions to its right. Under the air conditions existing along or defined by the comfort line, the body is able to maintain thermal equilibrium with its environment with the least conscious sensation to the individual, or with the minimum physiologic demand on the heat regulating mechanism.

The average winter comfort line (66 deg ET) applies to average American men and women living inside the broad geographic belt across the

[30]Loc. Cit. Note 25.

[31]The Summer Comfort Zone; Climate and Clothing, by C. P. Yaglou and Philip Drinker, (A.S.H.V.E. Transactions, Vol. 35, 1929, p. 269).

[32]How to Use the Effective Temperature Index and Comfort Charts (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 410).

United States in which central heating of the convection type is generally used during four to eight months of the year. It does not apply to rooms heated by radiant energy, rooms with excessive glass area or rooms with poorly insulated or cold walls.

In densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures may be necessary than those indicated by the comfort line on account of counter-radiation between the bodies of occupants in close proximity[33].

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated by the comfort line, it should not be expected that all the occupants of a room will feel perfectly comfortable. When the winter comfort line is applied in accordance with the foregoing recommendations, the majority of the occupants will be perfectly comfortable, but there will always be a few who would feel *a bit too cool* and a few *a bit too warm*. These individual differences among the minority should be counteracted by suitable clothing.

Air conditions lying outside the average comfort zone but within the extreme comfort zone may be comfortable to certain persons. In other words, it is possible for half of the occupants of a room to be comfortable in air conditions outside the *average* comfort zone, but in the majority of cases, if not in all, these conditions will be well within the extreme comfort zone as determined experimentally.

For prematurely born infants, the optimum temperature varies from 100 to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent[34]. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 to 68 F with natural indoor humidities. For school children, the studies of the *New York State Commission on Ventilation* place the optimum air conditions at 66 to 68 F temperature with a moderate humidity and a moderate but not excessive amount of air movement[35].

Satisfactory comfort conditions for men at work are found to vary from 40 to 70 deg ET, depending upon the rate of work and amount of clothing worn[36]. In hot industries, 80 deg ET is considered the upper limit compatible with efficiency, and, whenever possible, this should be reduced to 70 deg ET or less.

The summer comfort zone is much more difficult to fix than the winter zone owing to the complicating factor of sweating in warm weather. A given air condition which is comfortable for persons with dry skin and clothing may prove too cold for those perspiring, as is the case, for instance, with employees and customers in a cooled store, restaurant, or theater, on a warm summer day. The conditions to be maintained in different types of public buildings depend to a large extent upon the occupant's length of stay and upon the prevailing outdoor condition.

---

[33]Loc. Cit. Note 31.

[34]Application of Air-Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker, and K. D. Blackfan (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 383).

[35]Loc. Cit. Note 12.

[36]Loc. Cit. Note 28.

In Fig. 6 is shown the summer comfort zone for exposures of 3 hours or more, after adaptation has taken place. The average zone extends from 66 to 75 deg ET, with a comfort line at 71 deg ET, as determined at the *Harvard School of Public Health*[37]. These effective temperatures average about 4 deg higher than those found in winter when customary winter clothing was worn. The variation from winter to summer is probably due partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons.

The basic summer comfort zone presented in Fig. 6 prescribes conditions of choice for continuous exposures, as in homes, offices, etc., without regard to costs, prevailing outdoor air conditions, and temperature contrasts upon entering or leaving the cooled space. A great number of persons seem to be content with a higher plane of indoor temperature, particularly when the matter of first cost and cost of operation of a cooling plant is given due consideration.

According to previous investigations[38], an indoor temperature of about 80 F with relative humidities below 55 per cent, or 74.5 deg ET and lower, results in satisfactory comfort conditions in the living quarters of a residence, and while this condition is not representative of optimum comfort it provides for sufficient relief in hot weather to be acceptable to the majority of users. Experience in a number of air conditioned office buildings, including the *New Metropolitan Life Building in New York*[39], indicates that a temperature of about 80 F with a relative humidity between 45 and 55 per cent (73 to 74.5 deg ET) is generally satisfactory in meeting the requirements of the employees.

Recent studies carried out under the auspices of the Society have shown that effective temperatures of 71 F in Toronto, 72 F in Pittsburgh and Minneapolis, and 73 F in Texas, were generally found most comfortable by sedentary workers during the summer months[40]. The preferred indoor optimum varied with outdoor temperature during successive summer months[41]; and interesting data were obtained as to contrast effects between indoor and outdoor situations[42].

In artificially cooled theaters, restaurants, and other public buildings

[37]Loc. Cit. Note 31.

[38]A.S.H.V.E. RESEARCH REPORT No. 1012—Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 207).

[39]The Air Conditioned System of the New Metropolitan Building—First Summer's Experience, by W. J. McConnell and I. B. Kagey (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 217).

[40]A.S.H.V.E. RESEARCH REPORT No. 1035—Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215). A.S.H.V.E. RESEARCH REPORT No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145). A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet, and R. W. Qualley (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337). Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 549). A.S.H.V.E. RESEARCH REPORT No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 459). A.S.H.V.E. RESEARCH REPORT No. 1136—Summer Cooling Requirements in Washington, D. C., and Other Metropolitan Districts, by F. C. Houghten, Carl Gutberlet and Albert A. Rosenberg (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 577). Reactions of 745 Clerks to Summer Air Conditioning, by W. J. McConnell and M. Spiegelman (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1940, p. 317).

[41]A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337).

[42]A.S.H.V.E. RESEARCH REPORT No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air-Conditioned Offices, by A. B. Newton, F. C. Houghten, C. Gutberlet, R. W. Qualley, and M. C. W. Tomlinson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 571).

where the period of occupancy is short, the contrast between outdoor and indoor air conditions becomes the deciding factor in regards to the temperature and humidity to be maintained. The object of cooling such places in the summer is to provide sufficient relief from the heat without causing sensations of chill or intense heat on entering and leaving the building.

The Comfort Chart has proved one of the most valuable tools of the heating and ventilating engineer. Research in other laboratories[43] has shown somewhat different quantitative relationships but these are easily accounted for by differences in metabolic activity and clothing. It appears from this other work that the Comfort Chart may perhaps exaggerate somewhat the influence of relative humidity at low temperatures and underestimate it at high temperatures; but the chart gives an essentially correct picture of those relations which exist where radiant influences and high air movement are not important factors.

PHYSIOLOGICAL OBJECTIVES OF HEATING AND VENTILATION

Aside from the removal of toxic fumes and dusts from heating appliances and industrial processes, the chief task of the heating and ventilating engineer is to keep his clients warm in winter and cool in summer.

For the normally vigorous person, normally clothed, and at rest, an air temperature of 65 F should be provided at knee-height, 18 in. in order to prevent chilling of the legs and feet. With some heating systems, this will correspond to 70 F at a 5 ft height. Air temperature may be increased or decreased in order to compensate for deviations of *mean radiant temperature* above or below air temperature.

In rooms occupied by persons of sub-normal vitality, knee-height temperatures must be higher than 65 F. Since dwellings are designed for occupancy by old people and children, the heating system should be able to provide a temperature of 70 F at knee-height under ordinary winter conditions.

The maintenance of such conditions as these in winter depends on three major factors, the heat produced in the occupied space, the heat absorbed from the sun and the heat loss through the walls, floor and ceiling of the structure to cold air and earth. Taking these up in the order in which they occur, in planning a new structure it is essential to remember the important effect of orientation and fenestration of the building with respect to the absorption of radiant heat from the sun. It has recently been shown that, in the vicinity of New York effective sun-heat on a wall facing south is almost five times as great in winter as in summer, but on a wall facing west-north-west it is six times as great in summer as in winter[44]. The orientation of the same one-story house (in a laboratory model) was changed from a position in which its principal rooms faced northwest to a position in which these rooms (with rearranged and slightly increased fenestration) faced west of south. This change decreased average summer sun-heat to one-ninth and increased average winter sun-heat to fourfold of its value with the original orientation.

[43]Loc. Cit. Note 9.

[44]Solar Radiation as Related to Winter Heating in Residences, by H. N. Wright (*Report of John B. Pierce Foundation*, January 20, 1936).

The choice between the various methods of heating depends, of course, on many engineering and other factors. From the standpoint of human health and comfort, however, it is important to minimize floor-ceiling differentials as far as possible to avoid hot heads and cold feet. Furthermore, when the problem is a heating one, low air movement is desirable, since air temperature must be raised to balance the cooling effect of air motion.

Where occupants are closely aggregated, a new problem comes in, the removal of the excess heat produced by the human body itself. If the temperature of such a space be correctly adjusted when the occupants enter, it will steadily rise during the period of occupancy as a result of the heat produced by the occupants in the process of metabolism. Of the 400 Btu given off in metabolism 100 would perhaps be lost in evaporation, leaving 300 Btu per person per hour to warm the air. In a room containing many persons, the effects of this body heat can be neutralized by outside air without producing unpleasant and dangerous drafts on those near the windows or other inlets. The supply of air should be so tempered as to avoid drafts but in an amount and at a temperature which will remove the sensible heat produced by human metabolism. With no heat loss through walls (as in an interior auditorium) this will require 28 cfm of air per person with admitted air at 60 F and a maximum figure of 70 F, for air leaving the room. Under practical conditions, with one or more cold walls, and a room containing a moderate number of occupants and ample cubic space, window ventilation with deflectors and a gravity exhaust duct may suffice. With crowded rooms, and with any rooms containing 50 or more occupants, forced ventilation will be essential.

SUMMER COMFORT

The problem of keeping cool in summer is physiologically as important as keeping warm in winter. In summer the relative humidity of the atmosphere is of importance, along with air temperature, air movement, and wall temperature. There is no very practical method of cooling walls, but summer comfort can be promoted by modifying either one of the other three factors involved.

Increase of comfort by air movement can be effected in two ways. The first of these is promotion of natural circulation by cross or through ventilation; and here the architect is responsible for providing room planning and fenestration which will make such natural ventilation possible. In the lowest cost housing this should be considered as essential.

The direct control of air temperature and humidity is, of course, the ideal solution where the cost of a complete air conditioning equipment can be met. Where this objective is attained, there are two schools of thought concerning the relation between temperature and humidity to be maintained. For a given effective temperature some engineers favor comparatively low temperature with a high humidity as this results in a reduction of refrigeration requirements. Preliminary experiments at the A.S.H.V.E. Laboratory[45] would seem to indicate no appreciable impair-

[45]A.S.H.V.E. RESEARCH REPORT No. 1035—Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215). A.S.H.V.E. RESEARCH REPORT No. 1055—Cooling Requirements for Summer Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145.)

ment of comfort with relative humidities as high as 80 per cent, provided the effective temperature is between 70 and 75 deg.

The second school favors a higher dry-bulb temperature, according to the prevailing outdoor dry-bulb, with a comparatively low humidity (well below 50 per cent), the main purpose being to reduce temperature contrasts upon entering and leaving the cooled space and to keep the clothing and skin dry. This second scheme requires more refrigeration with the present conventional type of apparatus.

In connection with summer cooling mention should be made of the

TABLE 5. DESIRABLE INSIDE CONDITIONS IN SUMMER CORRESPONDING TO OUTSIDE TEMPERATURES[a]

Occupancy Over 40 Min

Outside Dry-Bulb Deg F	Inside Air Conditions				
	Effective Temperature	Dry-Bulb Deg F	Wet-Bulb Deg F	Dew-Point Deg F	Relative Humidity Per Cent
100	75	83	66	56	40
	75	82	67	59	45
	75	81	68	61	51
	75	80	70	65	60
95	74	82	64	53	36
	74	81	66	57	44
	74	80	67	60	51
	74	79	68	62	57
	74	78	70	66	68
90	73	81	63	52	36
	73	80	64	54	41
	73	79	66	59	50
	73	78	67	61	56
85	72	80	61	48	32
	72	79	63	53	41
	72	78	64	56	46
	72	77	66	60	56
80	71	78	61	49	36
	71	77	63	54	45
	71	76	64	57	52
	71	75	66	61	61

[a]Applicable to individuals engaged in sedentary or light muscular activity.

danger of over-doing it. Possible impairment to health may have resulted from the sharp contrast between air conditioned trains and the outside atmosphere in the early days of this practice. Current practice in theaters, restaurants, etc., follows a schedule similar to that shown in Table 5. This schedule should be used with considerable judgment, depending on the occupancy and local climatic conditions. There are some indications that a definite indoor effective temperature may be applicable throughout the cooling season, but other observations seem to show that changing indoor conditions are desirable with violently changing outdoor weather conditions. It is questionable whether entirely satisfactory air conditions could be adduced for practical use to meet the changing requirements of patrons from the time they enter to the time they leave a cooled space. Too many uncontrollable variables enter into the prob-

lem. Work now going on at the A.S.H.V.E. Research Laboratory and other institutions may throw considerable light on this complex problem.

For cooled banks and stores where the customers come and go, spending but a few minutes in the cooled space, observations indicate a schedule about 1 deg of effective temperature higher than that shown in Table 5. Laboratory experiments with exposures of 2 to 10 min indicate temperatures 2 to 10 F higher than those in Table 5, but with much lower relative humidities.

It should be kept in mind that southern people, with their more sluggish heat production and lack of adaptability, will demand a comfort zone several degrees higher than that for the more active people of northern climates. Instead of the summer comfort lines standing at 71 deg ET as here given, it was found to be much higher for foreigners in Shanghai where climatic conditions are similar to those of our gulf states. This difference in adaptability of people forms a very real problem for air conditioning engineers. Cooling of theaters, restaurants, and other public buildings in southern climates cannot be based on northern standards without considerable modification.

RELATION OF AIR AND WALL TEMPERATURES

In the previous discussion, it has generally been assumed that air and wall temperatures are alike and, of course, this is roughly the case. In a room heated by pure convection, the walls are generally heated by the warm air and in a room heated by pure radiation, the air is gradually warmed by convection from surfaces which have themselves been warmed by absorption of radiant heat. In ordinary indoor spaces, whatever the heating method, the mean of floor, ceiling and all four walls, is not likely to differ more than 3 F from the air temperature; but individual walls which have a dominant effect on certain areas of the room may be 20 F below air temperature[46] (see Fig. 9).

With an open fire, or a high temperature radiant heater the influence of radiant heat may be considerable; and with large window areas the converse cooling effect may be important. Outdoors in the sun, the influence of radiation is, of course, enormous. Where substantial differences between air and surrounding surfaces (or special radiant heat sources) do exist, this factor must be taken into account.

An interesting point indicated by the studies at the *John B. Pierce Laboratory*, is that when a given operative temperature (that is, the temperature which should physically exert a certain heat-demand upon a body of fixed surface temperature) is produced by (1) air and walls at approximately the same temperature, and (2) colder air and warmer walls (in the zone of body cooling), the skin temperature falls to a lower point for the cold air warm wall situation, thus decreasing the actual rate of heat loss. The reason for this effect is somewhat obscure but it is believed to be related to local stimulation by the colder air exerted on the membranes of the nose and throat and to the greater chilling of the exposed skin surfaces when those parts of the body are moved.

---

[46]Wall Surface Temperatures, by A. C. Willard and A. P. Kratz (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 447).

INFLUENCE OF HUMIDITY

Recent research indicates that, from the physiological viewpoint:

1. In the *hot zone* (above a point of ideal adjustment and comfort) high relative humidity tends to prevent evaporative regulation and is exceedingly harmful. Even when evaporation of sweat maintains a successful thermal balance, the process is accompanied by marked discomfort and interference with physical efficiency; and high relative humidity sharply narrows the zone of temperature within which adjustment can take place, a rise from 0 to 100 per cent relative humidity (under conditions cited in a preceding paragraph) lowering the limit of air-temperature tolerance by 40 deg (see Fig. 2 and Table 2 giving upper limits of evaporative regulation).

2. In the *cold zone* relative humidity has comparatively slight influence, lowering relative humidity from 75-80 per cent down to 30-35 per cent increasing heat loss only by the same amount as a 1 deg fall in temperature.

For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth[47] until the infants reach a weight of about 5 lb. No such clear-cut evidence exists in the case of adults. In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the *Harvard School of Public Health*, the majority of the subjects were unable to detect sensations of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 per cent and 60 per cent with ordinary room temperatures which is in accord with other studies[48,49].

The limitation of the comfort zones in Fig. 6 with respect to humidity must not be taken too seriously. Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort. In mild weather comparatively high relative humidities are entirely feasible, but in cold weather they are objectionable on account of condensation and frosting on the windows. Information on this subject is given in Chapter 3.

A degree of atmospheric humidity sufficiently high to cause deposition of moisture in the clothing may perhaps increase the chilling effects of cold air; but little or no exact information is available on this point. The dividing line at which humidity has no effect upon warmth varies with the air velocity and is about 46 F (dry-bulb) for still air and about 50, 56 and 60 F for air velocities of 100, 300 and 500 fpm, respectively.

As to the effects of dryness of the air, *per se*, and irrespective of thermal effects, there is a common belief that dry air in itself exerts a harmful effect upon the skin and mucous membranes; but there is no convincing evidence that the increase of atmospheric moisture which can practically be introduced by humidification into the air of cool occupied rooms has any effect upon health and comfort. All controlled experiments on this point have yielded negative results; and the respiratory membranes of industrial workers exposed to hot moist air are distinctly more abnormal than those of workers exposed to hot dry air[50].

---

[47]Loc. Cit. Note 34.

[48]Humidity and Comfort, by W. H. Howell (*The Science Press*, April, 1931).

[49]Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal of Hygiene*, Vol. 13, 1931, p. 432).

[50]Loc. Cit. Note 12.

INFLUENCE OF AIR MOVEMENT

The problem of the influence of air movement is a highly complex one as illustrated in Fig. 10, where heat losses per unit of body surface by radiation, convection and evaporation are plotted against air temperature for three different rates of air movement (with 50 per cent relative humidity). It will be noted that:

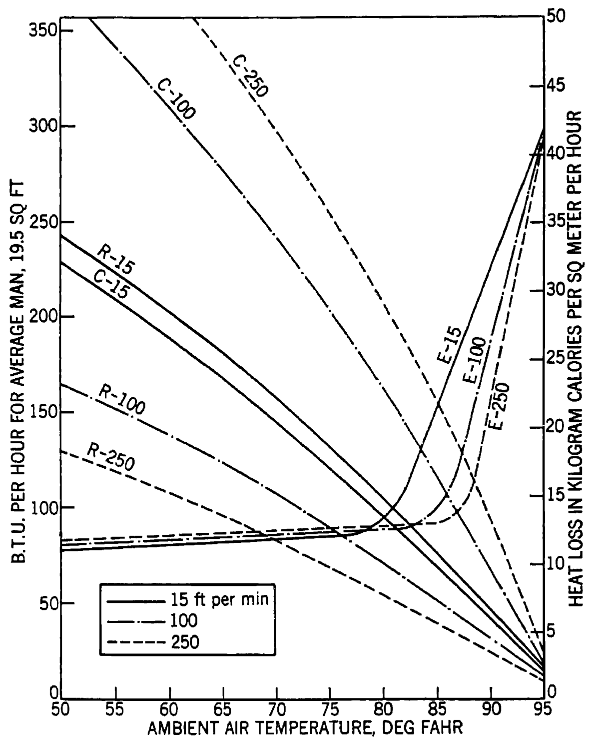


FIG. 10. CHANGES IN DISTRIBUTION OF HEAT LOSS BY RADIATION, CONVECTION, AND EVAPORATION AT VARIOUS AIR TEMPERATURES WITH THREE DIFFERENT AIR MOVEMENTS

1. Evaporative heat loss is constant and minimal at air temperatures below 80 F and is relatively uninfluenced in this area by air movement. With low air movement, sweat secretion begins to rise at 80 F, but with very high air movement only the rise does not begin until 87 F (because, with high air movement, the body cools more readily by convection and hence the sweat secreting mechanism need not operate at so low an air temperature). Above this critical point, evaporation increases very sharply with increasing air temperature.

2. Heat loss by radiation is *decreased* as air movement increases because the greater influence of convection, when air movement is high, lowers the skin temperature and thus lowers radiation which depends on the differential between walls and body surface.

3. Convection rises at all points sharply with increased air movement, according to a relation discussed in a succeeding paragraph.

It will be noted that the proportionate heat loss by the three processes involved varies widely, as indicated in Table 6.

At air temperatures above the temperatures of the body surfaces, the body will, of course, be gaining heat by convection and losing heat only by evaporation. Increased air movement will favor both these processes and its net effect will depend on the relative humidity of the atmosphere. The phenomena involved are illustrated in Fig. 11 which shows the influence of air movement (at varying air temperatures and humidities) upon the upper limit of the *zone of evaporative regulation*. It will be noted that (for the nude subject in a semi-reclining posture) increase in air movement consistently increases evaporative cooling, and therefore heat tolerance, when relative humidity is high and air temperature low. When

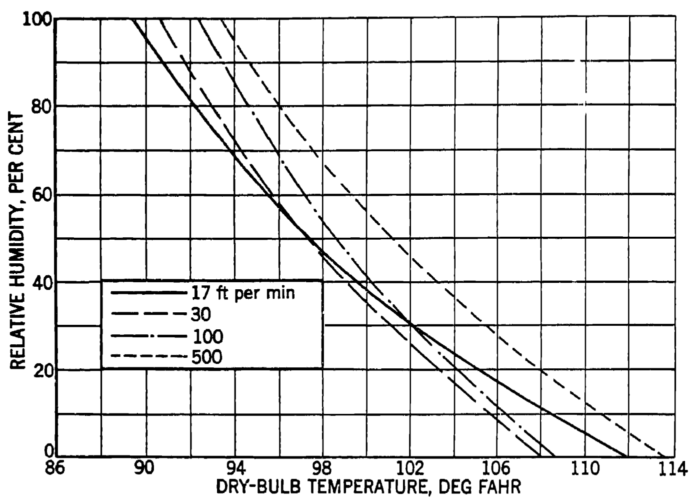


FIG. 11. CONTOUR CHART INDICATING UPPER LIMITS (WETTED AREA = 100 PER CENT) OF THE ZONE OF EVAPORATIVE REGULATION FOR VARIOUS AIR VELOCITIES FOR UNCLOTHED SUBJECTS

relative humidity is low and air temperature is high, however, an increase in air velocity from 17 to 30 or 100 fpm actually decreases heat tolerance. Under these conditions air temperature is above skin temperature and the increased demand for evaporative cooling exceeds the actual increase of evaporation due to the higher air movement. When an air velocity of 500 fpm is reached, the limits of the zone are broadened throughout its range. For very hot and dry environments, still air is more desirable than a slightly greater air movement but a high air movement is still better.

At low air temperatures the effect of increased air movement upon convection loss is a simple and direct one; but the exact quantitative influence of air movement upon the rate of cooling of a hot body has, until recently, been unknown. Studies at the *John B. Pierce Laboratory of Hygiene* have indicated that the cooling effect of air actually increases as

TABLE 6. PERCENTAGE OF TOTAL HEAT LOSS EFFECTED BY THE THREE CHIEF ROUTES OF THERMAL INTERCHANGE

Air Temperature	Deg F	60			70			80			90		
Air Movement	fpm	15	100	250	15	100	250	15	100	250	15	100	250
Evaporation	per cent	18	16	15	22	19	18	29	27	25	73	67	55
Radiation	per cent	43	26	19	40	26	17	37	24	17	14	10	13
Convection	per cent	39	58	66	38	55	65	34	49	58	13	23	32

the square root of its velocity[51]. Under the conditions of the experiments in question (semi-reclining lightly clothed subjects) the effect is represented by the formula:

$$\frac{C}{\Delta T} = 0.413\sqrt{V}, \tag{2}$$

where

C = Convection loss in kilogram-calories per square meter of body surface per hour.

ΔT = Surface temperature of the body (clothing and exposed skin) minus air temperature, degrees Fahrenheit.

V = Velocity of air, feet per minute.

The extent of the effect may be indicated by the computed data in Table 7.

As emphasized in an earlier paragraph the problem of local drafts causing differential cooling of special areas of the body is one that must always be kept in mind. Experience and recent field studies by the A.S.H.V.E. Research Laboratory[52] place the desirable air movement between 15 and 25 fpm under ordinary room temperatures during the

TABLE 7. CORRESPONDING EQUIVALENT AIR TEMPERATURES PRODUCING EQUAL HEAT LOSS AT 15 FPM FOR VARIOUS AIR TEMPERATURES AND AIR MOVEMENTS

Observed Ambient Air Temperature Deg F	Equivalent temperature at standard air movement of 15 fpm when observed air movement is			
	15 fpm	100 fpm	250 fpm	500 fpm
95	95	95.0	95.7	96.0
90	90	88.5	87.7	87.0
85	85	81.7	79.5	78.0
80	80	75.2	71.6	69.0
75	75	68.7	63.5	60.3
70	70	62.2	55.5	51.2
65	65	55.5	47.5	42.5
60	60	48.9	39.5	33.6
55	55	42.3	31.8	24.7
50	50	35.7	23.5	15.8

[51]The Influence of Air Movement on Heat Losses for the Clothed Human Body, by C.-E. A. Winslow, A. P. Gagge and L. P. Herrington (*American Journal of Physiology*, October, 1939, Vol. 127, p. 505).

[52]A.S.H.V.E. RESEARCH REPORT No. 1016—Classroom Drafts in Relation to Entering Air Stream Temperature, by F. C. Houghten, H. H. Trimble, Carl Gutberlet, and M. F. Lichtenfels (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 268).

heating season. Objectionable drafts are likely to occur when the velocity of the air current is 40 fpm and the temperature of the air current 2 F or more below the customary winter room temperature. Higher velocities are desirable in the summer time when the air temperature exceeds 80 F. Variations in air movement and temperature in different parts of occupied rooms are often indicative of relative air distribution. The work of the A.S.H.V.E. Research Laboratory indicates that an air movement between 15 and 25 fpm with a temperature variation of 3 F or less in different parts of a room, 36 in. above floor, represent satisfactory distribution.

THE FOUR VITAL FACTORS

From the preceding discussion it is clear that thermal environment cannot properly be adjusted to the requirements of human health and comfort without control of all the four basic factors concerned.

According to the recommendations of the Sub-Committee on the Hygiene of Environmental Conditions in the Dwelling[53], it is of great importance in all research studies to make an accurate record of each of the four independent factors governing bodily heat exchanges, temperature, movement and humidity of the air, and mean radiant temperature of the surrounding surfaces. For this purpose the committee suggested in the interest of comparability the use of the following four types of instruments or others yielding similar data:

1. Silvered dry-bulb thermometers or hair-pin thermometers (Bargeboer).
2. Silvered dry Kata-thermometers or the hot wire anemometer.
3. Psychrometer, wet- and dry-bulb, whirling or ventilated.
4. Globe thermometer (Vernon) or the dry resultant thermometer (Missenard).

Such instruments as these, when properly calibrated and their readings are compared, can be used for determining the four basic physical factors concerned separately or in certain combinations. The results of the four physical measurements thus determined can generally be translated into the terms of any special instrument combining two or more of them.

In some instances it may be important to record not only the movement and temperature of the air at various levels, but also the temperature of each wall and window, of the flooring, and of the ceiling, and to measure the total effective radiation of the surroundings in 6 directions; in order to trace the exact causes of defects in the building which have an unfavorable influence on the heat exchanges of its inhabitants. Facts of this type are of great practical importance.

In all fundamental studies of air conditions records should be obtained, therefore, showing:

1. True air temperature (free from radiation effects).
2. Air movement.
3. Humidity.
4. Mean radiant temperature of surrounding surfaces.

In interpreting the results of such studies, the clothing and the physical activity of the subjects are of primary importance.

---

[53]Housing Commission of the League of Nations, adopted at Geneva, June 25, 1937.

Chapter 3

HEAT TRANSMISSION COEFFICIENTS AND TABLES

Methods of Heat Transfer, Coefficients, Conductivity of Homogeneous Materials, Surface Conductance Coefficients, Air Space Conductance, Practical Coefficients, Table of Conductivities and Conductances, Tables of Over-all Coefficients of Heat Transfer for Typical Building Construction, Combined Coefficients of Transmission

IN order to maintain comfortable living temperatures within a building it is necessary to supply heat at the same rate that it is lost from the building. The loss of heat occurs in two ways, by direct transmission through the various parts of the structure and by air leakage or filtration between the inside and outside of the building. The purpose of this chapter is to show methods of calculation and to give practical transmission coefficients which may be applied to various structures to determine the heat loss by direct transmission. The amount lost by air filtration is determined by different methods, as outlined in Chapter 4, and must be added to that lost by direct transmission to obtain the total heating plant requirements.

METHODS OF HEAT TRANSFER

Heat transmission between the air on the two sides of a structure takes place by three methods, namely, radiation, convection and conduction. In a simple wall built up of two layers of homogeneous materials separated to give an air space between them, heat will be received from the high temperature surface by radiation, convection and conduction. It will then be conducted through the homogeneous interior section by conduction and carried across to the opposite surface of the air space by radiation, conduction and convection. From here it will be carried by conduction through to the outer surface and leave the outer surface by radiation, convection and conduction. The process of heat transfer through a built-up wall section is complicated in theory, but in practice it is simplified by dividing a wall into its component parts and considering the transmission through each part separately. Thus the average wall may be divided into external surfaces, homogeneous materials and interior air spaces. Practical heat transmission coefficients may be derived which will give the total heat transferred by radiation, conduction and convection through any of these component parts and if the selection and method of applying these individual coefficients is thoroughly understood it is usually a comparatively simple matter to calculate the over-all heat transmission coefficient for any combination of materials.

HEAT TRANSFER COEFFICIENTS

The symbols representing the various coefficients of heat transmission and their definitions are as follows:

U = thermal transmittance or over-all coefficient of heat transmission; the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

k = thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C = thermal conductance; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a non-homogeneous material for the thickness or type under consideration for a difference in temperature of 1 F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plasterboard and hollow clay tile.

f = film or surface conductance; the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = thermal conductance of an air space; the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of an air space for a temperature difference of 1 F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, *i.e.*:

$$\frac{1}{U} = \text{over-all or air-to-air resistance.}$$

$$\frac{1}{k} = \text{internal resistivity.}$$

$$\frac{1}{C} = \text{internal resistance.}$$

$$\frac{1}{f} = \text{film or surface resistance.}$$

$$\frac{1}{a} = \text{air space resistance.}$$

As an example in the application of these coefficients assume a wall with over-all coefficient U. Then,

$$H = A\,U\,(t - t_o) \tag{1}$$

where

H = Btu per hour transmitted through the material of the wall, glass, roof or floor.

A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured. (Use the net inside or heated surface dimensions in all cases.)

$t - t_o$ = temperature difference between inside and outside air, in which t must always be taken at the proper level. Note that t may not be the *breathing-line* temperature in all cases.

If the heat transfer between the air and the inside surface of the wall is being considered, then,

$$H = A f_i (t - t_1) \tag{2}$$

where

f_i = inside surface conductance.
t and t_1 = the temperatures of the inside air and the inside surface of the wall respectively.

In practice it is usually the over-all heat transmission coefficient that is required. This may be determined by a test of the complete wall, or it may be obtained from the individual coefficients by calculation. The simplest method of combining the coefficients for the individual parts of the wall is to use the reciprocals of the coefficients and treat them as resistance units. The total over-all resistance of a wall is equal numerically to the sum of the resistances of the various parts, and the reciprocal of the over-all resistance is likewise the over-all heat transmission coefficient of the wall. For a wall built up of a single homogeneous material of conductivity k and x inches thick the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} \tag{3}$$

If the coefficients f_i, f_o and k, together with the thickness of the material x are known, the over-all coefficient U may be readily calculated as the reciprocal of the total heat resistance.

For a compound wall built up of three homogeneous materials having conductivities k_1, k_2 and k_3 and thicknesses x_1, x_2 and x_3 respectively, and laid together without air spaces, the total resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} \tag{4}$$

For a wall with air space construction consisting of two homogeneous materials of thicknesses x_1 and x_2, and conductivities k_1 and k_2, respectively, separated to form an air space of conductance a, the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \tag{5}$$

Likewise any combination of homogeneous materials and air spaces can be put into the wall and the over-all resistance of the combination may be calculated by adding the resistances of the individual sections of the wall. In certain special forms of construction such as tile with irregular air spaces it is necessary to consider the conductance C of the unit as built instead of the unit conductivity k, and the resistance of the section is equal to $\frac{1}{C}$. The method of calculating the over-all heat transmission coefficient for a given wall is comparatively simple, but the selection of the proper coefficients is often complicated. In some cases the construction of the wall is such that the substituting of coefficients in the accepted formula will give erroneous results. This is the case with irregular cored

out air spaces in concrete and tile blocks, and walls in which there are parallel paths for heat flow through materials having different heat resistances. In such cases it is necessary to resort to test methods to check the calculations, and in practically all cases it has been necessary to determine fundamental coefficients by test methods.

Conductivity of Homogeneous Materials

The thermal conductivity of homogeneous materials is affected by several factors. Among these are the density of the material, the amount of moisture present, the mean temperature at which the coefficient is determined, and for fiberous materials the arrangement of fiber in the material. There are many fiberous materials used in building construction and considered as homogeneous for the purpose of calculation, whereas they are not really homogeneous but are merely considered so as a matter of convenience. In general, the thermal conductivity of a material increases directly with the density of the material, increases with the amount of moisture present, and increases with the mean temperature at which the coefficient is determined. The rate of increase for these various factors is not the same for all materials, and in assigning proper coefficients one should make certain that they apply for the conditions under which the material is to be used in a wall. Failure to do this may result in serious errors in the final coefficients.

Surface Conductance Coefficients

Heat is transmitted to or from the surface of a wall by a combination of radiation, convection and conduction. The coefficient will be affected by any factor which has an influence on any one of these three methods of transfer. The amount of heat by radiation is controlled by the character of the surface and the temperature difference between it and the surrounding objects. The amount of heat by conduction and convection is controlled largely by the roughness of the surface, by the air movement over the surface and by the temperature difference between the air and the surface. Because of these variables the surface coefficients may be subject to wide fluctuations for different materials and different conditions. The inside and outside coefficients f_i and f_o are in general affected to the same extent by these various factors and test coefficients determined for inside surfaces will apply equally well to outside surfaces under like conditions. Values for f_i in still and moving air at different mean temperatures have been determined for various building materials[1].

The relation obtained between surface conductances for different materials at mean temperatures of 20 F is shown in Fig. 1. These values were obtained with air flow parallel to the surface and from other tests in which the angle of incidence between the direction of air flow and the surface was varied from zero to 90 F it would appear that these values might be lowered approximately 15 per cent for average conditions. While for average building materials there is a difference due to mean temperature, the greatest variation in these coefficients is caused by the character of the surface and the wind velocity. If other surfaces, such as aluminum foil with low emissivity coefficients were substituted, a large

[1]A.S.H.V.E. RESEARCH REPORT No. 869—Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429).

part of the radiant heat would be eliminated. This would reduce the total coefficient for all wind velocities by about 0.7 Btu and would make but very little difference for the higher wind velocities. In many cases in building construction the heat resistance of the internal parts of the wall is high as compared with the surface resistance and the surface factors become of small importance. In other cases such as single glass windows the surface resistances constitute practically the entire resistance of the structure, and therefore become important factors. Due to the wide variation in surface coefficients for different conditions their selection for a practical building becomes a matter of judgment. In calculating the

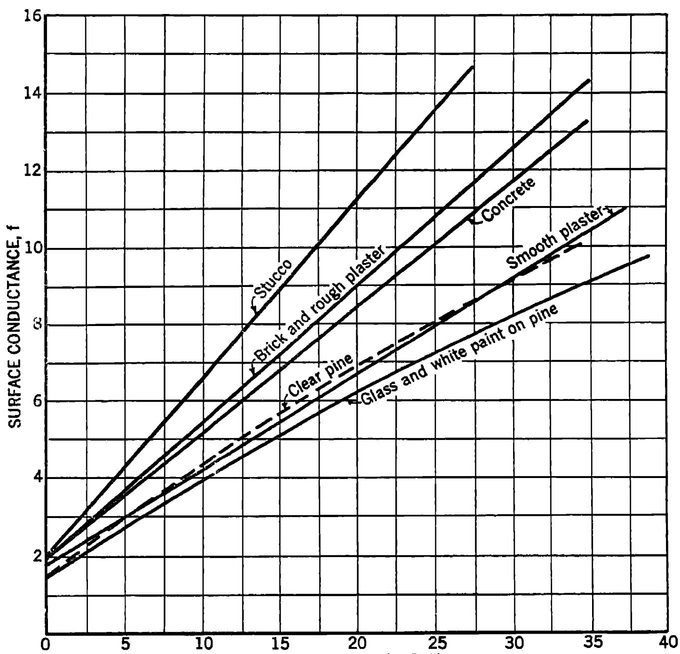


FIG. 1. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE

over-all coefficients for the walls of Tables 3 to 12, 1.65 has been selected as an average inside coefficient and 6.0 as an average outside coefficient for a 15-mile wind velocity. In special cases where surface coefficients become important factors in the over-all rate of heat transfer more selective coefficients may be required.

The surface conductance values given in Table 1, **Section A** are based on recent tests and are for still air conditions and emissivities of 0.83 and 0.05, respectively.

Air Space Conductance

Heat is conducted across an air space by a combination of radiation, conduction and convection. The amount of heat by radiation is governed

Table 1. Conductances (C) for Surfaces and Air Spaces

All conductance values expressed in Btu per hour per square foot per degree Fahrenheit temperature difference.

Section A. Surface Conductances for Still Air[a]

Position of Surface	Direction of Heat Flow	Surface Emissivity $e = 0.83$	Surface Emissivity $e = 0.05$
Horizontal	Upward	1.95	1.16
Horizontal	Downward	1.21	0.44
Vertical		1.52	0.74

Section B. Conductance of Vertical Spaces at Various Mean Temperatures[b]

Mean Temp Deg Fahr	Conductances of Air Spaces for Various Widths in Inches						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

Section C. Conductances and Resistances of Air Spaces Faced with Reflective Insulation[c]

Location and Position of Air Space	Direction of Heat Flow	Temp[d] Diff Deg Fahr		Conductance[e] (C)			Resistance[e] $\left(\frac{1}{C}\right)$		
		Winter	Summer	No. of Air Spaces 1	2	3	No. of Air Spaces 1	2	3
Rafter Space (8 in.)									
Horizontal	Down	45			0.10	0.07		10.00	14.29
Horizontal	Up	45			0.27	0.17		3.70	5.88
Horizontal	Down		25		0.09	0.06		11.11	16.67
Horizontal	Up		25		0.24	0.16		4.17	6.25
30 deg slope	Down	45			0.15	0.10		6.67	10.00
30 deg slope	Up	45			0.25	0.17		4.00	5.88
30 deg slope	Down		25		0.13	0.09		7.69	11.11
30 deg slope	Up		25		0.23	0.14		4.35	7.14
Stud Space (3⅝ in.)									
Vertical[f]		30		0.34			2.94		
Vertical		40			0.23	0.13		4.35	7.69
Vertical[f]			15	0.32			3.13		
Vertical			20		0.18	0.11		5.56	9.09
Vertical[g]		30		0.46			2.17		

[a]Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 513).

[b]A.S.H.V.E. Research Report No. 825—Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. Transactions, Vol. 35, 1939, p. 165).

[c]Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E. R. Queer (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1940, p. 68).

[d]Temperature difference is based on total space between plaster base and sheathing, flooring or roofing.

[e]These air space conductance and resistance values are based on one reflective surface (aluminum) having an emissivity of 0.05 facing each space and are based on total space between plaster base and sheathing, flooring or roofing. The rafter and stud spaces are divided into equal spaces.

[f]Stud space is lined on plaster base side with loose paper with aluminum on surface facing air space. The resistance of the small air space between the plaster base and paper was 0.43.

[g]Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 351).

largely by the nature of the surface and the temperature difference between the boundary surfaces of the air space. Conduction and convection are controlled largely by the width and shape of the air space and the roughness of the boundary surfaces.

The conductances of vertical air spaces bounded by such materials as paper, wood, plaster, etc., are given in Table 1, **Section B,** having emissivity coefficients of 0.8 or higher, and with extended parallel surfaces perpendicular to the direction of heat flow. A conductance of 1.10 Btu per hour per square foot per degree Fahrenheit temperature difference (resistance = 0.91) based on this table was used for calculating the overall coefficients given in Tables 3 to 12 inclusive for air spaces ¾ in. or more in width. Air space tests[2] reported by Wilkes and Peterson resulted in comparable values. For 3⅝ in. horizontal air spaces having an effective emissivity of 0.83, the conductance for heat flow upward was 1.32 and for heat flow downward, 0.94. The conductance for a similar vertical air space was 1.17, the resistances of course being the reciprocals of these values in each case.

A large part of the heat transferred across air spaces bounded by ordinary materials is by radiation. Therefore, if such air spaces are faced with metallic surfaces such as aluminum foil, coated sheet steel or other low-emissivity, infra-red reflective metal surfaces, the radiant heat transfer will be substantially reduced, thus causing the major portion of the remaining transmitted heat to be by convection. Table 1, **Section C,** gives conductances and resistances for air spaces bounded by one reflective surface having an emissivity of 0.05. It will be noted that the conductance values given in this table are a function of the temperature differences across the space rather than mean temperature, the larger the temperature difference, the larger the conductance. The radiant heat transfer is the same regardless of whether the low emissivity surface is on the high or low temperature surface of the space, and is independent of the width of the space. To minimize the convection transfer the vertical air space should be at least ¾ in. in width. A conductance of 0.46 was used for computing the overall coefficients in Tables 3 to 12 inclusive for air spaces bounded by aluminum foil applied to plasterboard.

When referring to reflective heat-insulating surfaces, the term *brightness* which deals with visible light has no specific meaning and should be avoided[3]. *Emissivity* and *reflectivity* definitely define the radiating and reflecting properties and values may be determined directly for long wavelength radiation corresponding to room temperature. As previously stated, the values in Table 1, **Section C,** are based on an emissivity of the reflective surface of 0.05. Obviously for higher emissivity values the conductances will increase accordingly. For example, non-metallic reflective materials are available having emissivity values approximately midway between those of metallic reflective insulations and ordinary building material surfaces. These materials will have a correspondingly higher radiant heat transfer and where such materials are under consideration, due allowance should be made for the higher emissivity value in arriving at the proper air space conductance.

[2]Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 351).

[3]Some Reflection and Radiation Characteristics of Aluminum, by C. S. Taylor and J. D. Edwards (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 179).

Where reflective insulating materials are involved the possible increase in the emissivity coefficient due to surface coatings or chemical action[4] should be studied by the engineer in order to satisfy himself as to the permanence of the reflective surface for the conditions under which this material will be used. In making installations of this material the partitions between air spaces should be tight, particularly at the top and bottom so that air cannot circulate between adjacent spaces.

When reflective insulating materials are installed with multiple air spaces, the position (vertical, horizontal or inclined) of the material in the structure must be taken into consideration. For example, the resistance to heat flow upward is about one-third that of downward flow in a horizontal position in the same construction, as will be apparent from Table 1, **Section C.** However, the difference between upward heat flow through single horizontal or sloping air spaces and through single vertical air spaces is comparatively small for the same temperature difference. Consequently the same conductance value (0.46) was used for computing the coefficients in Tables 8 and 12, involving horizontal and sloping air spaces bounded on one side by aluminum foil applied to plasterboard, as for similar vertical air spaces in Tables 3, 4, 5 and 6.

As already stated, a conductance value of 1.10 was similarly used in all cases for calculating the coefficients of construction involving vertical, horizontal and sloping air spaces bounded on both sides by ordinary building materials.

PRACTICAL COEFFICIENTS

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making tests on the individual material or combination of materials. In Table 2 coefficients are given for a group of materials which have been selected from various sources. Wherever possible the properties of material and conditions of tests are given. However, in selecting and applying these values to any construction a reasonable amount of caution is necessary; variations will be found in the coefficients for the same materials, which may be partly due to different test methods used, but which are largely due to variations in materials. The recommended coefficients which have been used for the calculation of over-all coefficients as given in Tables 3 to 12 are marked by an asterisk.

It should be recognized in these tables of calculated coefficients that space limitations will not permit the inclusion of all the combinations of materials that are used in building construction and the varied applications of insulating materials to these constructions. Typical examples are given of combinations frequently used, but any special construction not given in Tables 3 to 12 can generally be computed by using the conductivity values given in Table 2 and the fundamental heat transfer formulae. For example, the tabulation of all of the values for multiple layers of insulating materials would present extensive and detailed problems of calculations for the varied application combinations, but the engineer having the fundamental conductivity values can quickly obtain the proper coefficients.

[4]Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. C. Hechler and E. R. Queer (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1940, p. 68).

Attention is called to the fact that the conductivity values per inch of thickness do not afford a true basis for comparison between insulating materials as applied, although they are frequently used for that purpose. The value of an insulating material is measured in terms of the coefficient (U_i) of the insulated construction as compared to the coefficient (U) of the construction without insulation. Certain types of blanket installations are designed to be installed between the studs of a frame building in such manner as to give two air spaces. In order to get the full value of such materials they should be so installed that each air space is approximately 1 in. or more in thickness and the air spaces should be sealed at the top and bottom to prevent the circulation of air from one space to the other. Another common error in installing such a material is to nail the blanket on the outside of the studs underneath the sheathing, in which case one air space is lost and also the thickness of the insulating material is materially reduced at the studs. There are certain other types of insulation which are very porous, allowing air circulation within the material if not properly installed. The architect or engineer must carefully evaluate the economic considerations involved in the selection of an insulating material as adapted to various building constructions. Lack of good judgment in the intelligent choice of an insulating material, or its improper installation, frequently represents the difference between good or unsatisfactory results.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 3 to 13, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and ½ in. of plaster is 0.46, and the number assigned to a wall of this construction is *1-B*, Table 3.

Example 1. Calculate the coefficient of transmission (U) of an 8-in. brick wall with ½ in. of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph. It is assumed that the outside course is of hard (high density) brick having a conductivity of 9.20, and that the inside course is of common (low density) brick having a conductivity of 5.0, the thicknesses each being 4 in. The conductivity of the plaster is assumed to be 3.3, and the inside and outside surface coefficients are assumed to average 1.65 and 6.00, respectively, for still air and a 15 mph wind velocity.

Solution. k (hard high density brick) = 9.20; x = 4.0 in.; k (common low density brick) = 5.0; x = 4.0 in.; k (plaster) = 3.3; x = ½ in.; f_i = 1.65; f_o = 6.0. Therefore,

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}}$$

$$= \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

= 0.46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

The coefficients in the tables were determined by calculations similar to those shown in Example 1, using Fundamental Formulae 3, 4 and 5 and the values of k (or C), f_i, f_o and a indicated in Table 2 by asterisks. In computing heat transmission coefficients of floors laid directly on the ground (Table 10), only one surface coefficient (f_i) is used. For example,

the value of U for a 1-in. yellow pine floor (actual thickness, 25/32 in.) placed directly on 6-in. concrete on the ground, is determined as follows:

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.781}{0.80} + \frac{6.0}{12.0}} = 0.48 \text{ Btu per hour per square foot per degree difference}$$

in temperature between the ground and the air immediately above the floor.

Rigid insulation refers to so-called insulating board which may be used structurally, such as for sheathing. Flexible insulation refers to the blankets, quilts or semi-rigid types of insulation.

Actual thicknesses of lumber are used in the computations rather than nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as it is probable that the values of U for these two types of roofs will compare favorably.

The thicknesses upon which the coefficients in Tables 3 to 13 inclusive, are based are as follows:

Brick veneer	4 in.
Plaster and metal lath	3/4 in.
Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard)	1/2 in.
Slate (roofing)	1/2 in.
Stucco on wire mesh reinforcing	1 in.
Tar and gravel or slag-surfaced built-up roofing	3/8 in.
1-in. lumber (S 2-S)	25/32 in.
1½-in. lumber (S-2-S)	1 5/16 in.
2-in. lumber (S-2-S)	1 5/8 in.
2½-in. lumber (S-2-S)	2 1/8 in.
3-in. lumber (S-2-S)	2 5/8 in.
4-in. lumber (S-2-S)	3 5/8 in.
Finish flooring (maple or oak)	13/16 in.

Solid brick walls are based on 4 in. hard brick (high density) and the remainder common brick (low density). Stucco is assumed to be 1 in. thick on masonry walls. Where metal lath and plaster are specified, the metal lath is neglected.

The coefficients of transmission of the pitched roofs in Table 12 apply where the roof is over a heated attic or top floor so the heat passes directly through the roof structure including whatever finish is applied to the underside of the roof rafters.

It is the practice of many engineers in calculating heat losses to allow for possible defects in workmanship, poor construction and other factors which would diminish the efficiency of the insulation. The lower the theoretical wall or roof coefficient the greater will be the percentage of error due to construction defects.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

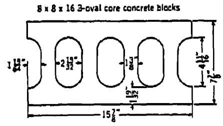
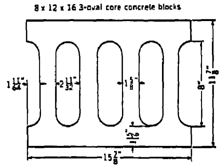
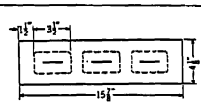
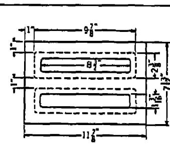
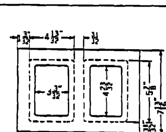
Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS						
Brick	Low density	—	—	5.00*	0.20	—
	High density	—	—	9.20*	0.11	—
	Adobe	—	—	3.56*	0.28	—
Brickwork	Damp or wet	—	—	5.00[c]	0.20	(2)
Cement Mortar	Typical	—	—	12.00*	0.08	—
Concrete	Typical	—	—	12.00*	0.08	—
	Various ages and mixes[d]	—	—	11.35 *to* 16.36	—	(5)
	Cellular	40.0	75	1.06	0.94	(3)
	Cellular	50.0	75	1.44	0.69	(3)
	Cellular	60.0	75	1.80	0.56	(3)
	Cellular	70.0	75	2.18	0.46	(3)
	Typical fiber gypsum, 87.5% gypsum and 12.5% wood chips	51.2	74	1.66*	0.60	(4)
	Special concrete made with an aggregate of hardened clay—1-2-3 mix	101.0	70	3.98	0.25	(3)
	Sand and gravel	142.0	75	12.6	0.08	(4)
	Limestone	132.0	75	10.8	0.09	(4)
	Cinder	97.0	75	4.9	0.22	(4)
	Burned clay aggregate	75.0	75	4.0	0.25	(4)
	Blast furnace slag aggregate	76.0	70	1.6	0.63	(3)
	Expanded vermiculite aggregate	20	90	0.68	1 47	(3)
	Expanded vermiculite aggregate	26.7	90	0 76	1 32	(3)
	Expanded vermiculite aggregate	35	90	0 86	1.16	(3)
	Expanded vermiculite aggregate	50	90	1 10	0.91	(3)
Stone	Typical	—	—	12.50*	0.08	—
Stucco	Typical	—	—	12.00*	0.08	—
Tile	Typical hollow clay (4 in.)	—	—	1.00†*	1.00	—
	Typical hollow clay (6 in.)[e]	—	—	0.64†*	1.57	—
	Typical hollow clay (8 in.)[e]	—	—	0.60†*	1.67	—
	Typical hollow clay (10 in.)[e]	—	—	0.58†*	1.72	—
	Typical hollow clay (12 in.)[e]	—	—	0.40†*	2.50	—
	Typical hollow clay (16 in.)[e]	—	—	0.31†*	3.23	—
	Hollow clay (2 in.) ½-in. plaster both sides	120.0	110	1.00†	1.00	(2)
	Hollow clay (4 in.) ½-in. plaster both sides	127.0	100	0.60†	1.67	(2)
	Hollow clay (6 in.) ½-in. plaster both sides	124.3	105	0.47†	2.13	(2)
	Hollow gypsum (4 in.)	—	—	0.46†*	2.18	—
	Solid gypsum	51.8	70	1.66	0.60	(4)
	Solid gypsum	75.6	76	2.96	0.34	(4)
Tile or Terrazzo	Typical flooring	—	—	12.00*	0.08	—

AUTHORITIES:

[1]U. S. Bureau of Standards, tests based on samples submitted by manufacturers.
[2]A. C. Willard, L. C. Lichty, and L. A. Harding, tests conducted at the University of Illinois.
[3]J. C. Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers.
[4]F. B. Rowley, tests conducted at the University of Minnesota.
[5]A.S.H.V.E. Research Laboratory.
[6]E. A. Allcut, tests conducted at the University of Toronto.
[7]Lees and Charlton.
*Recommended conductivities and conductances for computing heat transmission coefficients.
†For thickness stated or used on construction, not per 1 in. thickness.
[a]For additional conductivity data see *A.S.R.E. Data Book*.
[b]If outside surface of block is painted with an impervious coat of paint, add 0.07 to resistance for sand and gravel blocks. Add 0.18 to resistance for cinder blocks. Add 0.17 to resistance for burned clay aggregate blocks.
[c]Recommended value. See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.
[d]See A.S.H.V.E. RESEARCH REPORT No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).
[e]The 6-in., 8-in., and 10-in. hollow tile figures are based on two cells in the direction of heat flow. The 12-in. hollow tile is based on three cells in the direction of heat flow. The 16-in. hollow tile consists of one 10-in. and one 6-in. tile, each having two cells in the direction of heat flow.
[f]Not compressed.
[g]Roofing, 0.15-in. thick (1.34 lb per sq ft), covered with gravel (0.83 lb per sq ft), combined thickness assumed 0.25.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description: Cement	Description: Fine Aggregate 0-No. 4	Description: Coarse Aggregate No. 4-½	Description: Slump	Description: Per Cent	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS —Continued										
Steam Treated Limestone Slag	1	7.00			27.1	74.6	74.49	2.27	0.44	(4)
Pumice Mined in Calif.	1	8.00			26.5	65.0	74.68	2.42	0.41	(4)
By-Product of Manufacture of Phosphates	1	8.00	Fineness Modulus 3.75		25.5	86.6	74.62	3.19	0.31	(4)
	1	8.00			21.1	91.1	74.43	3.42	0.29	(4)
Expanded Burned Clay	1	8.00			18.4	57.9	75.57	2.28	0.44	(4)
Burned Clay Aggregate	1	8.50			21.8	67.1	75.89	2.89	0.35	(4)
Burned Clay Aggregate	1	8.50			21.8	67.1	74.60	2.82	0.35	(4)
8 x 8 x 16 3-oval core concrete blocks 	Sand and gravel aggregate					126.4	40	0.90†	1.11	(4)
	Sand and gravel aggregate used for calculations					....	....	1.00†*	1.00	(4)
	Cores filled with 5.14 lb density cork					....	40	0.56†*	1.79	(4)
	Crushed limestone aggregate					134.3	40	0.86†*	1.16	(4)
	Cinder aggregate					86.2	40	0.58†	1.73	(4)
	Cinder aggregate used for calculations					....	....	0.60†*	1.66	(4)
	Cores filled with 69.7 lb density cinders					....	40	0.39†	2.56	(4)
	Cores filled with 5.12 lb density cork					....	40	0.25†*	4.00	(4)
	Cores filled with 14.2 lb density rock wool					....	40	0.27†*	3.70	(4)
	Burned clay aggregate					67.7	40	0.50†*	2.00	(4)
	Cores filled with 5.06 lb density cork					....	40	0.21†	4.76	(4)
	Expanded blast furnace slag aggregate, 60% fine, 40% course					....	40	0.49†	2.04	(4)
8 x 12 x 16 3-oval core concrete blocks 	Sand and gravel aggregate					124.9	40	0.78†	1.28	(4)
	Sand and gravel aggregate used for calculations					....	....	0.80†*	....	(4)
	Cinder aggregate					86.2	40	0.53†*	1.88	(4)
	Cores filled with 5.24 lb density cork					....	40	0.24†*	4.16	(4)
	Burned clay aggregate					76.7	40	0.47†	2.13	(4)
	Cores filled with 5 6 lb density cork					....	40	0.17†*	5.88	(4)
	Cinder aggregate					100.0	40	1.00†	1.00	(4)
	Double wall with 1 in. air space between					100.0	40	0.36†	2.78	(4)
	1 in. space filled with 9.97 lb density rock wool					100.0	40	0.20†	5.00	(4)
	5 x 8 x 12 block sand and gravel aggregate					133.7	40	0.38†	2.63	(4)
	5 x 8 x 12 block sand and gravel aggregate[b]					134.0	40	0.95†	1.15	(4)

For notes see page 79.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
INSULATION—BLANKET OR FLEXIBLE TYPES						
FIBER	Typical	—	—	0.27*	3.70	—
	Chemically treated wood fibers held between layers of strong paper[f]	3.62	70	0.25	4.00	(3)
	Eel grass between strong paper[f]	4.60	90	0.26	3.85	(1)
	" " " " "	3.40	90	0.25	4.00	(1)
	Flax fibers between strong paper[f]	4.90	90	0.28	3.57	(1)
	Chemically treated hog hair between kraft paper[f]	5.76	71	0.26	3.85	(3)
	Chemically treated hog hair between kraft paper and asbestos paper[f]	7.70	71	0.28	3.57	(3)
	Hair felt between layers of paper[f]	11.00	75	0.25	4.00	(3)
	Kapok between burlap or paper[f]	1.00	90	0.24	4.17	(1)
	Jute fiber[f]	6.70	75	0.25	4.00	(3)
	Ground paper between two layers, each ⅜-in. thick made up of two layers of kraft paper (sample ¾-in. thick)	12.1	75	0.40†	2.50	(4)
	Stitched and creped expanding fibrous blanket	1.50	70	0.27	3.70	(3)
	Paper and asbestos fiber with emulsified asphalt binder	4.2	94	0.28	3.57	(1)
	Cotton insulating bat	0.875	72	0.24	4.17	(3)
INSULATION—SEMI-RIGID TYPE						
FIBER	Felted cattle hair[f]	13.00	90	0.26	3.84	(1)
	" " "	11.00	90	0.26	3.84	(1)
	Felted hair and asbestos[f]	7.80	90	0.28	3.57	(1)
	75% hair and 25% jute[f]	6.30	90	0.27	3.70	(1)
	50% hair and 50% jute[f]	6.10	90	0.26	3.85	(1)
	Jute[f]	6.70	75	0.25	4.00	(3)
	Felted jute and asbestos[f]	10.00	90	0.37	2.70	(1)
	Compressed peat moss	11.00	70	0.26	3.84	(3)
INSULATION—LOOSE FILL OR BAT TYPE						
FIBER	Made from ceiba fibers[f]	1.90	75	0.23	4.35	(3)
	" " " "	1.60	75	0.24	4.17	(3)
FIBER	Fibrous material made from dolomite and silica	1.50	75	0.27	3.70	(3)
	Fibrous material made from slag	9.40	103	0.27	3.70	(1)
	Redwood bark	3.00	90	0.31	3.22	(1)
	Redwood bark	5.00	75	0.26	3.84	(3)
GLASS WOOL	Fibrous material 25 to 30 microns in diameter, made from virgin bottle glass	1.50	75	0.27	3.70	(3)
GRANULAR	Made from combined silicate of lime and alumina	4.20	72	0.24	4.17	(3)
	Expanded vermiculite	6.32	86	0.29	3.45	(3)
GYPSUM	Flaked, dry and fluffy[f]	34.00	90	0.60	1.67	(1)
	" " " "	26.00	90	0.52	1.92	(1)
	" " " "	24.00	75	0.48*	2.08	(3)
	" " " "	19.80	90	0.35	2.86	(1)
	" " " "	18.00	75	0.34	2.94	(3)
MINERAL WOOL	All forms, typical	—	—	0.27*	3.70	—
REGRANULATED CORK	About $\frac{3}{16}$-in. particles	8.10	90	0.31	3.22	(1)
ROCK WOOL	Fibrous material made from rock	21.00	90	0.30	3.33	(1)
	" " " " "	18.00	90	0.29	3.45	(1)
	" " " " "	14.00	90	0.28	3.57	(1)
	" " " " "	10.00	90	0.27*	3.70	(1)
	Rock wool with a binding agent	14.50	77	0.33	3.03	(1)
	Rock wool with flax, straw pulp, and binder	14.50	75	0.38	2.63	(3)
	Rock wool with vegetable fibers	11.50	72	0.31	3.22	(3)
SAWDUST	Various	12.00	90	0.41	2.44	(1)
SHAVINGS	Various from planer	8.80	90	0.41	2.44	(1)
	From maple, beech and birch (coarse)	13.20	90	0.36	2.78	(1)

For notes see Page 79.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
INSULATION—RIGID						
Corkboard	Typical	.....	....	0.30*	3.33	....
	No added binder	14.00	90	0.34	2.94	(1)
	" " "	10.60	90	0.30	3.33	(1)
	" " "	7.00	90	0.27	3.70	(1)
	" " "	5.40	90	0.25	4.00	(1)
	Asphaltic binder	14.50	90	0.32	3.12	(1)
Fiber	Typical	.....	....	0.33*	3.03	—
	Chemically treated hog hair covered with film of asphalt	10.00	75	0.28	3.57	(3)
	Made from corn stalks	15.00	71	0.33	3.03	(3)
	" " exploded wood fibers	17.90	78	0.32	3.12	(4)
	" " hard wood fibers	15.20	70	0.32	3.12	(3)
	Insulating plaster 9/10 in. thick applied to 3/8 in. plaster board base	54.00	75	1.07†	0.93	(3)
	Made from licorice roots	16.10	81	0.34	2.94	(3)
	" " 85% magnesia and 15% asbestos	19.30	86	0.51	1.96	(1)
	" " shredded wood and cement	24.20	72	0.46	2.17	(3)
	" " sugar cane fiber	13.50	70	0.33	3.03	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.80	70	0.30	3.33	(3)
	Made from wheat straw	17.00	68	0.33	3.03	(3)
	" " wood fiber	15.90	72	0.33	3.03	(3)
	" " " "	15.00	70	0.33	3.03	(3)
	" " " "	.....	52	0.33	3.03	(6)
	" " " "	8.50	72	0.29	3.45	(3)
	" " " "	15.20	....	0.33	3.03	(3)
	" " " "	16.90	90	0.34	2.94	(1)
INSULATION-REFLECTIVE	See Table 1, Section C					
BUILDING BOARDS						
Asbestos	Compressed cement and asbestos sheets	123.00	86	2.70	0.37	(1)
	Corrugated asbestos board	20.40	110	0.48	2.08	(2)
	Pressed asbestos mill board	60.50	86	0.84	1.19	(1)
Gypsum	Gypsum between layers of heavy paper	62.80	70	1.41	0.71	(3)
	Rigid, gypsum between layers of heavy paper (1/2 in. thick)	53.50	90	2.60†	0.38	(1)
	Gypsum mixed with sawdust between layers of heavy paper (0.39 in. thick)	60.70	90	3.60†	0.28	(1)
Plasterboard	(3/8 in.)	.....	....	3.73†*	0.27	—
	(1/2 in.)	.....	....	2.82†*	0.35	—
ROOFING CONSTRUCTION						
Roofing	Asphalt, composition or prepared	70.00	75	6.50†*	0.15	(3)
	Built up—3/8 in. thick	.....	....	3.53†*	0.28	—
	Built up, bitumen and felt, gravel or slag surfaced[g]	.....	—	1.33	0.75	(2)
	Plaster board, gypsum fiber concrete and 3-ply roof covering 2½ in. thick	52.40	76	0.58†	1.72	(4)
Shingles	Asbestos	65.00	75	6.00†*	0.17	(3)
	Asphalt	70.00	75	6.50†*	0.15	(3)
	Slate	201.00	....	10.37*	0.10	(7)
	Wood	.....	....	1.28†*	0.78	—
PLASTERING MATERIALS						
Plaster	Cement	—	—	8.00	0.13	(2)
	Gypsum, typical	.....	....	3.30*	0.30	—
	Gyysum and expanded vermiculite mix 4 to 1	39.9	75	0.85	1.18	(3)
	Thickness 3/8 in.	.....	73	8.80†	0.11	(4)
Metal Lath and Plaster	Total thickness 3/4 in.	.....	....	4.40†*	0.23	....
Wood Lath and Plaster	3/8 in. plaster. total thickness 3/4 in.	.....	70	2.50†*	0.40	(4)
BUILDING CONSTRUCTIONS						
Frame	1-in. fir sheathing and building paper	—	30	0.86†*	1.16	(4)
	1-in. fir sheathing, building paper, and yellow pine lap siding	—	20	0.50†*	2.00	(4)
	1-in. fir sheathing, building paper and stucco	—	20	0.82	1.22	(4)
	Pine lap siding and building paper—siding 4 in. wide	—	16	0.85†*	1.18	(4)
	Yellow pine lap siding	.....	....	1.28†*	0.78	....

For notes see Page 79.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]—Concluded

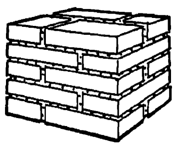
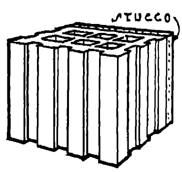
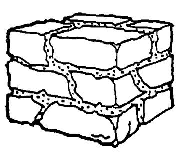
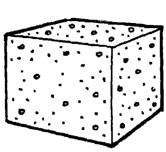
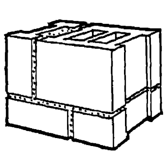
The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
BUILDING CONSTRUCTIONS —(Continued)						
Flooring	Maple—across grain	40.00	75	1.20	0.83	(3)
	Battleship linoleum (¼ in.)	—	—	1.36†*	0.74	—
WOODS (Across Grain)						
Balsa		20.0	90	0.58	1.72	(1)
		8.8	90	0.38	2.63	(1)
		7.3	90	0.33	3.03	(1)
California Redwood	0% moisture	22.0	75	0.66	1.53	(4)
	0% "	28.0	75	0.70	1.43	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	28.0	75	0.80	1.25	(4)
Cypress		28.7	86	0.67	1.49	(1)
Douglas Fir	0% moisture	26.0	75	0.61	1.64	(4)
	0% "	34.0	75	0.67	1.49	(4)
	16% "	26.0	75	0.76	1.32	(4)
	16% "	34.0	75	0.82	1.22	(4)
Eastern Hemlock	0% moisture	22.0	75	0.60	1.67	(4)
	0% "	30.0	75	0.76	1.32	(4)
	16% "	22.0	75	0.67	1.49	(4)
	16% "	30.0	75	0.85	1.18	(4)
Hard Maple	0% moisture	40.0	75	1.01	0.99	(4)
	0% "	46.0	75	1.05	0.95	(4)
	16% "	40.0	75	1.15	0.87	(4)
	16% "	46.0	75	1.21	0.83	(4)
Longleaf Yellow Pine	0% moisture	30.0	75	0.76	1.32	(4)
	0% "	40.0	75	0.86	1.16	(4)
	16% "	30.0	75	0.89	1.12	(4)
	16% "	40.0	75	1.03	0.97	(4)
Mahogany		34.3	86	0.90	1.11	(1)
Maple		44.3	86	1.10	0.91	(1)
Maple or Oak		—	—	1.15*	0.87	—
Norway Pine	0% moisture	22.0	75	0.62	1.61	(4)
	0% "	32.0	75	0.74	1.35	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.91	1.10	(4)
Red Cypress	0% moisture	22.0	75	0.67	1.49	(4)
	0% "	32.0	75	0.79	1.27	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.90	1.11	(4)
Red Oak	0% moisture	38.0	75	0.98	1.02	(4)
	0% "	48.0	75	1.18	0.85	(4)
	16% "	38.0	75	1.07	0.94	(4)
	16% "	48.0	75	1.29	0.78	(4)
Shortleaf Yellow Pine	0% moisture	26.0	75	0.74	1.35	(4)
	0% "	36.0	75	0.91	1.10	(4)
	16% "	26.0	75	0.84	1.19	(4)
	16% "	36.0	75	1.04	0.96	(4)
Soft Elm	0% moisture	28.0	75	0.73	1.37	(4)
	0% "	34.0	75	0.88	1.14	(4)
	16% "	28.0	75	0.81	1.24	(4)
	16% "	34.0	75	0.97	1.03	(4)
Soft Maple	0% moisture	36.0	75	0.89	1.12	(4)
	0% "	42.0	75	0.95	1.05	(4)
	16% "	36.0	75	1.01	0.99	(4)
	16% "	42.0	75	1.09	0.92	(4)
Sugar Pine	0% moisture	22.0	75	0.54	1.85	(4)
	0% "	28.0	75	0.64	1.56	(4)
	16% "	22.0	75	0.65	1.54	(4)
	16% "	28.0	75	0.78	1.28	(4)
Virginia Pine		34.3	86	0.96	1.04	(1)
West Coast Hemlock	0% moisture	22.0	75	0.68	1.47	(4)
	0% "	30.0	75	0.79	1.27	(4)
	16% "	22.0	75	0.78	1.28	(4)
	16% "	30.0	75	0.91	1.10	(4)
White Pine		31.2	86	0.78	1.28	(1)
Yellow Pine		—	—	1.00	1.00	(3)
Yellow Pine or Fir		—	—	0.80*	1.25	—

For notes see Page 79.

Table 3. Coefficients of Transmission (U) of Masonry Walls[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

Typical Construction	Type of Wall	Thickness of Masonry (Inches)	Wall No.
	**Solid Brick** Based on 4-in. hard brick and the remainder common brick.	8	1
		12	2
		16	3
	**Hollow Tile** Stucco Exterior Finish. The 8-in. and 10-in. tile figures are based on two cells in the direction of flow of heat. The 12-in. tile is based on three cells in the direction of flow of heat. The 16-in. tile consists of one 10-in. tile and one 6-in. tile each having two cells in the direction of heat flow.	8	4
		10	5
		12	6
		16	7
	**Limestone or Sandstone**	8	8
		12	9
		16	10
		24	11
	**Concrete (Monolithic)** These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.	6	12
		10	13
		16	14
		20	15
	Cinder (Monolithic) Conductivity k = 4.36	6	16
		10	17
		16	18
		20	19
	Burned Clay aggregate (Monolithic) Conductivity k = 3.96	6	20
		10	21
		16	22
		20	23
	**Cinder Blocks**	8	24
	Cores filled with dry cinders, 69.7 lb per cu ft.	8	25
	Cores filled with granulated cork, 5.12 lb per cu ft.	8	26
	Cores filled with rock wool, 14.2 lb per cu ft.	8	27
	Based on one air cell in direction of heat flow.	12	28
	Cores filled with granulated cork, 5.24 lb per cu ft.	12	29
	Concrete Blocks	8	30
	Cores filled with granulated cork, 5.14 lb per cu ft.	8	31
	Based on one air cell in direction of heat flow.	12	32
	Burned Clay aggregate Blocks	8	33
	Cores filled with granulated cork, 5.06 lb per cu ft.	8	34
	Burned Clay aggregate Blocks	12	35
	Cores filled with granulated cork, 5.6 lb per cu ft.	12	36

[a]Computed from factors marked by * in Table 2.

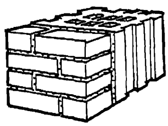
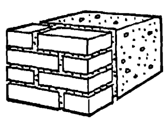
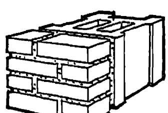
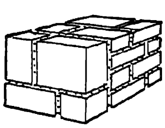
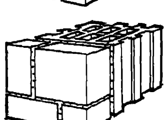
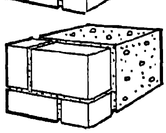
[b]Based on the actual thickness of 2 in. furring strips.

INTERIOR FINISH											
UNINSULATED WALLS						INSULATED WALLS					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plasterboard (⅜ in.)—furred	Decorated building board (½ in.) without plaster—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.)	Plaster (½ in.) on plasterboard (⅜ in.)—airspace faced on one side with bright aluminum foil cemented to plasterboard	Plaster on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[a]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b]) — flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.50	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.22	0.12	0.20
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.14	0.12	0.18	0.11	0.17
0.28	0.27	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.16	0.10	0.15
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.30	0.29	0.22	0.22	0.22	0.17	0.17	0.14	0.12	0.17	0.10	0.16
0.25	0.24	0.19	0.19	0.19	0.16	0.15	0.12	0.11	0.16	0.097	0.14
0.71	0.64	0.37	0.39	0.37	0.26	0.25	0.18	0.15	0.25	0.13	0.23
0.58	0.53	0.33	0.34	0.33	0.24	0.23	0.17	0.14	0.23	0.13	0.21
0.49	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.19	0.11	0.18
0.79	0.70	0.39	0.42	0.39	0.27	0.26	0.19	0.16	0.26	0.13	0.23
0.62	0.57	0.34	0.37	0.34	0.25	0.24	0.18	0.15	0.24	0.13	0.22
0.48	0.44	0.29	0.31	0.29	0.22	0.21	0.16	0.14	0.21	0.12	0.20
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.20	0.12	0.18
0.46	0.43	0.29	0.30	0.29	0.22	0.21	0.16	0.14	0.21	0.12	0.19
0.33	0.31	0.23	0.24	0.23	0.18	0.18	0.14	0.12	0.18	0.11	0.16
0.22	0.22	0.17	0.18	0.17	0.15	0.14	0.12	0.10	0.15	0.09	0.13
0.19	0.18	0.15	0.15	0.15	0.13	0.13	0.11	0.09	0.13	0.09	0.12
0.44	0.41	0.28	0.29	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.30	0.29	0.22	0.23	0.22	0.17	0.17	0.14	0.12	0.17	0.10	0.16
0.21	0.20	0.16	0.17	0.16	0.14	0.14	0.11	0.10	0.13	0.09	0.13
0.17	0.17	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.42	0.39	0.27	0.28	0.27	0.21	0.20	0.16	0.13	0.21	0.12	0.19
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.17	0.11	0.16
0.22	0.21	0.17	0.18	0.17	0.14	0.14	0.12	0.11	0.14	0.09	0.13
0.23	0.22	0.19	0.18	0.18	0.15	0.14	0.12	0.10	0.15	0.09	0.14
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.18	0.11	0.17
0.20	0.19	0.17	0.16	0.16	0.13	0.13	0.11	0.10	0.14	0.09	0.13
0.56	0.52	0.32	0.34	0.32	0.24	0.23	0.17	0.14	0.23	0.12	0.21
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.21	0.12	0.18
0.49	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.22	0.12	0.20
0.36	0.34	0.26	0.26	0.24	0.19	0.19	0.15	0.13	0.18	0.11	0.17
0.18	0.17	0.15	0.15	0.14	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.34	0.32	0.25	0.25	0.24	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.15	0.14	0.13	0.13	0.12	0.11	0.11	0.09	0.08	0.11	0.08	0.10

[a]A waterproof (not vaporproof) membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 4. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS WITH VARIOUS TYPES OF VENEERS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL — FACING	TYPE OF WALL — BACKING	WALL No.
	4 in. **Brick Veneer**[d]	6 in. 8 in. **Hollow Tile**[c] 10 in. 12 in.	**37** **38** **39** **40**
	4 in. **Brick Veneer**[d]	6 in. 10 in. **Concrete** 16 in.	**41** **42** **43**
	4 in. **Brick Veneer**[d]	8 in. **Cinder Blocks**	**44**
		8 in. **Cinder Blocks** — Cores filled with granulated cork, 5.12 lb per cu ft.	**45**
		12 in. **Cinder Blocks**	**46**
		12 in. **Cinder Blocks** — Cores filled with granulated cork, 5.24 lb per cu ft.	**47**
		8 in. **Concrete Blocks**	**48**
		8 in. **Concrete Blocks**—Cores filled with granulated cork, 5.14 lb per cu ft.	**49**
		12 in. **Concrete Blocks**	**50**
		8 in. **Burned Clay aggregate Block**	**51**
		8 in. **Burned Clay aggregate Block**—Cores filled with granulated cork, 5.06 lb per cu ft.	**52**
		12 in. **Burned Clay aggregate Block**	**53**
		12 in. **Burned Clay aggregate Block**—Cores filled with granulated cork, 5.6 lb per cu ft.	**54**
	4 in. **Cut-Stone Veneer**[d]	8 in. 12 in. **Common Brick** 16 in.	**55** **56** **57**
	4 in. **Cut-Stone Veneer**[d]	6 in. 8 in. **Hollow Tile**[c] 10 in. 12 in.	**58** **59** **60** **61**
	4 in. **Cut-Stone Veneer**[d]	6 in. 10 in. **Concrete** 16 in.	**62** **63** **64**

[a]Computed from factors marked by * in Table 2.
[b]Based on the actual thickness of 2-in. furring strips.
[c]The 6-in., 8-in. and 10-in. tile figures are based on two cells in the direction of heat flow. The 12-in. tile is based on three cells in the direction of heat flow.

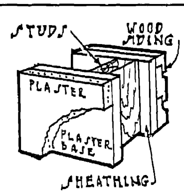
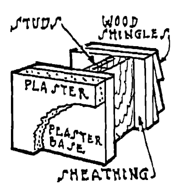
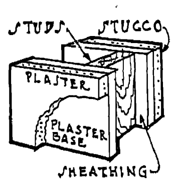
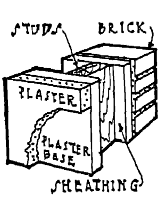
INTERIOR FINISH

Uninsulated Walls						Insulated Walls					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plasterboard (⅜ in.)—furred	No plaster—decorated rigid or building board interior finish (½ in.)—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on cork board (1½ in.) set in cement mortar (½ in.)	Plaster (½ in.) on plasterboard (⅜ in.)—air space faced on one side with bright aluminum foil cemented to plasterboard	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[f]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.18	0.11	0.17
0.34	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.34	0.32	0.23	0.24	0.23	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.27	0.26	0.20	0.21	0.20	0.16	0.16	0.13	0.11	0.16	0.10	0.15
0.57	0.53	0.33	0.35	0.33	0.24	0.23	0.17	0.14	0.23	0.13	0.21
0.48	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.20	0.19	0.16	0.16	0.16	0.13	0.13	0.11	0.10	0.13	0.09	0.12
0.31	0.30	0.22	0.23	0.22	0.18	0.17	0.14	0.12	0.17	0.11	0.16
0.18	0.18	0.15	0.15	0.15	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.44	0.42	0.28	0.30	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.34	0.32	0.24	0.25	0.23	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.17	0.11	0.16
0.17	0.16	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.29	0.28	0.21	0.22	0.21	0.17	0.17	0.13	0.12	0.17	0.10	0.16
0.14	0.14	0.12	0.12	0.12	0.10	0.10	0.09	0.08	0.10	0.07	0.10
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.28	0.27	0.21	0.21	0.21	0.17	0.16	0.13	0.12	0.17	0.10	0.15
0.23	0.22	0.18	0.18	0.18	0.15	0.14	0.12	0.11	0.15	0.095	0.14
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.19	0.11	0.18
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.18	0.11	0.17
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.18	0.11	0.17
0.28	0.26	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.61	0.56	0.34	0.36	0.34	0.25	0.24	0.18	0.15	0.24	0.13	0.22
0.51	0.47	0.31	0.32	0.31	0.23	0.22	0.17	0.14	0.22	0.12	0.20
0.41	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18

[d]Calculations include cement mortar (½ in.) between veneer or facing and backing.
[e]Based on one air cell in direction of heat flow.
[f]A waterproof (not vaporproof) membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 5. COEFFICIENTS OF TRANSMISSION (U) OF VARIOUS TYPES OF FRAME CONSTRUCTION[a]

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	EXTERIOR FINISH	TYPE OF SHEATHING	WALL No.
	Wood Siding or Clapboard	1 in. Wood[d]	65
		25/32 in. Rigid Insulation	66
		½ in. Plasterboard	67
	Wood Shingles	1 in. Wood[d]	68
		25/32 in. Rigid Insulation[e]	69
		½ in. Plasterboard[e]	70
	Stucco	1 in. Wood[d]	71
		25/32 in. Rigid Insulation	72
		½ in. Plasterboard	73
	Brick[f] Veneer	1 in. Wood[d]	74
		25/32 in. Rigid Insulation	75
		½ in. Plasterboard	76

[a]Computed from factors marked by * in Table 2.

[b]These coefficients may also be used with sufficient accuracy for plaster on wood lath or plaster on plasterboard.

[e]Based on the actual width of 2 by 4-in. studding, namely, 3⅝ in.

INTERIOR FINISH

No Insulation Between Studding								Insulation Between Studding				
Plaster on wood lath on studding	Plaster (3/4 in.) on metal lath on studding	Plaster (1/2 in.) on plasterboard (3/8 in.) on studding	Plaster (1/2 in.) on rigid insulation (1/2 in.) on studding	Plaster (1/2 in.) on rigid insulation (1 in.) on studding	Plaster (1/2 in.) on corkboard (1 1/2 in.) on studding	No plaster—decorated rigid or building board interior finish (1/2 in.)	1 in. wood sheathing,[d] furring strips, plaster (1/2 in.) on wood lath	Plaster (1/2 in.) on plasterboard (3/8 in.)—air space faced on one side with bright aluminum foil cemented to plasterboard	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1/2 in.) between studding and in contact with sheathing	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1/2 in.) between studding—2 air spaces	Plaster (3/4 in.) on metal lath[b] on studding—flexible insulation (1 in.) between studding—2 air spaces	Plaster (3/4 in.) on metal lath[b] on studding—rock wool fill (3 5/8 in.[e]) between studding[g,h]
A	B	C	D	E	F	G	H	I	J	K	L	M
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.19	0.17	0.15	0.12	0.072
0.19	0.20	0.19	0.15	0.13	0.10	0.16	0.14	0.15	0.15	0.13	0.10	0.068
0.31	0.33	0.31	0.22	0.17	0.13	0.23	0.19	0.22	0.20	0.17	0.13	0.076
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.17	0.20	0.17	0.15	0.12	0.072
0.17	0.17	0.17	0.14	0.11	0.092	0.14	0.14	0.15	0.13	0.11	0.094	0.064
0.24	0.25	0.24	0.19	0.15	0.11	0.19	0.19	0.22	0.17	0.15	0.12	0.071
0.30	0.32	0.30	0.22	0.16	0.12	0.22	0.19	0.23	0.20	0.17	0.13	0.076
0.22	0.23	0.22	0.17	0.14	0.11	0.19	0.15	0.17	0.16	0.14	0.11	0.071
0.40	0.43	0.40	0.26	0.19	0.14	0.28	0.22	0.26	0.24	0.20	0.14	0.081
0.27	0.28	0.27	0.20	0.15	0.12	0.21	0.17	0.21	0.18	0.16	0.12	0.074
0.21	0.21	0.21	0.16	0.14	0.10	0.17	0.15	0.16	0.15	0.13	0.11	0.068
0.35	0.37	0.35	0.24	0.18	0.13	0.25	0.21	0.24	0.22	0.18	0.14	0.079

[d]Yellow pine or fir—actual thickness about 25/32 in.

[e]Furring strips between wood shingles and sheathing.

[f]Small air space and mortar between building paper and brick veneer neglected.

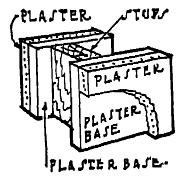
[g]A waterproof (not vaporproof) membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

[h]The coefficients in this column are corrected for the effect of studs.

Table 6. Coefficients of Transmission (*U*) of Frame Interior Walls and Partitions[a]

Coefficients are expressed in Btu per hour per square oot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



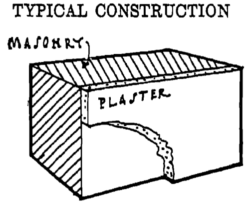
Type of Wall	Wall No.	Single Partition (Finish on One Side of Studding)	Double Partition (Finished on Both Sides of Studding): Air Space Between Studding	Flaked Gypsum Fill[b] Between Studding	Rock Wool Fill[b] Between Studding	½ in. Flexible Insulation Between Studding (One Air Space)
		A	B	C	D	E
Wood Lath and Plaster On Studding	77	0.62	0.34	0.11	0.076	0.21
Metal Lath and Plaster[c] On Studding	78	0.69	0.39	0.11	0.078	0.23
Plasterboard (⅜ in.) and Plaster[d] On Studding	79	0.61	0.34	0.10	0.075	0.21
Plasterboard (⅜ in.) and Plaster[d] On Studding—bright aluminum foil cemented to plasterboard on surface nailed to studding	80	0.42	0.24			0.16
½ in. Rigid Insulation and Plaster[d] On Studding	81	0.35	0.18	0.083	0.063	0.14
1 in. Rigid Insulation and Plaster[d] On Studding	82	0.23	0.12	0.066	0.054	0.097
1½ in. Corkboard and Plaster[d] On Studding	83	0.16	0.081	0.052	0.044	0.070
2 in. Corkboard and Plaster[d] On Studding	84	0.12	0.063	0.045	0.038	0.057

[a]Computed from factors marked by * in Table 2.
[b]Thickness assumed 3⅝ in.
[c]Plaster on metal lath assumed ¾ in. thick.
[d]Plaster assumed ½ in. thick.

Table 7. Coefficients of Transmission (*U*) of Masonry Partitions[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



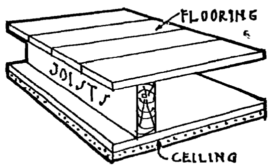
Type of Wall	No.	Plain Walls (No Plaster)	Walls Plastered on One Side	Walls Plastered on Both Sides
		A	B	C
4-in. Hollow Clay Tile	85	0.45	0.42	0.40
4-in. Common Brick	86	0.50	0.46	0.43
4-in. Hollow Gypsum Tile	87	0.30	0.28	0.27
2-in. Solid Plaster	88	--------	--------	0.53

[a]Computed from factors marked by * in Table 2.

TABLE 8. COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



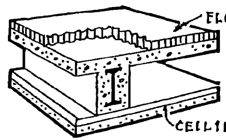
Type of Ceiling	Insulation Between Joists	No.	Type of Flooring: No Flooring	Yellow Pine Flooring[b] on Joists	Yellow Pine Flooring on Rigid Insulation (½ in.) on Joists	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring[b] on Joists	¼ in. Battleship Linoleum on Yellow Pine Flooring[b]
			A	B	C	D	E
No Ceiling	None	1	....	0.46	0.27	0.34	0.34
Metal Lath and Plaster (¾ in.)	None	2	0.69	0.30	0.21	0.25	0.25
Wood Lath and Plaster	None	3	0.62	0.28	0.20	0.24	0.24
Plasterboard (⅜ in.) and Plaster (½ in.)	None	4	0.61	0.28	0.20	0.24	0.23
Rigid Insulation (½ in.) and Plaster (½ in.)	None	5	0.35	0.21	0.16	0.18	0.18
Rigid Insulation (1 in.) and Plaster (½ in.)	None	6	0.23	0.16	0.13	0.14	0.14
Plasterboard (⅜ in.) and Plaster (½ in.)	Bright Aluminum Foil[e]	7	0.53	0.21	0.16	0.18	0.18
Metal Lath and Plaster	Flexible[d] Insulation (1 in.)	8	0.17	0.13	0.11	0.12	0.12
Metal Lath and Plaster	Flexible[d] Insulation (2 in.)	9	0.10	0.086	0.076	0.081	0.081
Metal Lath and Plaster	Rock Wool Fill (3⅝ in.)	10	0.079	0.068	0.063	0.066	0.066
Corkboard (1½ in.) and Plaster (½ in.)	None	11	0.16	0.12	0.10	0.11	0.11
Corkboard (2 in.) and Plaster (½ in.)	None	12	0.12	0.10	0.087	0.094	0.094

[a]Computed from factors marked by * in Table 2.
[b]Thickness assumed to be 25/32 in.
[c]Thickness assumed to be 13/16 in.
[d]Based on one air space with no flooring, and two air spaces with flooring. The value of U will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1 in. furring strips.
[e]Bright aluminum foil cemented to plasterboard

Table 9. Coefficients of Transmission (U) of Concrete Construction Floors and Ceilings[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION

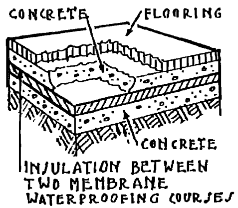


Type of Ceiling	Thickness of Concrete (Inches)	No.	TYPE OF FLOORING: No Flooring (Concrete Bare)[b]	Yellow Pine Flooring[c] on Wood Sleepers Embedded in Concrete[d]	Maple or Oak Flooring[e] on Yellow Pine Sub-Flooring[c] on Wood Sleepers Embedded in Concrete	Tile or Terrazzo[f] Flooring on Concrete	¼ in. Battleship Linoleum Directly on Concrete
			A	B	C	D	E
No Ceiling	4	1	0.65	0.40	0.31	0.61	0.44
	6	2	0.59	0.37	0.30	0.56	0.41
	8	3	0.53	0.35	0.28	0.51	0.38
	10	4	0.49	0.33	0.27	0.47	0.36
½ in. Plaster Applied Directly to Under Side of Concrete	4	5	0.59	0.38	0.30	0.56	0.41
	6	6	0.54	0.35	0.28	0.52	0.38
	8	7	0.50	0.33	0.27	0.47	0.36
	10	8	0.45	0.32	0.26	0.44	0.34
Suspended or Furred Metal Lath and Plaster (¾ in.) Ceiling	4	9	0.37	0.28	0.23	0.36	0.29
	6	10	0.35	0.26	0.22	0.34	0.28
	8	11	0.33	0.25	0.21	0.32	0.27
	10	12	0.32	0.24	0.21	0.31	0.25
Suspended or Furred Ceiling of Plasterboard (⅜ in.) and Plaster (½ in.)	4	13	0.35	0.26	0.22	0.34	0.28
	6	14	0.33	0.25	0.21	0.32	0.26
	8	15	0.31	0.24	0.21	0.30	0.25
	10	16	0.30	0.23	0.20	0.29	0.24
Suspended or Furred Ceiling of Rigid Insulation (½ in.) and Plaster (½ in.)	4	17	0.24	0.20	0.17	0.24	0.21
	6	18	0.23	0.19	0.17	0.23	0.20
	8	19	0.22	0.18	0.16	0.22	0.19
	10	20	0.22	0.18	0.16	0.21	0.19
Plaster (½ in.) on Corkboard (1½ in.) Set in Cement Mortar (½ in.) on Concrete	4	21	0.15	0.13	0.12	0.14	0.14
	6	22	0.14	0.13	0.12	0.14	0.13
	8	23	0.14	0.12	0.11	0.14	0.13
	10	24	0.14	0.12	0.11	0.14	0.13

[a]Computed from factors marked by * in Table 2.
[b]The figures in Column A may be used with sufficient accuracy for concrete floors covered with carpet.
[c]Thickness of yellow pine flooring assumed to be 25/32 in.
[d]The figures in Column B may be used with sufficient accuracy for maple or oak flooring[e] applied directly over the concrete on wood sleepers.
[e]Thickness of maple or oak flooring assumed to be 13/16 in.
[f]Thickness of tile or terrazzo assumed 1 in.

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING[a, e]

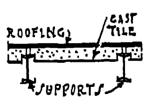
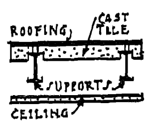
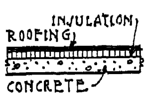
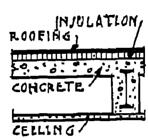
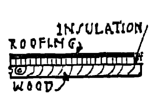
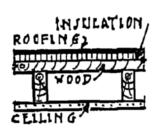
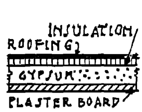
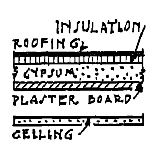
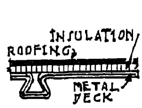
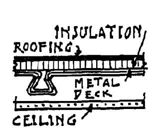
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor, and are based on still air (no wind) conditions.



TYPICAL CONSTRUCTION	Thickness of Concrete (Inches)	No.	TYPE OF FINISH FLOORING				
			No Flooring (Concrete Bare)	Yellow Pine Flooring[b] on Wood Sleepers Resting on Concrete	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring on Wood Sleepers Resting on Concrete	Tile or Terrazzo[d] on Concrete	1/4 in. Battleship Linoleum Directly on Concrete
Type and Thickness of Insulation			A	B	C	D	E
None	4	1	1.07	0.35	0.28	0.98	0.60
	6	2	0.90	0.33	0.27	0.84	0.54
	8	3	0.79	0.32	0.26	0.74	0.50
	10	4	0.70	0.30	0.25	0.66	0.46
None[e]	4	5	0.66	0.29	0.24	0.63	0.44
	8	6	0.54	0.27	0.23	0.52	0.39
1 in. Rigid Insulation[e]	4	7	0.22	0.16	0.14	0.22	0.19
1 in. Rigid Insulation[e]	8	8	0.21	0.15	0.13	0.20	0.18
2 in. Corkboard[e]	4	9	0.12	0.099	0.093	0.12	0.11
2 in. Corkboard[e]	8	10	0.12	0.096	0.090	0.12	0.11

[a]Computed from factors marked by * in Table 2.
[b]Assumed 25/32 in. thick.
[c]Assumed 13/16 in. thick.
[d]Assumed 1 in. thick.
[e]The figures for Nos. 5 to 10, inclusive, include 3 in. cinder concrete placed directly on the ground. The insulation is applied between the cinder concrete and the stone concrete. Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations.

Table 11. Coefficients of Transmission (*U*) of Various Types of Flat Roofs Covered with Built-Up Roofing[a]

Typical Construction: Without Ceilings	Typical Construction: With Metal Lath and Plaster Ceilings[d]	Type of Roof Deck	Thickness of Roof Deck (Inches)	No.
		**Precast Cement Tile**	1⅝	1
		**Concrete** **Concrete** **Concrete**	2 4 6	2 3 4
		**Wood** **Wood** **Wood** **Wood**	1[b] 1½[b] 2[b] 4[b]	5 6 7 8
		**Gypsum Fiber Concrete[c] (2 in.) on Plasterboard (⅜ in.)** **Gypsum Fiber Concrete[c] (3 in.) on Plasterboard (⅜ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (½ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (1 in.)**	2⅜ 3⅜ 2½ 3	9 10 11 12
		**Flat Metal Roofs** Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph.	......	13

[a]Computed from factors marked by * in Table 2.
[b]Nominal thicknesses specified—actual thicknesses used in calculations.
[c]Gypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber.

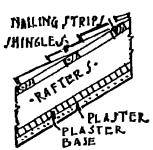
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

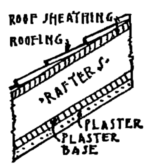
WITHOUT CEILING—UNDER SIDE OF ROOF EXPOSED								WITH METAL LATH AND PLASTER CEILINGS[d]							
No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)	No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	0.42	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11
0.64	0.33	0.22	0.16	0.13	0.21	0.15	0.12	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11
0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095
0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092
0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082
0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097
0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091
0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.10	0.20	0.16	0.13	0.11	0.09	0.12	0.10	0.087
0.19	0.15	0.12	0.10	0.09	0.12	0.10	0.08	0.16	0.13	0.11	0.09	0.08	0.10	0.09	0.077
0.95	0.39	0.25	0.18	0.14	0.23	0.17	0.13	0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11

[d]These coefficients may be used with sufficient accuracy for wood lath and plaster, or plasterboard and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.





Typical Construction	Type of Roofing and Roof Sheathing	Insulation Between Roof Rafters	No.	Type of Ceiling (Applied Directly to Roof Rafters): No Ceiling (Rafters Exposed)	Metal Lath and Plaster (3/4 in.)	Plasterboard (3/8 in.) and Plaster (1/2 in.)	Wood Lath and Plaster	Rigid Insulation (1/2 in.)	Rigid Insulation (1/2 in.) and Plaster (1/2 in.)	Rigid Insulation (1 in.) and Plaster (1/2 in.)	Corkboard (1½ in.) and Plaster (1/2 in.)	Corkboard (2 in.) and Plaster (1/2 in.)
				A	B	C	D	E	F	G	H	I
(figure 1)	Wood Shingles on Wood Strips[b]	None	1	0.46	0.30	0.29	0.29	0.22	0.21	0.16	0.12	0.10
		Bright Aluminum Foil[g]	2	....	....	0.21	....	....	....	....	....	....
		1 in. Flexible[c]	3	....	0.13	0.12	0.12	0.11	0.11	0.092	0.078	0.069
		2 in. Flexible[c]	4	....	0.086	0.083	0.083	0.076	0.075	0.068	0.060	0.054
		3⅝ in. Rock Wool[e]	5	....	0.070	0.070	0.070	0.064	0.064	0.057	0.052	0.047
(figure 2)	Asphalt Shingles, Rigid Asbestos Shingles, Composition Roofing, or Slate or Tile Roofing[d] on Wood Sheathing[f]	None	6	0.56	0.34	0.32	0.32	0.24	0.23	0.17	0.13	0.11
		Bright Aluminum Foil[g]	7	....	....	0.24	....	....	....	....	....	....
		1 in. Flexible[c]	8	....	0.13	0.13	0.13	0.11	0.11	0.095	0.080	0.071
		2 in. Flexible[c]	9	....	0.088	0.087	0.087	0.079	0.078	0.070	0.062	0.056
		3⅝ in. Rock Wool[e]	10	....	0.073	0.072	0.072	0.067	0.065	0.059	0.053	0.048

[a]Computed from factors marked by * in Table 2. Nos. 6 to 10, inclusive, based on ½ in. thick slate.
[b]Based on 1 in. by 4 in. strips spaced 2 in.
[c]Figures based on two air spaces. Insulation may also be applied to under side of roof rafters with furring strips between.
[d]Roofing felt between roof sheathing and slate or tile neglected in calculations.
[e]Assumed 3⅝ in. thick based on the actual width of 2 in. by 4 in. rafters. These coefficients are corrected for the effect of studs.
[f]Sheathing assumed 25/32 in. thick.
[g]Air space faced on one side with bright aluminum foil applied to plasterboard.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS BLOCK WALLS

Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window, skylight or wall

Section A. Windows and Skylights

DESCRIPTION	U
Single	1.13[a, c]
Double	0.45[a, e]
Triple	0.281[a, e]

Section B. Solid Wood Doors[b, c]

NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U EXPOSED DOOR	U[d] WITH GLASS STORM DOOR
1	25/32	0.69	0.42
1¼	1 1/16	0.59	0.38
1½	1 5/16	0.52	0.35
1¾	1⅜	0.51	0.35
2	1⅝	0.46	0.32
2½	2⅛	0.38	0.28
3	2⅝	0.33	0.25

Section C. Hollow Glass Block Walls

DESCRIPTION	U STILL AIR BOTH SIDES	U STILL AIR INSIDE, 15 MPH OUTSIDE
Smooth surface glass blocks 7¾ x 7¾ x 3⅞ in. thick	0.40	0.49
Ribbed surface glass blocks 7¾ x 7¾ x 3⅞ in. thick	0.38	0.46

[a]See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.
[b]Computed using $C = 1.15$ for wood; $f_i = 1.65$ and $f_o = 6.0$.
[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per hour per square foot per degree difference between inside and outside air temperatures.
[d]These values may also be used with sufficient accuracy for wood storm doors. Neglect storm doors if loose and use values for exposed doors.
[e]Air spaces assumed to be ¾ in. or more in width.

coefficient of transmission of a top floor ceiling, unheated attic space, and pitched roof, per square foot of ceiling area, is as follows:

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}} \tag{6}$$

where

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

Stating the formula in terms of the total heat resistance of the ceiling and roof,

$$\frac{1}{U} = R = \frac{1}{U_{ce}} + \frac{1}{U_r \times n} \tag{7}$$

In selecting the values to be used for U_r and U_{ce} it should be noted that the under surface of the roof and the upper surface of the ceiling are

more nearly equivalent to the boundary surfaces of an internal air space than they are to the external surfaces of a wall. It would be more nearly correct to use a value of 2.2 rather than the usual value of 1.65 as coefficients for these surfaces. In most cases this would make only a minor change in U. It should be noted that the over-all coefficient should be multiplied by the ceiling and not the roof area.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall spaces the combined coefficients may be used for determining the heat loss through the roof construction between the attic and top floor ceiling. If the unheated attic contains windows and vertical wall spaces these must be taken into consideration in calculating the roof area and also its coefficient U_r. In this case an approximate value of U_r may be obtained as the summation of the coefficient of each individual section such as the roof, vertical walls or windows times its percentage of total area. This coefficient may be used with reasonable accuracy in the above formulae. If, however, there are roof ventilators such that the attic air is substantially at outside temperature, then the roof should be neglected and only the coefficient for the top floor ceiling construction used.

Basements and Unheated Rooms

The heat loss through floors into basements and into unheated rooms kept closed may be computed by assuming a temperature for these rooms of 32 F. The coefficients of transmission for concrete floors on ground (Table 10) are based on the assumption that the heat-resisting value of the floor extends downward and stops at the under side of the concrete. It is probable, however, that the dirt underneath has some heat-resistance value extending to a considerable depth, which would result in substantially lower heat transmission coefficients than given in Table 10. This problem is now the subject of research. Additional information on the inside and outside temperatures to be used in heat loss calculations is given in Chapter 5.

CONDENSATION IN BUILDINGS

The water vapor or moisture mixed with the air in buildings will be transmitted through many types of building construction if there is a difference in the vapor pressures on the two sides of the structure. Such water vapor will also condense whenever it comes in contact with surfaces or objects at or below the dew-point temperature. Thus two types of condensation problems are encountered in building practice, namely (1) *Surface condensation* or condensation on the interior building surfaces including the walls, ceiling (or roof) and glass, and (2) *Interstitial condensation* or the transmittance of the vapor through the building materials and condensation of the moisture on surfaces or voids within the materials of construction.

Condensation within the construction as well as condensation on the interior surfaces does not necessarily occur in all buildings but only in isolated cases when conditions conducive to such condensation exist. The probability of condensation increases with the relative humidity or vapor pressure and with the temperature difference and, in the case of inter-

stitial condensation, decreases with the vapor resistance on the warm side of the wall.

Condensation on interior building surfaces[5] (surface condensation) may be eliminated by either reducing the relative humidity or by maintaining the interior surfaces at or above the dew-point temperature. Permissible relative humidities for various wall, roof or glass coefficients and temperature differences may be determined from Fig. 2. The permissible relative humidity for any specific type of construction may be determined by first ascertaining the coefficient of transmission (U) of the construction and then locating this coefficient on the horizontal scale of Fig. 2. A vertical line drawn to the proper outside temperature curve and then to the left hand scale will indicate the permissible relative humidity for the conditions involved. The dotted line shown in Fig. 2 indicates the per-

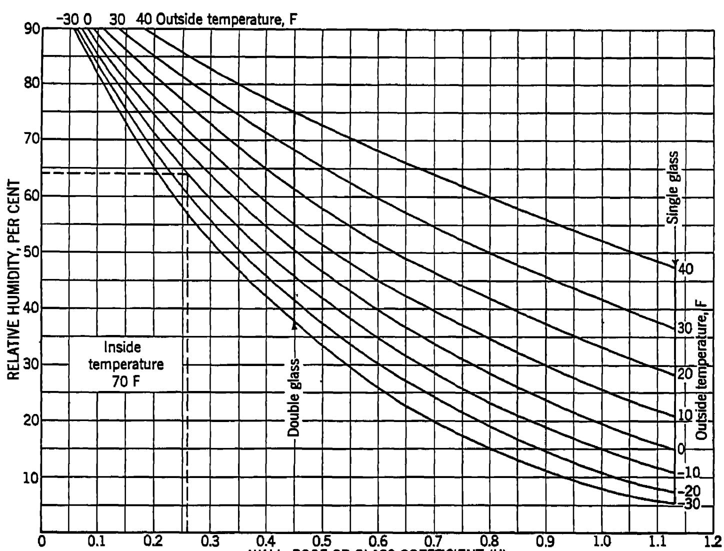


FIG. 2. PERMISSIBLE RELATIVE HUMIDITIES FOR VARIOUS TRANSMISSION COEFFICIENTS

missible relative humidity (64 per cent) if surface condensation is to be avoided, for a frame wall having a coefficient of 0.26 and for an outside temperature of −10 F.

Condensation within the construction may likewise be prevented by eliminating the moisture at the source or by providing a barrier on the warm side of the insulation construction. A good vapor barrier construction may be obtained with a vapor-proof paper properly applied under the plaster or a vapor-proof finish on the interior surface of the wall[6]. In the case of attics, the greater the heat resistance in the top floor ceiling, the lower the attic temperature and consequently the

[5]Permissible Relative Humidities in Humidified Buildings, by Paul D. Close (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1939, p. 766).

[6]Condensation within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 95).

greater the tendency for condensation to take place on the under side of the roof boards which moisture will drop on to the ceiling. Thus where thick insulations are installed between ceiling joists, it is desirable to allow openings for outside air circulation through attic space as a precaution against condensation on the underside of the roof even though barriers are used in the ceiling below.

REFERENCES

A.S.H.V.E. RESEARCH REPORTS:

No. 852—Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 123).

No. 895—Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 301).

No. 914—Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 33).

No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S. H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).

No. 964—The Heat Conductivity of Wood at Climatic Temperature Differences, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 329).

No. 996—Insulating Value of Bright Metallic Surfaces, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 413).

No. 1026—Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Clifford Carlson (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 33).

No. 1048—Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Robert Lander (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 33).

Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 285).

Importance of Radiation in Heat Transfer Through Air Spaces, by E. R. Queer (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 77).

Condensation of Moisture and Its Relation to Building Construction and Operation, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 231).

A Theory Covering the Transfer of Vapor Through Materials, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 545).

Properties of Metal Foil as an Insulating Material, by J. L. Gregg (*Refrigerating Engineering*, May, 1932).

Thermal Insulation with Aluminum Foil, by R. B. Mason (*Industrial and Engineering Chemistry*, March, 1933).

Thermal Insulation of Buildings, Technical Paper No. 11 (*American Architect*, May, 1934).

Heat Insulation as Applied to Buildings and Structures, by E. A. Allcut, University of Toronto, 1934.

House Insulation, Its Economies and Applications, by Russell E. Backstrom (Report of the National Committee on Wood Utilization, *United States Government Printing Office*, 1931).

Heat Transmission Through Building Materials, by F. B. Rowley and A. B. Algren, *University of Minnesota Engineering Experiment Station Bulletin* No. 8.

Aluminum Foil Insulation (*National Bureau of Standards Letter Circular* No. LC535, October 15, 1938).

Effect of Ceiling Insulation upon Summer Comfort, by T. D. Phillips (*National Bureau of Standards*, Report BMS52, July 1, 1940).

Chapter 4

AIR LEAKAGE

Nature of Air Infiltration, Infiltration Through Walls, Window Leakage, Door Leakage, Selection of Wind Velocity, Crack Length used for Computations, Multi-Story Buildings, Heat Equivalent of Air Infiltration

AIR leakage losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building, such as cracks around doors and windows, fireplaces and chimneys. This leakage of air must be considered in heating and cooling calculations. (See Chapters 5 and 6.)

NATURE OF AIR INFILTRATION

The natural movement of air through building construction is due to two causes. One is the pressure exerted by the wind; the other is the difference in density of outside and inside air because of differences in temperature.

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages owing to this building up of pressure will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Mechanical ventilating systems are frequently designed to produce positive or negative pressures in an enclosure which are greater or lower than prevalent wind pressures. In such designs, if the rate at which air is specified to be introduced to or removed from the enclosure by positive means exceeds the infiltration rate, it is common practice to use the greater value in determining the heating capacity to warm the outside air.

The air exchange owing to temperature difference, inside to outside, is not appreciable in low buildings. In tall, single story buildings with openings near the ground level and near the ceiling, this loss must be considered. Also in multi-story buildings it is a large item unless the sealing between various floors and rooms is quite perfect. This temperature effect is a *chimney action*, causing air to enter through openings at lower levels and to leave at higher levels.

A complete study of all of the factors involved in air movement through building constructions would be very complex. Some of the complicating factors are: the variations in wind velocity and direction; the exposure of the building with respect to air leakage openings and with respect to adjoining buildings; the variations in outside temperatures as influencing the chimney effect; the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors; the influence of a planned air supply and the related outlet vents; and the variation from the average of individual building units. A study of infiltration points to the need for care in the obtaining of good building construction, or unnecessarily large heat losses will result.

INFILTRATION THROUGH WALLS

Table 1 gives data on infiltration through brick and frame walls. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it

TABLE 1. INFILTRATION THROUGH WALLS[a]

Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	1.75	4.20	7.85	12.2	18.6	22.9
	Plastered	0.017	0.037	0.066	0.107	0.161	0.236
13 in. Brick Wall	Plain	1.44	3.92	7.48	11.6	16.3	21.2
	Plastered	0.005	0.013	0.025	0.043	0.067	0.097
Frame Wall, with lath and plaster[b]		0.03	0.07	0.13	0.18	0.23	0.26

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed at the end of this chapter.

[b]Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster.

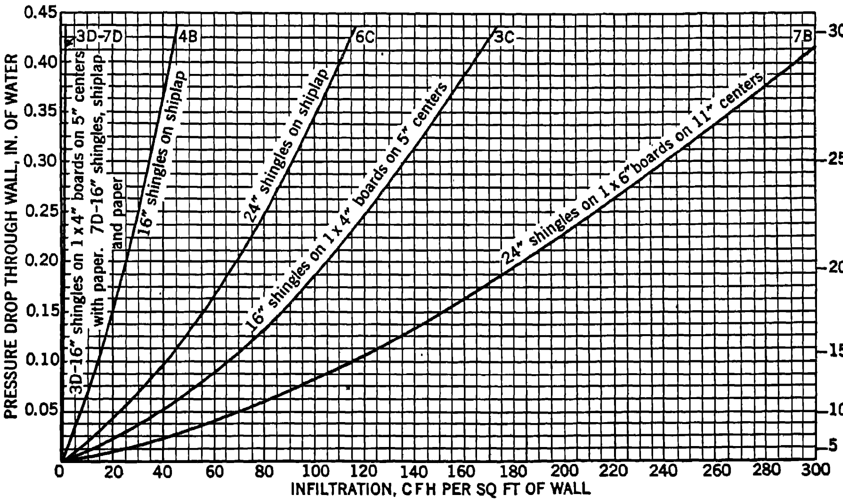


FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them.

The amount of infiltration that may be expected through single walls used in farm and other shelter buildings, is shown in Fig. 2. The infiltration indicated in Figs. 1 and 2 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

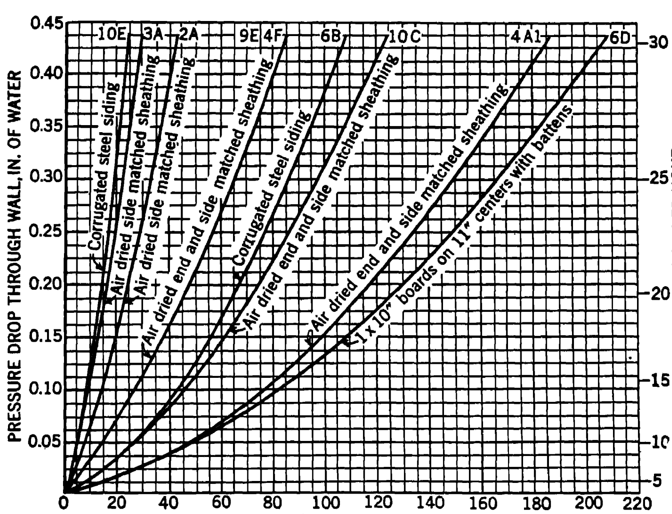


FIG. 2. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

Table 2. Infiltration Through Windows

Expressed in Cubic Feet per Foot of Crack per Hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked[b]	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance.[c] Includes wood frame leakage[d]	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped[d]	4.3	13.0	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance.[e] Includes wood frame leakage[d]	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped[d]	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted, 1/16-in. crack[g]	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in. crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in. crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in. crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in. crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in. crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table, with the exception of those for double-hung and hollow metal windows, are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed at the end of this chapter.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests.

[c]The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e]A 3/32-in. crack and clearance represents a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h]Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and it has refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. 1/32-in. crack is obtainable in the best practice of manufacture and installation, 3/64-in. crack considered to represent average practice.

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[j]Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

WINDOW LEAKAGE

The amount of infiltration for various types of windows is given in Table 2. The fit of double-hung wood windows is determined by crack and clearance. Crack thickness is equivalent to one-half the difference between the inside window frame dimension and the outside sash width. The difference between the width of the window frame guide and the sash thickness is considered as the clearance. The length of the perimeter opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. Values of leakage shown in Table 2 for the average double-hung wood window were determined by setting the average measured crack and clearance found in a field survey of a large number of windows on nine windows tested in the laboratory. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows. When storm sash are applied to poorly fitted windows, a reduction in leakage of 50 per cent may be secured.

DOOR LEAKAGE

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should

be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making cooling load determinations and in the absence of adequate research data the values given in Table 3 represent current engineering practice. These values are based on the average number of persons in a room at a specified time, which may also be the same occupancy assumed for determining the outside ventilation requirements outlined in Chapters 2 and 6.

TABLE 3. INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS[a]
Expressed in Cubic Feet per Minute per Person in Room

Application	Pair 36 in. Swinging Doors, Single Entrance[b]
Bank	7.5
Barber Shop	4.5
Broker's Office	7.0
Candy and Soda	6.0
Cigar Store	25.0
Department Store	8.0
Dress Shop	2.5
Drug Store	7.0
Furrier	2.5
Hospital Room	3.5
Lunch Room	5.0
Men's Shop	3.5
Office	3.0
Office Building	2.0
Public Building	2.5
Restaurant	2.5
Shoe Store	3.5

[a]For doors located in only one wall or where doors in other walls are of revolving type.

[b]Vestibules with double pair swinging doors, infiltration may be assumed 75 per cent of swinging door values.

Infiltration for 72 in. revolving doors may be assumed 60 per cent of swinging door values.

SELECTION OF WIND VELOCITY

Although all authorities do not agree upon the value of the wind velocity that should be chosen for any given locality, it is common engineering practice to use the average wind velocity during the three coldest months of the year. Average wind velocities for the months of December, January and February for various cities in the United States and Canada are given in Table 2, Chapter 5.

In considering both the transmission and infiltration losses, the more exact procedure would be to select the outside temperature and the wind velocity corresponding thereto, based on Weather Bureau records, which would result in the maximum heat demand. Since the proportion of transmission and infiltration losses varies with the construction and is

different for every building, the proper combination of temperature and wind velocity to be selected would be different for every type of building, even in the same locality. Furthermore, such a procedure would necessitate a laborious cut-and-try process in every case in order to determine the worst combination of conditions for the building under consideration. It would also be necessary to consider heat lag due to heat capacity in the case of heavy masonry walls, and other factors, to arrive at the most accurate solution of the problem. Although heat capacity should be considered wherever possible, it is seldom possible to accurately determine the worst combination of outside temperature and wind velocity for a given building and locality. The usual procedure, with modification explained in Chapter 5, is to select an outside temperature which is not more than 15 F above the lowest recorded, and the average wind velocity during the months of December, January and February.

The direction of prevailing winds may usually be included within an angle of about 90 deg. The windows that are to be figured for prevailing and non-prevailing winds will ordinarily each occupy about one-half the perimeter of the structure, the proportion varying to a considerable extent with the plan of the structure. (See discussion of wind movement in Chapter 41 and Table 2 in Chapter 5.)

CRACK LENGTH USED FOR COMPUTATIONS

In no case should the amount of crack used for computation be less than half of the total crack in the outside walls of the room. Thus, in a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building.

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 4. This method may be used to advantage as a check on the calculations made in the more exact manner.

MULTI-STORY BUILDINGS

In tall buildings, infiltration may be considerably influenced by temperature difference or chimney effect which will operate to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone* is located at mid-height of a building, and that the temperature difference is 70 F, the following formulae may be used to determine an equivalent wind velocity

TABLE 4. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION

KIND OF ROOM OR BUILDING	NUMBER OF AIR CHANGES TAKING PLACE PER HOUR
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Living Rooms	1 to 2
Dining Rooms	1 to 2
Bath Rooms	2
Drug Stores	2 to 3
Clothing Stores	1
Churches, Factories, Lofts, etc.	½ to 3

to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$M_e = \sqrt{M^2 - 1.75\,a} \qquad (1)$$

$$M_e = \sqrt{M^2 + 1.75\,b} \qquad (2)$$

where

M_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
M = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height.
b = distance if *below* mid-height.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the greater heights makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels even to the point of omitting all heating surface on the top several floor levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

HEAT EQUIVALENT OF AIR INFILTRATION

Sensible Heat Loss

The heat required to warm cold outside air, which enters a room by infiltration, to the temperature of the room is given by the equation:

$$H_s = 0.24\,Q\,d\,(t_i - t_o) \tag{3}$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t_i Btu per hour.
0.24 = specific heat of air.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_o, pounds per cubic foot.
t_i = room air temperature, degrees Fahrenheit.
t_o = outside air temperature, degrees Fahrenheit.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room for the maintenance of proper winter comfort conditions, it is necessary to determine the heat equivalent to evaporate the required amount of water vapor, which may be calculated by the equation:

$$H_l = Q\,d\left(\frac{M_i - M_o}{7000}\right)L \tag{4}$$

where

H_l = heat required to increase moisture content of air leaking into building from M_o to M_i, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_i, pounds per cubic foot.
M_i = vapor density of inside air, grains per pound of dry air.
M_o = vapor density of outside air, grains per pound of dry air.
L = latent heat of vapor at M_i, Btu per pound.

It is sufficiently accurate to use d = 0.075 lb, in which case Equation 3 reduces to 5 and if the latent heat of vapor is assumed for general conditions as 1060 Btu per pound Equation 4 reduces to 6.

$$H_s = 0.018\,Q\,(t_i - t_o) \tag{5}$$
$$H_l = 0.0114\,Q\,(M_i - M_o) \tag{6}$$

Changing the temperature and vapor subscripts in Equations 5 and 6 to $(t_o - t_i)$ and $(M_o - M_i)$ permits the use of these same formulae for determining the sensible and latent heat gains due to infiltration in cooling load computations.

If a building has more than one room which is divided by interior walls or partitions, it is sufficiently accurate to use half of the total infiltration losses for determining the total heat requirements. Where buildings

4. Measure up net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building.

5. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See Items 1 and 2.)

6. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 4.)

7. The sum of the heat losses by transmission (Item 5) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 6) of the cold air entering by infiltration represents the total heat loss equivalent for any building.

Item 7 represents the heat losses after the building is heated and under stable operating conditions in coldest weather. Additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified standard inside temperature.

The rate at which this additional heat is required depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated.

This additional heat may be figured and allowed for as conditions require, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed

TABLE 1. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

TYPE OF BUILDING	DEG FAHR	TYPE OF BUILDING	DEG FAHR
SCHOOLS		THEATERS—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	HOTELS—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
HOSPITALS—			
Private rooms	70–72	HOMES	70–72
Private rooms (surgical)	70–80	STORES	65–68
Operating rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	FOUNDRIES AND BOILER SHOPS	50–60
		PAINT SHOPS	80

[a]The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. (See Chapter 2.)

for heating-up during the few minimum temperature days, no allowance is made except in the size of boilers or furnaces.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building and which should always be stated in the heating specifications is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 1 presents values which conform with good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 2. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H. V.E. Research Laboratory, is 66 deg.

According to Fig. 6, Chapter 2, for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures listed in Table 1, consideration should be given to whether provision is to be made for humidification, and if so, the actual relative humidity to be maintained.

Temperature at Proper Level: In making the actual heat loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 1 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level. In the case of heated spaces adjacent to unheated spaces, it will usually be sufficient to assume the temperature in such spaces as the mean between the temperature of the inside heated spaces and the outside air temperature, excepting where attic temperature may be calculated as discussed later or where the combined heat transmission coefficient of the roof and ceiling can be used, in which case the usual inside and outside temperatures should be applied. (See discussion regarding the use of combined coefficients of roofs, attics and top-floor ceilings Chapter 3.

Attic Temperature: It is the practice in many cases to estimate the heat loss through top-floor ceilings by assuming the attic temperature to be the mean between the inside and outside temperatures. In the case of attics with thick insulations between the ceiling joists, the attic temperature will ordinarily be only a few degrees above the outside temperatures rather than the mean between the inside and outside temperatures.

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G
State	City	Average Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Recommended Design Temperature	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
Ala.	Birmingham	53.8	−10	10	8.5	N
	Mobile	58.9	−1	20	10.4	N
Ariz.	Flagstaff	35.8	−25	−10	7.8	SW
	Phoenix	59.5	12	25	6.4	E
Ark.	Fort Smith	50.4	−15	5	8.1	E
	Little Rock	51.6	−12	10	8.7	NW
Calif.	Los Angeles	58.5	28	30	6.3	NE
	San Francisco	54.2	27	30	7.6	N
Colo.	Denver	38.9	−29	−15	7.5	S
	Grand Junction	38.9	−21	−10	5.3	NW
Conn.	New Haven	38.4	−15	0	9.7	N
D. C.	Washington	43.4	−15	0	7.1	NW
Fla.	Jacksonville	62.0	10	25	9.2	NE
Ga.	Atlanta	51.5	−8	10	12.1	NW
	Savannah	58.5	8	15	9.5	NW
Idaho	Lewiston	42.3	−23	−5	5.3	E
	Pocatello	35.7	−28	−10	9.6	SE
Ill.	Chicago	36.4	−23	−10	12.5	W
	Springfield	39.8	−24	−10	10.1	NW
Ind.	Evansville	45.1	−16	0	9.8	S
	Indianapolis	40.3	−25	−10	11.5	SW
Iowa	Dubuque	33.9	−32	−20	7.1	NW
	Sioux City	32.6	−35	−20	11.6	NW
Kans.	Concordia	39.8	−25	−10	8.1	S
	Dodge City	41.4	−26	−10	9.8	NW
Ky.	Louisville	45.3	−20	−5	9.9	SW
La.	New Orleans	61.6	7	20	8.8	N
	Shreveport	56.2	−5	10	8.9	SE
Me.	Eastport	31.5	−23	−10	12.0	W
	Portland	33.8	−21	−10	9.2	NW
Md.	Baltimore	43.8	−7	10	7.8	NW
Mass.	Boston	38.1	−18	0	11.2	W
Mich.	Alpena	29.6	−28	−10	12.4	W
	Detroit	35.8	−24	−10	12.7	SW
	Marquette	28.3	−27	−10	11.1	NW
Minn.	Duluth	24.3	−41	−30	12.6	SW
	Minneapolis	29.4	−33	−20	11.3	NW
Miss.	Vicksburg	56.8	−1	15	8.3	SE
Mo.	St. Joseph	40.7	−24	−10	9.3	NW
	St. Louis	43.6	−22	−5	11.6	S
	Springfield	44.3	−29	−10	10.8	SE
Mont.	Billings	34.0	−49	−30	....	W
	Havre	27.6	−57	−30	9.5	SW
Nebr.	Lincoln	37.0	−29	−15	10.5	S
	North Platte	35.4	−35	−20	8.5	W
Nev.	Tonopah	39.4	−10	5	10.0	SE
	Winnemucca	37.9	−28	−15	8.7	NE
N. H.	Concord	33.3	−35	−20	6.6	NW
N. J.	Atlantic City	41.6	−9	5	15.9	NW
N. Y.	Albany	35.2	−24	−5	8.1	S
	Buffalo	34.8	−20	0	17.2	W
	New York	40.7	−14	0	17.1	NW
N. M.	Santa Fe	38.3	−13	0	7.8	NE

[a]United States data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]—(Concluded)

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G
State or Province	City	Average Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Recommended Design Temperature	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
N. C.	Raleigh	50.0	−2	15	8.2	SW
	Wilmington	54.2	5	20	8.5	SW
N. D.	Bismarck	24.6	−45	−30	9.1	NW
	Devils Lake	20.3	−44	−30	10.6	W
Ohio	Cleveland	37.2	−17	−5	13.0	SW
	Columbus	39.9	−20	−10	12.0	SW
Okla.	Oklahoma City	47.9	−17	0	12.0	N
Ore.	Baker	35.2	−24	−15	6.9	SE
	Portland	46.1	−2	10	7.5	S
Pa.	Philadelphia	42.7	−6	0	11.0	NW
	Pittsburgh	41.0	−20	−5	11.7	W
R. I.	Providence	37.2	−17	0	12.8	NW
S. C.	Charleston	57.4	7	15	10.6	SW
	Columbia	54.0	−2	10	8.1	NE
S. D.	Huron	28.2	−43	−25	10.6	NW
	Rapid City	33.4	−34	−20	8.2	W
Tenn.	Knoxville	47.9	−16	0	7.8	SW
	Memphis	51.1	−9	0	9.7	S
Texas	El Paso	53.5	−5	0	10.4	NW
	Ft. Worth	55.2	−8	0	10.4	NW
	San Antonio	60.6	4	10	8.0	NE
Utah	Modena	36.3	−24	−15	8.8	W
	Salt Lake City	40.0	−20	−10	6.7	SE
Vt.	Burlington	31.5	−29	−20	11.8	S
Va.	Lynchburg	46.8	−7	10	7.1	NW
	Norfolk	49.3	2	15	12.5	N
	Richmond	47.0	−3	10	7.9	SW
Wash.	Seattle	44.8	3	15	11.3	SE
	Spokane	37.7	−30	−15	7.1	SW
W. Va.	Elkins	39.4	−28	−10	6.6	W
	Parkersburg	42.6	−27	−10	7.5	SW
Wis.	Green Bay	30.0	−36	−20	10.4	SW
	LaCrosse	31.7	−43	−25	7.3	S
	Milwaukee	33.4	−25	−10	11.5	W
Wyo.	Lander	30.0	−40	−25	5.0	SW
	Sheridan	30.7	−41	−25	6.0	NW
Alta.	Edmonton	23.0	−57	−20	6.5	SW
B. C.	Vancouver	42.0	2	15	4.5	E
	Victoria	43.9	−1.5	15	12.5	N
Man.	Winnipeg	17.5	−47	−30	10.0	NW
N. B.	Fredericton	27.0	−35	−10	9.6	NW
N. S.	Yarmouth	35.0	−12	0	14.2	NW
Ont.	London	32.6	−27	0	10.3	SW
	Ottawa	26.5	−34	−10	8.4	NW
	Port Arthur	22.4	−37	−15	7.8	NW
	Toronto	32.9	−26.5	−5	13.0	SW
P. E. I.	Charlottetown	29.0	−27	−5	9.4	SW
Que.	Montreal	27.8	−29	−10	14.3	SW
	Quebec	24.2	−34	−10	13.6	SW
Sask.	Prince Albert	15.8	−70	−55	5.1	W
Yukon	Dawson	2.1	−68	−50	3.7	S

[a]United States data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to work out the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified. Therefore, pending further studies of actual buildings, it is recommended that the average wind movement in any locality during December, January and February be provided for in computing (1) the heat transmission of a building, and (2) the heat required to take care of the infiltration of outside air.

The first condition is readily taken care of, as explained in Chapter 3, by using a surface coefficient f_0 for the outside wall surface which is based on the proper wind velocity. In case specific data are lacking for any given locality, it is sufficiently accurate to use an average wind velocity of approximately 15 mph which is the velocity upon which the heat transmission coefficient tables in Chapter 3 are based.

In a similar manner, the heat allowance for infiltration through cracks and walls (Tables 1 and 2, Chapter 4) must be based on the proper wind velocity for a given locality. In the case of *tall buildings* special attention must be given to infiltration factors. (See Chapter 4.)

In the past many designers have used empirical *exposure factors* which were arbitrarily chosen to increase the calculated heat loss on the side or sides of the building exposed to the prevailing winds. It is also possible to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities. Recent investigations show, however, that the wind direction indicated by Weather Bureau instruments does not always correspond with the direction of actual impact on the building walls, due to deflection by surrounding buildings.

The exposure factor, which is still in use by many engineers, is usually taken as 15 per cent, and is added to the calculated heat loss on the side or sides exposed to what is considered the prevailing winter wind. There is a need for actual test data on this point, and pending the time when it can be secured, the question must be left to the judgment of the designing engineer. It should be remembered that the values of U in the tables in Chapter 3 are based on a wind velocity of 15 mph and that the infiltration figures are supposed to be selected from the tables in Chapter 4 to correspond to the wind velocities given in Table 2 of the present chapter.

The *Heating, Piping and Air Conditioning Contractors National Association* has devised a method[2] for calculating the square feet of equivalent direct radiation required in a building. This method makes use of ex-

[2]See Standards of *Heating, Piping and Air Conditioning Contractors National Association.*

posure factors which vary according to the geographical location and the angular situation of the construction in question in reference to prevailing winds and their velocity.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In the first case the Btu supplied per hour $= \frac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2546$, and in the second case Btu per hour = bhp $\times$ 2546, in which 2546 is the Btu equivalent of 1 hp-hr. In some mills this is the chief source of heating and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.413. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of illuminating gas about 535 Btu per hour; and one cubic foot of natural gas about 1000 Btu per hour. A Welsbach burner averages 3 cu ft of gas per hour and a fish-tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, see Chapter 2.

In intermittently heated buildings, besides the capacity necessary to care for the normal heat loss which may be calculated according to customary rules, additional capacity should be provided to supply the heat necessary to warm up the cold material of the interior walls, floors, and furnishings. Tests have shown that when a cold building has had its temperature raised to about 60 F from an initial condition of about 0 F, the heat absorbed from the air by the material in the structure may vary from 50 per cent to 150 per cent of the normal heat loss of the building. It is therefore necessary, in order to heat up a cold building within a reasonable length of time, to provide such additional capacity. If the

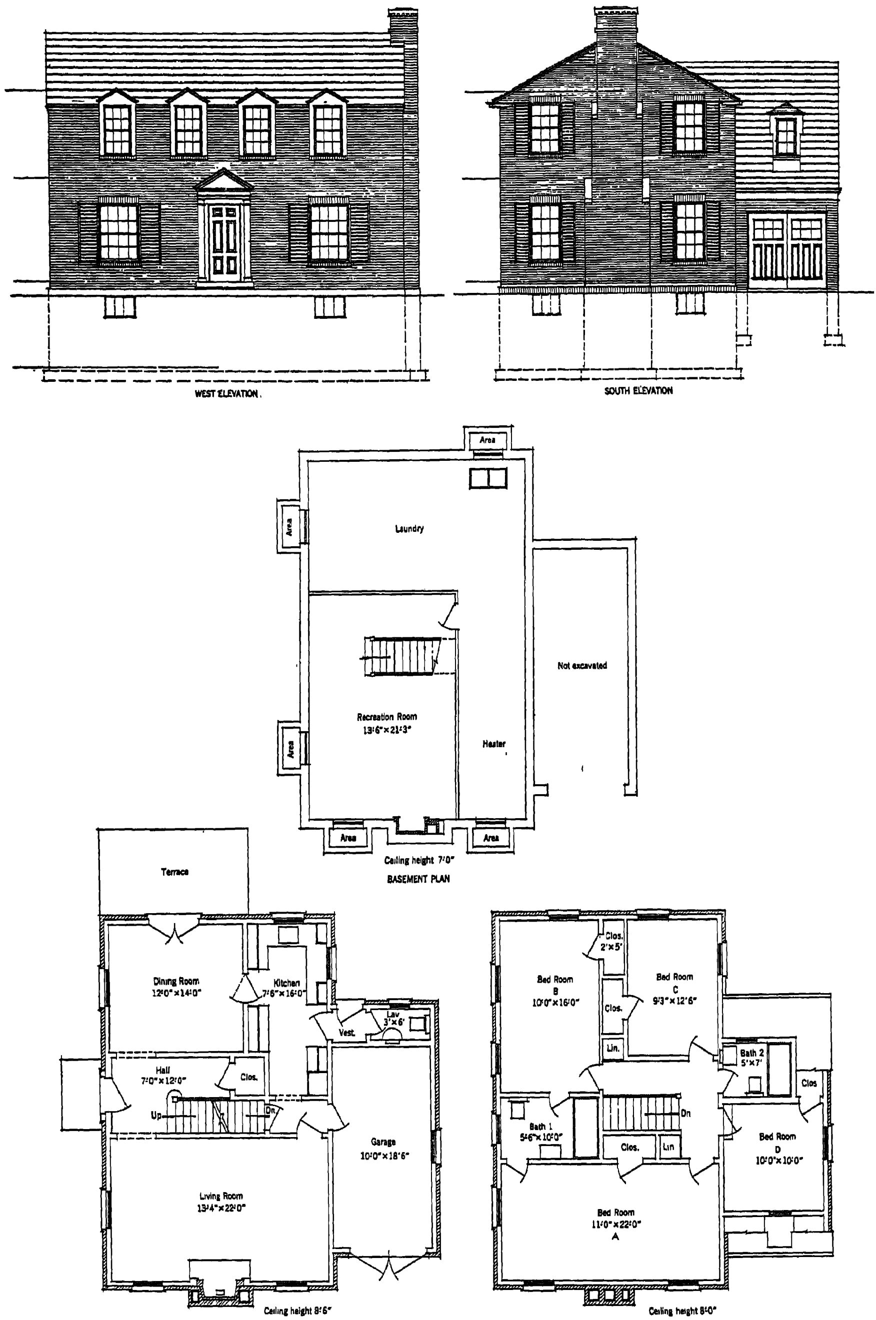


FIG. 1. ELEVATIONS AND FLOOR PLANS OF RESIDENCE

interior material is cold when people enter a building, the radiation of heat from the occupants to the cold material will be greater than is normal and discomfort will result. (See Chapter 2.)

RESIDENCE HEAT LOSS PROBLEMS

Example 2. Calculate the heat loss of residence shown in Fig. 1 located in the vicinity of Chicago. Assume inside and outside design temperatures to be 70 F and −10 F respectively. The attic is unheated. Assume ground temperature to be 45 F. Estimate infiltration by crack method, assuming average wind velocity to be 12.5 mph during December, January and February. No wall, ceiling or roof insulation is to be figured in this problem, but all first and second floor windows are to have storm sash. The building is constructed as follows (transmission coefficients (U) in parentheses):

Walls: Brick veneer, building paper, wood sheathing, studding, metal lath and plaster (0.28). Walls of dormer over garage, same except wood siding in place of brick veneer (0.26).

Attic Walls: Brick veneer, building paper, wood sheathing on studding (0.42).

Basement Walls: 10 in. concrete (0.70).

Roof: Asphalt shingles on wood sheathing on rafters (0.56).

Ceiling: (Second floor): Metal lath and plaster (0.69).

Windows: Double-hung wood windows with storm sash (0.45). Steel casement sash in basement (1.13).

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; metal lath and plaster ceiling below (0.25).

Floor (Basement and Garage): 4 in. stone concrete on 3 in. cinder concrete (0.64).

Solution: The calculations for this problem are given in Table 3, and a summary of the results in Table 4. The values in column F of Table 3 were obtained by multiplying together the figures in columns C, D and E. The heat losses are calculated to the nearest 10 Btu. See reference notes for Table 3 for further explanation of data.

Attention is called to the summary of heat losses (Table 4) of the uninsulated residence (Fig. 1). As storm windows are used in this instance the *glass* and *door* transmission heat losses of 19.5 per cent are relatively small. The infiltration losses (12.0 per cent) are also comparatively small in this case because the storm windows serve substantially the same purpose as weatherstripping. In this problem, the wall, ceiling and floor transmission losses comprise 68.5 per cent of the total. If the building is insulated, the relative heat loss percentages will materially change. (See Example 3 and Table 5.)

Example 3. Calculate the heat loss of residence shown in Fig. 1 based on the same conditions as in Example 2 but insulated throughout as follows (coefficients in parentheses):

Walls: Brick veneer, $^{25}\!/_{32}$ in. insulation board sheathing, studding, 1 in. insulation board lath and plaster (0.14). Walls of dormer over garage same except wood siding in place of brick veneer (0.13).

Attic Walls: Brick veneer, $^{25}\!/_{32}$ in. insulation board sheathing on studding (0.28).

Walls Adjoining Garage: Plaster on 1 in. insulation board, studding, metal lath and plaster (0.18).

Basement Walls (Recreation Room): 10 in. concrete, furring strips, ½ in. insulation board (0.26).

Roof: Asphalt shingles on wood sheathing on rafters (0.56).

Ceiling (Second floor): 1 in. insulation board and plaster; ½ in. insulation board on top of ceiling joists (0.15).

Windows: Same as Example 2.

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; ½ in. insulation board and plaster ceiling below (0.18).

Floor (Under Recreation Room): 4 in. stone concrete, 1 in. insulation board and 3 in cinder concrete (0.22).

Solution: The procedure for calculating the heat losses is similar to that for Example 2. A summary of the results is given in Table 5.

TABLE 3. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 1)

A	B	C	D	E	F	G
Room or Space	Part of Structure	Net Area or Crack Length	Coefficient	Temp. Diff.[a]	Heat Loss (Btu per hour)	Totals (Btu per hour)
Bedroom A	Walls	238 sq ft	0.28	80	5330	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[b]	0.35[c]	80	1010	
	Ceiling[d]	242 sq ft	0.69	39.8	6660	14,440
Bedroom B and Closet	Walls	156 sq ft	0.28	80	3490	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[e]	0.35	80	1010	
	Ceiling[d]	160 sq ft	0.69	39.8	4400	10,340
Bedroom C	Walls	114 sq ft	0.28	80	2560	
	Glass	27 sq ft	0.45	80	970	
	Infiltration	18 lin ft[f]	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	7,330
Bedroom D and Closet	Walls	118 sq ft	0.28	80	2650	
	Glass	20 sq ft	0.45	80	720	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	
	Floor over Garage	110 sq ft	0.25	35[g]	960[m]	8,130
Bathroom 1	Walls	30 sq ft	0.28	80	670	
	Glass	14 sq ft	0.45	80	500	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	55 sq ft	0.69	39.8	1510	3,180
Bathroom 2	Walls	79 sq ft	0.26	80	1770	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	15 lin ft	0.35	80	420	
	Ceiling[d]	35 sq ft	0.69	39.8	960	
	Floor over Garage	35 sq ft	0.25	35	310[m]	3,780
Living Room	Walls	267 sq ft	0.28	80	5980	
	Walls (adjoining garage)	94 sq ft	0.39[h]	35	1280[m]	
	Glass	50 sq ft	0.45	80	1800	
	Infiltration	40 lin ft	0.35	80	1120	10,180
Dining Room	Walls	166 sq ft	0.28	80	3720	
	Glass (doors)	35 sq ft	1.13	80	3160	
	Glass (window)	20 sq ft	0.45	80	720	
	Infiltration[i]	31 lin ft	0.35	80	870	8,470
Kitchen and Entrance to Garage	Walls (outside)	96 sq ft	0.28	80	2150	
	Walls (adjoining garage)	51 sq ft	0.39[h]	35	700[m]	
	Infiltration	27 lin ft	0.35	80	760	
	Glass	18 sq ft	0.45	80	650	
	Door to garage	17 sq ft	0.51	35	300[m]	4,560
Lavette and Vestibule	Walls (outside)	82 sq ft	0.28	80	1840	
	Walls (adjoining garage)	85 sq ft	0.39[h]	35	1160[m]	
	Door	19 sq ft	0.51	80	780	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	19 lin ft	0.35	80	530	4,630

TABLE 3. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 1) (Concluded)

A	B	C	D	E	F	G
ROOM OR SPACE	PART OF STRUCTURE	NET AREA OR CRACK LENGTH	COEFFICIENT	TEMP. DIF.[a]	HEAT LOSS (Btu per hour)	TOTALS (Btu per hour)
Entrance Hall	Walls	39 sq ft	0.28	80	870	
	Door	21 sq ft	0.38	80	640	
	Infiltration	20 lin ft	0.35	80	560	
	Ceiling[d, p]	87 sq ft	0.69	39.8	2490	4,560
Garage	Walls	167 sq ft	0.28	45	2110	
	Glass	53 sq ft	1.13	45	2700	
	Doors	44 sq ft	0.51	45	1010	
	Infiltration	37 lin ft	1.62[j]	45	2700	
	Floor (heat gain)	185 sq ft	0.64[k]	−10[k]	−1180	
	Heat gain				−4710[m]	2,630
Recreation[n] Room	Floor	287 sq ft	0.64	25	4600	
	Walls	220 sq ft	0.70	25	3850	
	Glass	8 sq ft	1.13	80	720	
	Infiltration	8 lin ft	0.76	80	490	9,660
Total						91,890

[a]The inside-outside temperature difference is 70 − (−10) or 80 F, except where otherwise noted.

[b]Only the south windows are used for arriving at the window crack for this room, on the assumption that whatever air enters through the south window cracks will leave through the west window cracks or elsewhere.

[c]Double-hung wood windows with storm sash are assumed to have the same leakage per foot of crack as weatherstripped windows. The air leakage per foot of crack is about 19.5 cu ft per foot of crack for a wind velocity of 12.5 mph. (See Table 2, Chapter 4.) The heat equivalent of the air leakage per hour per degree temperature difference per foot of crack is obtained by multiplying this value by 0.018, or $19.5 \times 0.018 = 0.35$.

[d]In this problem the ceiling heat losses are calculated by estimating the attic temperature and then calculating the loss through the ceiling using the proper temperature difference. This unheated attic is not ventilated during the winter months. The attic temperature is estimated from Equation 1 to be 30.2 F when the outside temperature is −10 F and the room temperature is 70 F. The temperature difference is therefore 70 − 30.2 or 39.8 F.

[e]The window crack in the west wall having two windows is used.

[f]One-half the total crack is used in these rooms.

[g]Temperature in garage assumed to be 35 F.

[h]Coefficient for wall adjoining garage calculated on basis of metal lath and plaster on both sides of studs. ($U = 0.39$.)

[i]The door crack is used for estimating the infiltration in this room and as the French doors are weatherstripped the infiltration coefficient is assumed to be the same as in Note b.

[j]The leakage for the garage doors is assumed to be twice that for poorly-fitted double-hung wood windows or about 90 cu ft per foot of crack for a wind velocity of 12.5 mph. The infiltration coefficient is therefore 0.018×90 or 1.62.

[k]The ground temperature is assumed to be 45 F and, as the garage temperature is 35 F, the heat transfer will be from the ground to the garage, and this heat gain should therefore be subtracted from the heat loss. The floor coefficient ($U = 0.64$) is based on 4 in. stone concrete and 3 in. cinder concrete. This coefficient should probably be lower as the ground itself has some heat resistance value. However, complete data are not as yet available.

[m]The heat losses from various rooms into the garage are heat gains for the garage.

[n]Heat is to be provided for the recreation room and this space is therefore figured on the basis of a 70 F temperature. Heat loss into the basement from recreation room is neglected, the calculations being based only on losses through the outside walls, glass and floor. Ground temperature assumed to be 45 F.

[p]The upstairs hall ceiling is included with the downstairs entrance hall because these are connected by means of the stairway. The heat should be provided downstairs.

Table 4. Summary of Heat Losses of Uninsulated Residence

Heat losses given in Btu per hour

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	5330	6660	....	1440	1010	14,440
Bedroom B	3490	4400	....	1440	1010	10,340
Bedroom C	2560	3300	....	970	500	7,330
Bedroom D	2650	3300	960	720	500	8,130
Bathroom 1	670	1510	....	500	500	3,180
Bathroom 2	1770	960	310	320	420	3,780
Living Room	7260	....	....	1800	1120	10,180
Dining Room	3720	....	....	3880	870	8,470
Kitchen	2850	....	....	950	760	4,560
Lavette	3000	....	....	1100	530	4,630
Entrance Hall	870	2490	....	640	560	4,560
Garage	−1030*	....	−2450†	3410	2700	2,630
Recreation	3850	....	4600	720	490	9,660
Totals	36,990	22,620	3,420	17,890	10,970	91,890
Percentages	40.2	24.6	3.7	19.5	12.0	100.0

*Wall heat loss of 2110 Btu minus wall heat gain of 3140 Btu.

†Heat gains; 960, 310 and 1180 Btu.

Table 5. Summary of Heat Losses of Insulated Residence

Heat losses given in Btu per hour

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	2670	2370	....	1440	1010	7,490
Bedroom B	1750	1570	....	1440	1010	5,770
Bedroom C	1280	1170	....	970	500	3,920
Bedroom D	1320	1170	690	720	500	4,400
Bathroom 1	340	540	....	500	500	1,880
Bathroom 2	820	340	220	320	420	2,120
Living Room	3580	....	....	1800	1120	6,400
Dining Room	1860	....	....	3880	870	6,610
Kitchen	1400	....	....	950	760	3,110
Lavette	1460	....	....	1100	530	3,090
Entrance Hall	440	850	....	640	560	2,490
Garage	−400*	....	−2090†	3410	2700	3,620
Recreation	1430	....	1580	720	490	4,220
Totals	17,950	8,010	400	17,890	10,970	55,220
Percentages	32.5	14.5	0.7	32.4	19.9	100.0

*Wall heat loss of 1050 Btu minus wall heat gains of 590, 320 and 540 Btu.

†Heat gains; 690, 220 and 1180 Btu.

Chapter 6

COOLING LOAD

Design Outside Temperatures, Components of Heat Gain, Normal Heat Transmission, Solar Heat Transmission, Solar Radiation Through Glass, Heat Introduced by Outside Air, Heat Emission of Appliances

LOAD calculations for summer air conditioning are more complicated than heating load calculations because there are more factors to be considered. Due to the variable nature of some of the contributing load components and the fact that they do not necessarily impose their maximum effect simultaneously, considerable care must be used in determining their phase relationship so that equipment of proper capacity may be selected to maintain specified indoor conditions.

The conditions to be maintained in an enclosure are variable and depend upon several factors, especially the outside design conditions, duration of occupancy and relationship between air motion, dry-bulb and wet-bulb temperatures. Information concerning the proper effective temperature to be maintained is given in Chapter 2, where are also tabulated the most desirable indoor conditions to be maintained in summer for exposures over 40 min (see Table 5, Chapter 2).

Summer dry-bulb and wet-bulb temperatures of various cities are given in Table 1. The temperatures are not the maximums but the design temperatures which should be used in air conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system for them. The temperatures shown in Table 1 are based on available design conditions known to be successfully applied.

COMPONENTS OF HEAT GAIN

A cooling load determination is composed of five components which are classified in the following manner:

1. Normal heat transfer through windows, walls, partitions, doors, floors, ceilings, etc.
2. Transfer of solar radiation through windows, walls, doors, skylights, or roof.
3. Heat emission of occupants within enclosures.
4. Heat introduced by infiltration of outside air or controlled ventilation.
5. Heat emission of mechanical, chemical, gas, steam, hot water and electrical appliances located within enclosures.

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Ala.	Birmingham	95	78	5.2	S
	Mobile	95	80	8.6	SW
Ariz.	Phoenix	105	76	6.0	W
Ark.	Little Rock	95	78	7.0	NE
Calif.	Los Angeles	90	70	6.0	SW
	San Francisco	90	65	11.0	SW
Colo.	Denver	95	64	6.8	S
Conn.	New Haven	95	75	7.3	S
Del.	Wilmington	95	78	9.7	SW
D. C.	Washington	95	78	6.2	S
Fla.	Jacksonville	95	78	8.7	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	95	76	7.3	NW
	Savannah	95	78	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	95	76	8.2	S
Ind.	Indianapolis	95	76	9.0	SW
Iowa	Des Moines	95	77	6.6	SW
Kansas	Wichita	100	75	11.0	S
Ky.	Louisville	95	76	8.0	SW
La.	New Orleans	95	79	7.0	SW
	Shreveport	100	78	6.2	S
Maine	Portland	90	73	7.3	S
Md.	Baltimore	95	78	6.9	SW
Mass.	Boston	92	75	9.2	SW
Mich.	Detroit	95	75	10.3	SW
Minn.	Minneapolis	95	75	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	100	76	9.5	S
	St. Louis	95	78	9.4	SW
Mont.	Helena	95	67	7.3	SW
Nebr.	Lincoln	95	75	9.3	S
Nev.	Reno	95	65	7.4	W
N. H.	Manchester	90	73	5.6	NW
N. J.	Trenton	95	78	10.0	SW
N. Y.	Albany	92	75	7.1	S
	Buffalo	93	75	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	90	65	6.5	SE
N. C.	Asheville	90	75	5.6	SE
	Wilmington	95	79	7.8	SW
N. Dak.	Bismarck	95	73	8.8	NW
Ohio	Cincinnati	95	78	6.6	SW
	Cleveland	95	75	9.9	S
Okla.	Oklahoma City	101	76	10.1	S
Ore.	Portland	90	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	95	75	9.0	NW
R. I.	Providence	93	75	10.0	NW
S. C.	Charleston	95	80	9.9	SW
	Greenville	95	76	6.8	NE
S. Dak.	Sioux Falls	95	75	7.6	S
Tenn.	Chattanooga	95	77	6.5	SW
	Memphis	95	78	7.5	SW

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Concluded)

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Texas	Dallas	100	78	9.4	S
	El Paso	100	69	6.9	E
	Galveston	95	80	9.7	S
	Houston	95	78	7.7	S
	San Antonio	100	78	7.4	SE
Utah	Salt Lake City	92	63	8.2	SE
Vt.	Burlington	90	73	8.9	S
Va.	Norfolk	95	78	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	85	65	7.9	S
	Spokane	90	65	6.5	SW
W. Va.	Parkersburg	95	75	5.3	SE
Wis.	Madison	95	75	8.1	SW
	Milwaukee	95	75	10.4	S
Wyo.	Cheyenne	95	65	9.2	S

The components of heat gain, classified by source, are further classified as sensible and latent heat gain.

The first two components fall into the classification of sensible heat gain, that is, they tend to raise the temperature of the air within the structure. The last three components not only produce sensible heat gain but they may also tend to increase the moisture content of the air within the structure.

Normal Heat Transmission

By normal heat transmission, as distinguished from solar heat transmission, is meant the transmission of heat through windows, walls, partitions, etc. from without to interior of enclosure by virtue of difference between outside and inside air temperatures. This load is calculated in a manner similar to that described in Chapter 5 (except that flow of heat is reversed) by means of the formula:

$$H_t = AU(t_o - t) \tag{1}$$

where

H_t = heat transmitted through the material of wall, glass, floor, etc., Btu per hour.
A = net inside area of wall, glass, floor, etc., square feet.
t = inside temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
U = coefficient of transmission of wall, glass, floor, etc., Btu per hour per square foot per degree Fahrenheit difference in temperature (Tables 3 to 13, Chapter 3).

Solar Heat Transmission

Calculations of the solar heat transmitted through walls and roofs are difficult to determine because of periodic character of heat flow and time lag due to heat capacity of construction.

TABLE 2. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE

For 30 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
5:20	0	0	0	0	0	0	0	0
6:00	37	47	23	4.5	4.5	4.5	4.5	11
7:00	119	145	91	11	11	11	11	64
8:00	153	207	149	17	17	17	17	147
9:00	130	194	158	35	21	21	21	213
10:00	86	152	143	63	23.5	23.5	23.5	262
11:00	35	94	85	80	25.5	25.5	25.5	290
12:00	26	26	65	85	65	26	26	300
1:00	25.5	25.5	25.5	80	85	94	35	290
2:00	23.5	23.5	23.5	63	143	152	86	262
3:00	21	21	21	35	158	194	130	213
4:00	17	17	17	17	149	207	153	147
5:00	11	11	11	11	91	145	119	64
6:00	4.5	4.5	4.5	4.5	23	47	37	11
6:40	0	0	0	0	0	0	0	0

TABLE 3. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE

For 35 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
5:07	0	0	0	0	0	0	0	0
6:00	43	49	27	4.5	4.5	4.5	4.5	13
7:00	121	151	97	11	11	11	11	72
8:00	147	207	155	25	17	17	17	151
9:00	120	194	169	49	21	21	21	213
10:00	71	152	156	83	23.5	23.5	23.5	245
11:00	28	94	129	103	25.5	25.5	25.5	288
12:00	26	26	84	109	84	26	26	298
1:00	25.5	25.5	25.5	103	129	94	28	288
2:00	23.5	23.5	23.5	83	156	152	71	245
3:00	21	21	21	49	169	194	120	213
4:00	17	17	17	25	155	207	147	151
5:00	11	11	11	11	97	151	121	72
6:00	4.5	4.5	4.5	4.5	27	49	43	13
6:53	0	0	0	0	0	0	0	0

TABLE 4. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE
For 40 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4:50	0	0	0	0	0	0	0	0
5:00	5	6	4	2.5	2.5	2.5	2.5	5
6:00	49	56	32	4.5	4.5	4.5	4.5	20
7:00	123	162	109	11	11	11	11	85
8:00	137	211	166	29	17	17	17	160
9:00	102	195	181	74	21	21	21	212
10:00	54	152	171	103	23.5	23.5	23.5	244
11:00	28	94	144	124	41	25.5	25.5	281
12:00	26	26	98	128	98	26	26	290
1:00	25.5	25.5	41	124	144	94	28	281
2:00	23.5	23.5	23.5	103	171	152	54	244
3:00	21	21	21	74	181	195	102	212
4:00	17	17	17	29	166	211	137	160
5:00	11	11	11	11	109	162	123	85
6:00	4.5	4.5	4.5	4.5	32	56	49	20
7:00	2.5	2.5	2.5	2.5	4	6	5	5
7:10	0	0	0	0	0	0	0	0

TABLE 5. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE
For 45 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4:25	0	0	0	0	0	0	0	0
5:00	22	20	17	3.5	3.5	3.5	3.5	9
6:00	87	99	56	5.5	5.5	5.5	5.5	27
7:00	151	192	134	12	12	12	12	89
8:00	144	237	188	48	17	17	17	156
9:00	100	199	197	93	21	21	21	205
10:00	46	153	184	121	23.5	23.5	23.5	243
11:00	28	94	158	146	63	25.5	25.5	259
12:00	26	26	116	156	116	26	26	281
1:00	25.5	25.5	63	146	158	94	28	259
2:00	23.5	23.5	23.5	121	184	153	46	243
3:00	21	21	21	93	197	199	100	205
4:00	17	17	17	48	188	237	144	156
5:00	12	12	12	12	134	192	151	89
6:00	5.5	5.5	5.5	5.5	56	99	87	27
7:00	3.5	3.5	3.5	3.5	17	20	22	9
7:35	0	0	0	0	0	0	0	0

The variation in radiation intensity on differently oriented surfaces is given in Fig. 1, and in Tables 2, 3, 4 and 5. The greater part of the radiation intensity is always direct radiation from the sun. However, during the time when the sun is shining, any surface receives radiation of a lower intensity coming from all parts of the sky due to reflection and refraction. This scattered radiation intensity was found to vary from a very low value to values as high as 20 per cent of the total radiation observed on certain days in Pittsburgh. The curves and tables are for combined direct solar and scattered sky radiation, and are given to represent expected design radiation intensity for August 1. They were prepared by the A.S.H.V.E. Laboratory from data[1] obtained by pyrheliometer observations.

A study of these curves discloses the periodic relationship and wide variation in solar intensity on various surfaces. It will be observed that

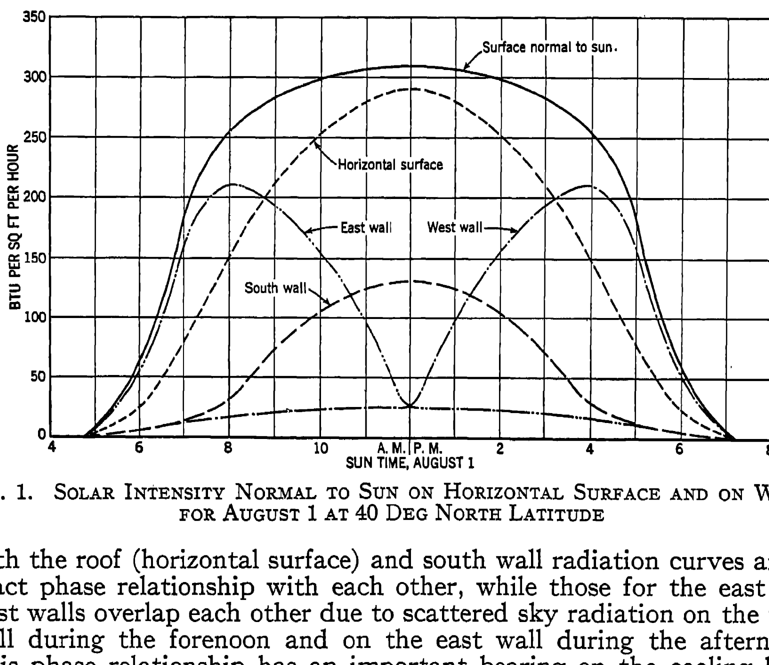


FIG. 1. SOLAR INTENSITY NORMAL TO SUN ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1 AT 40 DEG NORTH LATITUDE

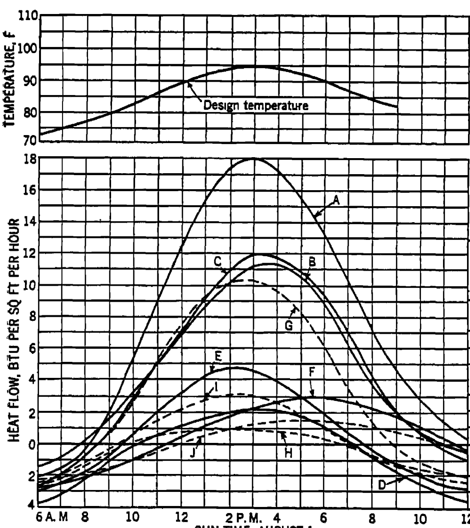
both the roof (horizontal surface) and south wall radiation curves are in exact phase relationship with each other, while those for the east and west walls overlap each other due to scattered sky radiation on the west wall during the forenoon and on the east wall during the afternoon. This phase relationship has an important bearing on the cooling load. Failure to consider the periodic character of heat flow resulting from diurnal movement of the sun and the lag due to heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall, may result in a large error in load calculations.

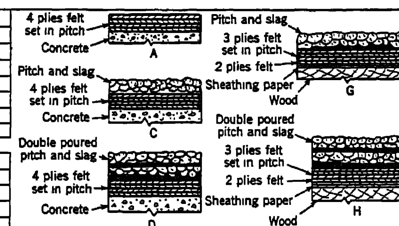
The values of solar intensity appearing in Fig. 1 must not be confused with the actual heat transmission through the wall for much of the solar radiation impinging against the outer surface fails to pass through the wall. Instead it is delivered to the outside air by reflection, radiation,

[1]A.S.H.V.E. RESEARCH PAPER—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson and Burt Gunst (A.S.H.V.E. JOURNAL SECTION *Heating, Piping and Air Conditioning*, April, 1940, p. 264).

convection and conduction. A mathematical solution for the determination of solar heat transmission has been developed but the equations involved are too complex for practical application[2].

The heat flow in summer through various types of roofs and walls has been measured by the A.S.H.V.E. Laboratory. The curves in Fig. 2, give the heat flow through the inside surface of roofs[3] with details of the construction of the roofs tested. The condition for which these results are given are: solar radiation for 40 deg north latitude on August 1 as given in Fig. 1 and Table 4; outdoor design temperature reaching a maximum of 95 F as shown by the temperature curve in Fig. 2 and an indoor temperature of 75 F.





Curve	Type of roof and condition tested	U=Btu per sq ft per hr per F temp. difference on two sides	Heat capacity, Btu per sq ft per F (Values for dry roof)
A	2 in. concrete, smooth asphalt finish (black)	0.40	6.28
B	A plus aluminum paint		
C	2 in. concrete, pitch and slag	0.31	7.37
D	2 in. concrete, double poured pitch and slag (sprinkled)	0.26	8.27
E	D with 1 in. water		
F	D with 6 in. water		
G	2 in. plank, pitch and slag	0.20	3.97
H	2 in. plank, double poured pitch and slag (sprinkled)	0.18	4.95
I	H with 1 in. water		
J	H with 6 in. water		

FIG. 2. RELATION BETWEEN TIME AND HEAT FLOW THROUGH INSIDE SURFACE OF HORIZONTAL ROOFS CORRECTED TO DESIGN DAY OF AUGUST 1

Curves in Fig. 3 were prepared by the A.S.H.V.E. Laboratory from recent tests made there and show the heat flow through the inside surface of three types of walls for various orientations. The results are given for the following conditions: 90 per cent of the solar radiation given in Fig. 1 and Table 4 for 40 deg N latitude on August 1; outdoor design temperature reaching a maximum of 93 F as shown by the temperature curve in Fig. 3 and an indoor temperature of 78 F and 50 per cent relative humidity.

The heat flow shown in Figs. 2 and 3 is a combination of normal transmission and solar radiation transmission and is the total heat flow through the wall or roof. Due to the heat capacity of walls and roofs there is a

[2]A.S.H.V.E. RESEARCH REPORT No. 923—Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. W. Pugh and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 231).

[3]A.S.H.V.E. RESEARCH PAPER — Summer Cooling Load as Affected by Heat Gain Through Dry, Sprinkled and Water Covered Roofs, by F. C. Houghten, H. T. Olson and Carl Gutberlet (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1940, p. 451).

time lag[4] in the transmission of heat through them as shown by the curves. For the types of construction covered in Figs. 2 and 3 and for the conditions indicated, the heat flow through the inside surface at any given time can be read directly. For other types of construction, the curves may be used as a guide in estimating the heat flow. The time lag for other types of construction is included in Table 6 which was prepared by the A.S.H.V.E. Laboratory from data collected by it and by other authorities.

Solar Radiation Transmitted Through Glass

Windows present a problem somewhat different from that of opaque walls, because they permit a large percentage of the solar energy to pass

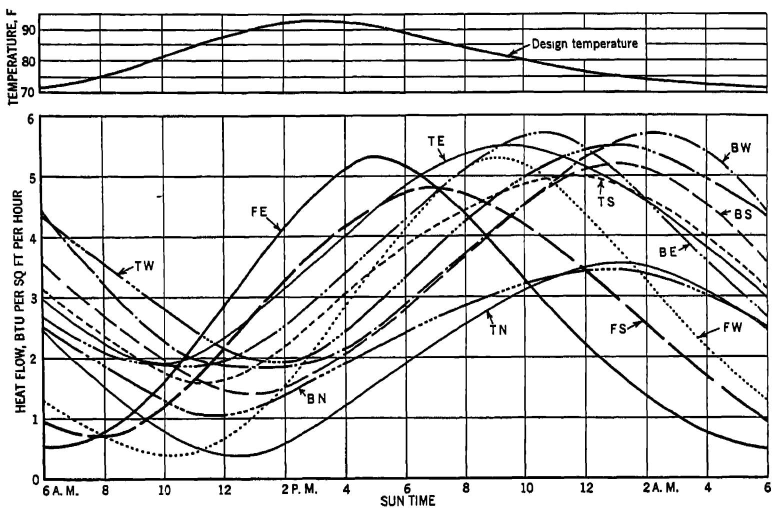


FIG. 3. RELATION BETWEEN TIME AND HEAT FLOW THROUGH THE INSIDE SURFACE OF WALLS OF DIFFERENT CONSTRUCTION AND ORIENTATION ON A 93 F DESIGN DAY WITH 90 PER CENT OF DESIGN SOLAR RADIATION

Walls BE, BS, BW and BN—12 in. solid brick and plaster facing east, south, west and north respectively.
Walls TE, TS, TW and TN—4 in. brick veneer, 8 in. tile and plaster facing east, south, west and north respectively.
Walls FE, FS and FW—4 in. brick veneer, building paper, ⅞ in., matched sheathing, 2 x 4 in. studs, metal lath and plaster facing east, south and west respectively.

through. A small amount is reflected and the balance is absorbed by the glass. The amount absorbed depends upon the character and thickness of the glass and the angle between it and the sun's rays. The temperature of the glass is raised by the absorbed heat and this heat is then delivered to the air on each side in proportion to the difference between the glass and air temperatures[5].

The A.S.H.V.E. tests[6] indicate that a single pane of double strength glass 0.127 in. thick absorbs approximately 11 per cent of the solar

[4]Loc. Cit. Note 2.

[5]Heat Absorbing Glass Windows, by W. W. Shaver (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 287).

[6]A.S.H.V.E. RESEARCH REPORT No. 974—Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 93). A.S.H.V.E RESEARCH REPORT No. 975—Studies of Solar Radiation Through Bare and Shaded Windows by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 101).

TABLE 6. TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

TYPE AND THICKNESS OF WALL OR ROOF	TIME LAG, HOURS
1-in. yellow pine horizontal roof, water proofing, smooth black finish	1
2-in. yellow pine horizontal roof, water proofing, smooth black finish	1¾
4-in. reinforced clay tile horizontal roof, water proofing, slag finish	2¼
2-in. gypsum horizontal roof, water proofing, slag finish	2¼
Slate and slaters felt on 2¼ in. tongue and grooved yellow pine, sloped roof	2½
4-in. gypsum horizontal roof, water proofing, slag finish	4½
6-in. concrete horizontal roof, water proofing, slag finish	5
1-in. concrete, 4-in. cinders, 1½-in. concrete, water proofing, smooth black finish	8
Wood siding, 1-in. sheathing, 2 x 4 studs, lath and plaster	2
Wood siding, 1-in. sheathing, 2 x 4 studs (studding space filled with insulation) lath and plaster	5
4-in. brick, 1-in. sheathing, 2 x 4 studs, lath and plaster	7
4-in. brick, 8-in. tile and plaster	10½
13-in. brick, plastered	12
9-in. brick, 3¾-in. tile, 5¼-in. air space, 3¾-in. tile and 1¼-in. plaster	16

radiation passing through it when the impingement is normal. For smaller angles of impingement, the glass retards percentages of the total radiant energy approximately in proportion to the sine of the angle.

The amount of solar radiation delivered to an unshaded glass surface may be obtained from Tables 2, 3, 4 or 5. These values must be used only for the net glass area on which the sun shines and not the entire glass area. Tests at the A.S.H.V.E. Research Laboratory[7] have determined the percentage of heat from solar radiation actually delivered to a room with various types of outdoor and indoor shading. The data in Table 7 are taken from these tests.

The percentage values in this table were obtained by dividing the total amount of heat actually entering through the shaded window by the total amount of heat calculated to enter through a bare window (solar radiation plus glass transmission, based on observed outside glass temperature). For bare windows on which the sun shines, the transmission of heat from outside air to glass may be small or negative as the glass temperature is raised by the solar radiation absorbed.

In calculating the total heat gain through windows on the sunny side of buildings, it is sufficiently accurate to proceed as outlined herewith:

TABLE 7. SOLAR RADIATION TRANSMITTED THROUGH SHADED WINDOWS

TYPE OF APPURTENANCE	FINISH FACING SUN	PER CENT DELIVERED TO ROOM
Canvas awning	Plain	28
Canvas awning	Aluminum	22
Inside shade, fully drawn	Aluminum	45
Inside shade, one-half drawn	Buff	68
Inside Venetian blind, fully covering window, slats at 45 deg	Aluminum	58
Outside Venetian blind, fully covering window, slats at 45 deg	Aluminum	22

[7]Loc. Cit. Note 6.

Consider the total heat gain as that resulting from solar radiation and neglect the heat transmission through the glass caused by the difference between the temperatures of the inside and outside air. This method should be used except at times when the calculated heat gain per square foot due to normal transmission exceeds the solar intensity. At such times, solar radiation may be neglected and the total heat gain considered as resulting from normal transmission.

The solar heat transmission through windows or skylights may be expressed by the formula:

$$H_G = A_G f I \tag{2}$$

where

H_G = solar radiation transmitted through a window, Btu per hour.

A_G = net area of glass exposed to sun's rays, square feet.

f = percentage of solar radiation (expressed as a decimal) transmitted to the inside (Table 7). For bare windows, $f = 1$.

I = intensity of solar radiation striking surface, Btu per hour per square foot (Tables 2, 3, 4 and 5).

In Equation 2, $f = 1$ for bare windows because the tests from which Table 7 was obtained showed that approximately all of the solar radiation impinging on a bare window became a part of the heat load in the room. This was because almost all of the heat absorbed by the glass flowed into the room by conduction. Other tests[8] have indicated that in the case of a building having floors of high heat capacity such as concrete floors on which the solar radiation falls, approximately one-half of the heat entering a bare window is absorbed by the floor and does not immediately become a part of the cooling load, but is delivered back to the air in the building at a slow rate over a period of 24 hours or longer.

The maximum solar intensity on any surface is of limited duration as shown in Fig. 1. In the case of windows the total energy impinging on the glass before and after the time of maximum intensity is further reduced by increased shading of the glass from the frame, or wall. The cooling load due to solar radiation therefore does not have to be calculated as a steady load. Another point which should be noted is that the maximum solar radiation load on the east wall occurs early in the morning when the outside temperature is low.

Tests have been made which indicated that solar radiation through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sun load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the amount of cooling required will be reduced and there will also be less difference in the cooling requirements of different sides of the building. The total cooling load for a building exposed to the sun on more than one side is of course less than the sum of the maximum cooling loads in the individual rooms since the maximum solar radiation load on the different sides occurs at different

[8]A.S.H.V.E. RESEARCH REPORT No. 1002—Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet, and Albert J. Wahl (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 53).

times. In determining the total cooling load for a building if the time when the maximum load occurs is not obvious, the load should be calculated for various times of day to determine the times at which the sum of the loads on the different sides of the building is a maximum.

The direct solar and scattered sky radiation penetration through glass block panels is given in Table 8 for various times of the day for south east and west exposures for different latitudes on August 1. This table also gives the total heat gain into an air conditioned space when 78 F is maintained indoors, resulting from the effect of both radiation and air to air transmission. These values result from A.S.H.V.E. Laboratory data[9] and apply for expected design radiation intensity, and for a design day

TABLE 8. HEAT GAIN THROUGH GLASS BLOCKS[a]

		Solar Radiation Heat Gain (Direct Plus Sky) Btu per Sq Ft per Hour						Total Heat Gain[b] (Solar Radiation Plus Normal Transmission) Btu per Sq Ft per Hour					
Side		East[c]	West[c]	South				East[c]	West[c]	South			
N. Latitude Degrees		40	40	30	35	40	45	40	40	30	35	40	45
Sun Time	Outside Temp F												
7:00	74	65.0		1.0	2.8	3.0	5.0	61.0		−4.5	−2.0	−0.5	1.0
8:00	76	63.0	0.0	3.0	4.4	6.5	11.0	77.5		0.0	2.0	4.0	5.0
9:00	79	40.0	5.0	5.5	7.1	10.2	13.4	73.5	5.0	5.0	7.0	10.0	12.0
10:00	83	24.0	6.0	8.5	11.3	14.7	17.1	57.5	6.5	11.0	15.0	18.0	20.8
11:00	87	15.5	7.0	12.0	15.2	18.7	21.8	45.0	7.5	16.5	22.0	25.5	32.0
12:00	90	10.0	10.0	14.0	17.4	21.0	24.8	36.5	10.5	21.5	28.0	33.8	40.8
1:00	93	7.0	15.5	12.0	15.2	18.7	21.8	30.0	22.0	25.0	31.8	38.5	46.0
2:00	94	6.0	24.0	8.5	11.3	14.7	17.1	24.0	35.0	26.0	32.0	39.0	47.0
3:00	95	5.0	40.0	5.5	7.1	10.2	13.4	19.5	55.0	24.0	29.8	36.5	45.0
4:00	95	4.5	65.0	3.0	4.4	6.5	11.0	15.5	77.0	20.0	25.5	31.5	40.5
5:00	93	4.0	63.0	1.0	2.8	3.0	5.0	12.5	85.5	15.0	20.0	25.2	33.5
6:00	91	2.5	23.5	0.0	0.7	0.7	3.0	10.5	55.0	9.5	13.5	18.0	25.5
7:00	89	1.5	0.0			0.0	0.7	8.0	18.5	3.5	7.0	11.0	18.0

[a]For August 1.
[b]Inside temperature, 78 F.
[c]For east and west walls these values can be applied to all latitudes between 30 and 45 deg N without excessive errors.

having a maximum temperature of 95 F. The resulting heat gains are averages for four typical glass block designs, two having smooth exterior faces, and the other two having exterior ribbed faces, as made by two different manufacturers.

Heat Emission of Occupants

The heat and moisture given off by human beings under different states of activity are shown in various tables and figures of Chapter 2 which covers the physical and physiological principles of air conditioning. It will be observed from these data that the rate of sensible and latent heat emission by human beings varies greatly depending upon state of activity. In many applications this component becomes a large per-

[9]Loc. Cit. Note 1.

centage of total load. Metabolic rates are markedly variable for some extreme environmental conditions and this is another important factor which must be considered in cooling load computations.

Heat Introduced by Outside Air

An allowance must be made for the heat and moisture in the outside air introduced for ventilation purposes or entering the building through cracks, doors, and other places where infiltration might occur.

The volume of air entering due to infiltration may be estimated from data given in Chapter 4. Information on the amount of outside air required for ventilation will be found in Chapter 2.

In the event the volume of air entering an enclosure due to infiltration exceeds that required for ventilation, the former should be used as a basis for determining the portion of the load contributed by outside air. Where volume of air required for ventilation exceeds that due to infiltration it is assumed that a slight positive pressure will exist within the enclosure with a resulting exfiltration instead of infiltration. In this case the air required for ventilation is used in determining outside air load.

The heat gain resulting from outside air introduced may be determined by the following formula:

$$H = \frac{Q}{v} (h_o - h_i) \tag{3}$$

where

H = heat to be removed from outside air entering the building, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
v = cubic feet of outside air per pound of dry air.
h_o = enthalpy of outside air, Btu per pound of dry air.
h_i = enthalpy of inside air, Btu per pound of dry air.

The latent heat gain resulting from outside air introduced may be determined by the following formula:

$$H_l = \frac{Q}{v} h_{fg} (W_o - W_i) \tag{4}$$

where

H_l = latent heat to be removed, Btu per hour.
h_{fg} = latent heat of evaporation at temperature at which water is condensed, Btu per pound.
W_o = humidity ratio of outside air, pounds water per pound dry air.
W_i = humidity ratio of inside air, pounds water per pound dry air.

Heat Emission of Appliances

Heat generating appliances which give off either sensible heat or both sensible and latent heat in an air conditioned enclosure may be divided into three general classes of equipment or devices:

1. Electrical appliances.
2. Gas appliances.
3. Steam heating appliances.

In the first group may be found such devices as lights, motors, toasters, waffle irons, etc. The capacities of most electrical devices may be determined from the watt capacity indicated on their name plates. The Btu equivalent of heat generated per hour is determined by multiplying the watt capacity by 3.4 (one watthour is equivalent to 3.413 Btu).

The capacities of electric motors are usually expressed in terms of horsepower instead of watts. If the motor efficiency is known, the watts input may be calculated from the formula:

$$P = \frac{746\ (hp)}{n} \tag{5}$$

where

P = motor input, watts.
hp = motor load, horsepower.
n = motor efficiency (expressed as a decimal).

When the motor efficiency is not known the heat equivalent of electrical input can be approximately determined by applying data given in Table 9.

Table 9. Heat Generated by Motors

Nameplate Rating Horsepower	Heat Gain in Btu per Hour per Horsepower	
	Connected Load in Same Room	Connected Load Outside of Room
1/8 to 1/2	4250	1700
1/2 to 3	3700	1150
3 to 20	2950	400

In the second group belong such appliances as coffee urns, gas ranges, steam tables, broilers, hot plates, etc. For heat generating capacities of such appliances refer to Table 10.

Considerable judgment must be exercised in the use of data given in Table 10. Consideration must be given to time of day when appliances are used and the heat they contribute at time of peak load. Only those appliances in use at the time of the peak load need be considered. Consideration must also be given to the way appliances are installed, whether products of combustion are vented to a flue, whether they escape into the space to be conditioned, or whether appliances are hooded allowing part of the heat to escape through a stack. There are no generally accepted data available on the effects of venting and shielding heating appliances but it is believed that, when they are properly hooded with a positive fan exhaust system through the hood, 50 per cent of the heat will be carried away and 50 per cent dissipated in the space to be conditioned. Where latent as well as sensible heat is given off, it is usually safe to assume that all latent heat will be removed by a properly designed and operated vent or hood.

ILLUSTRATION

From the foregoing discussion it is obvious that the determination of the maximum cooling load is rather complicated by reason of the variable nature of contributing load components.

Table 10. Heat Gain from Various Sources

Source	Btu Per Hour		
	Sensible	Latent	Total
Electric Heating Equipment			
Electrical Equipment—Dry Heat—No Evaporated Water	100%	0%	100%
Electric Oven—Baking	80%	20%	100%
Electric Equipment—Heating Water—Stewing, Boiling, etc.	50%	50%	100%
Electric Lights and Appliances per Watt (Dry Heat)	3.4	0	3.4
Electric Lights and Appliances per Kilowatt (Dry Heat)	3413	0	3413
Coffee Urn—Per Gallon Capacity	1025	1025	2050
Electric Range, Household—Small Burner (60% of connected load)	*	*	2050
Electric Range, Household—Large Burner (60% of connected load)	*	*	4505
Electric Range, Household—Oven	8000	2000	10000
Steam Table—Per Square Foot of Top Surface (35% of connected load)	105	300	405
Plate Warmer—Per Cubic Foot Inside Volume (50% of connected load)	615	0	615
Bakers Oven—Per Cubic Foot Inside Volume (60% of connected load)	1300	500	1800
Frying Griddles—Per Square Foot of Top Surface (60% of connected load)	2160	240	2400
Hot Plates—Per Square Foot of Top Surface (60% of connected load)	*	*	6000
Waffle Baker—Per Section (40% of connected load)	*	*	1365
Toaster—Per Slice (50% of connected load)	945	105	1050
Glass Coffee Maker—Per Section	*	*	1365
Sandwich Grille—Per Square Foot of Area (60% of connected load)	*	*	2750
Fry Kettle—Per Pound Fat Capacity (60% of connected load)	*	*	700
Hair Dryer in Beauty Parlor—600 w	2050	0	2050
Permanent Wave Machine in Beauty Parlor—24-25 w Units	2050	0	2050
Gas Burning Equipment			
Gas Equipment—Dry Heat—No Water Evaporated	90%	10%	100%
Gas Heated Oven—Baking	67%	33%	100%
Gas Equipment—Heating Water—Stewing, Boiling, etc.	50%	50%	100%
Stove, Domestic Type—No Water Evaporated—Per Medium Size Burner	9000	1000	10000
Gas Heated Oven—Domestic Type	12000	6000	18000
Stove, Domestic Type—Heating Water—Per Medium Size Burner	5000	5000	10000
Residence Gas Range—Giant Burner (About 5½ in. Diameter)	*	*	12000
Residence Gas Range—Medium Burner (About 4 in. Diameter)	*	*	9000
Residence Gas Range—Double Oven (Total Size 18 in. x 18 in. x 22 in. High)	*	*	18000
Residence Gas Range—Pilot	*	*	250
Restaurant Range—4 Burners and Oven	*	*	100000
Cast-Iron Burner—Low Flame—Per Hole	*	*	100
Cast-Iron Burner—High Flame—Per Hole	*	*	250
Simmering Burner	*	*	1800
Coffee Urn—Large, 18 in. Diameter—Single Drum	5000	5000	10000
Coffee Urn—Small, 12 in. Diameter—Single Drum	3000	3000	6000
Coffee Urn—Per Gallon of Rated Capacity	500	500	1000
Egg Boiler—Per Egg Compartment	2500	2500	5000
Steam Table or Serving Table—Per Square Foot of Top Surface	400	900	1300
Dish Warmer—Per Square Foot of Shelf	540	60	600
Cigar Lighter—Continuous Flame Type	2250	250	2500
Curling Iron Heater	2250	250	2500
Bunsen Type Burner—Large—Natural Gas	*	*	5000
Bunsen Type Burner—Large—Artificial Gas	*	*	3000
Bunsen Type Burner—Small—Natural Gas	*	*	3000
Bunsen Type Burner—Small—Artificial Gas	*	*	1800
Welsbach Burner—Natural Gas	*	*	3000
Welsbach Burner—Artificial Gas	*	*	1800
Fish-tail Burner—Natural Gas	*	*	5000
Fish-tail Burner—Artificial Gas	*	*	3000
Lighting Fixture Outlet—Large, 3 Mantle 480 C.P.	4500	500	5000
Lighting Fixture Outlet—Small, 1 Mantle 160 C.P.	2250	250	2500
One Cubic Foot of Natural Gas Generates	900	100	1000
One Cubic Foot of Artificial Gas Generates	500	50	550
One Cubic Foot of Producer Gas Generates	135	15	150
Steam Heated Equipment			
Steam Heated Surface Not Polished—Per Square Foot of Surface	330	0	330
Steam Heated Surface Polished—Per Square Foot of Surface	130	0	130
Insulated Surface, Per Square Foot	80	0	80
Bare Pipes, Not Polished Per Square Foot of Surface	400	0	400
Bare Pipes, Polished Per Square Foot of Surface	220	0	220
Insulated Pipes, Per Square Foot	110	0	110
Coffee Urn—Large, 18 in. Diameter—Single Drum	2000	2000	4000
Coffee Urn—Small, 12 in. Diameter—Single Drum	1200	1200	2400
Egg Boiler—Per Egg Compartment	2500	2500	5000
Steam Table—Per Square Foot of Top Surface	300	800	1100
Miscellaneous			
Heat Liberated By Food per person, as in a Restaurant	30	30	60
Heat Liberated from Hot Water used direct and on towels per hour—Barber Shops	100	200	300

*Per cent sensible and latent heat depends upon use of equipment; dry heat, baking or boiling.

Application of the foregoing data in determining cooling load requirements is illustrated in Example 1.

Example 1. Determine cooling load requirements for a clothing store illustrated in Fig. 4 and located in Pittsburgh, Pa., Latitude 40 deg. This is a one-story building located on a corner and it faces south and west. Assume building on east and north sides conditioned.

Wall construction, 8 in. hollow tile, 4 in. brick veneer, plaster on walls, $U = 0.33$ (Table 4, Chapter 3, No. 38 B).

Roof construction, 2 in. concrete, ½ in. rigid insulation, metal lath and plaster ceiling, $U = 0.26$ (Table 11, Chapter 3, No. 2 J).

Floor, maple flooring on yellow pine, no ceiling below, $U = 0.34$ (Table 8, Chapter 3, No. 1 D).

Partition, wood lath and plaster on both sides of studding, $U = 0.34$ (Table 6, Chapter 3, No. 77 B).

Show windows, provided with awnings and thin panel partition at rear.

Front doors, 2 ft 6 in. x 7 ft (glass paneled).

Side door, 3 ft x 7 ft (glass paneled), $U = 1.13$ (Table 13 A, Chapter 3).

Occupancy, 10 clerks, 40 patrons.

Lights, 4200 w.

Outside design conditions, dry-bulb 95 F; wet-bulb 75 F.

Inside design conditions, dry-bulb 80 F; wet-bulb 67 F.

Basement temperature, 85 F.

Store room temperature, 88 F.

Solution. It is obvious from the shape and exposure of this store and the large glass area on the west side that the maximum cooling load will occur during the afternoon when the sun is shining on the west wall. From Fig. 1, the peak load may be expected at 4:00 p.m.

The combined normal transmission and solar radiation transmission through the roof at 4:00 p. m. is obtained from Fig. 2. While none of the roofs in Fig. 2 is exactly like this one, roof *C* is similar. A heat flow of 11 Btu per square foot per hour was assumed,

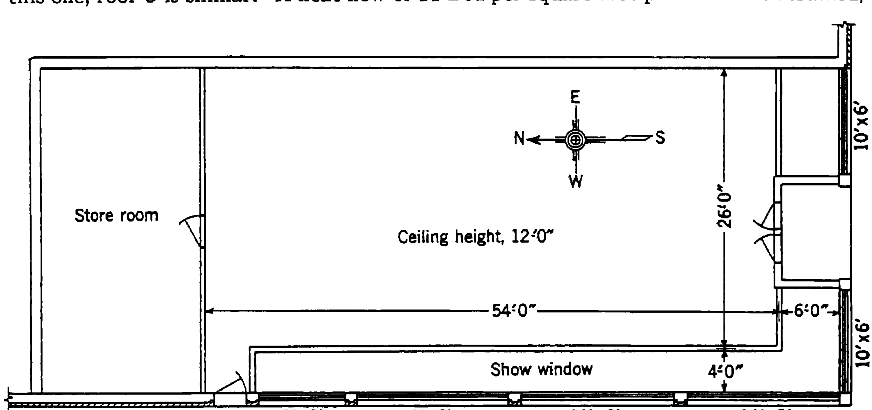


Fig. 4. Plan Diagram of Clothing Store

slightly less than for roof *C*. The combined normal transmission and solar radiation transmission through the south and west walls at 4:00 p.m. is obtained from curves *TS* and *TW* in Fig. 3.

The normal heat transmission through the south glass, floor and partition is determined by application of Formula 1. Solar radiation transmission through the south glass can be neglected. The solar intensity I for the south side at 4:00 p.m. is 29. Applying a shade factor of 0.28, the calculated solar radiation transmission is $29 \times 0.28 = 8$ Btu per square foot per hour which is less than the normal transmission, therefore the total heat gain can be taken as that due to normal transmission.

Solar radiation intensity on the west glass at 4:00 p.m. from Table 4 is 211 Btu per square foot per hour. As explained in the text, normal transmission can be neglected because it is small in comparison with solar radiation transmission.

To determine the heat gain from the outside air it is necessary first to determine the volume of the outside air to be introduced. Since the show windows are sealed so as not to permit infiltration and since there are only three doors in this store through which infiltration can take place, it is obvious that infiltration of air will be a negligible quantity. The volume of the store is 21,600 cu ft. Good practice indicates that in a store of this character there should be a minimum of from 1 to 1½ outside air changes per hour. On a basis of 1½ air changes the volume of outside air to be introduced would be 32,400 cfh. The minimum ventilation requirements as given in the CODE OF MINIMUM REQUIREMENTS FOR COMFORT AIR CONDITIONING[10] are 10 cfm per person. On this basis the ventilation requirements would be 30,000 cfh. Since this will produce approximately 1½ outside air changes per hour, 30,000 cfh will be considered in this application.

To determine load imposed by occupants it will be found from Table 4, Chapter 2 that the average person standing at rest will dissipate 431 Btu per hour and that the moisture dissipated is 0.199 lb per hour.

To determine the latent heat load, the sum of the moisture evaporated from occupants and that to be removed from outside air is multiplied by the latent heat of evaporation at the temperature at which the moisture is condensed in the conditioner. Since outside air is positively introduced, a mixture of outside and recirculated air passes through the conditioner. To remove the moisture, the air must be cooled to a temperature below the dew-point of the mixture. To obtain an approximate value of the latent heat of evaporation, assume that the air is cooled to 55 F. At this temperature, $h_{fg} = 1062.7$ Btu per hour (steam table).

COMBINED NORMAL AND SOLAR RADIATION TRANSMISSION:

SURFACE	DIMENSIONS	AREA SQ FT	BTU PER HOUR PER SQ FT	BTU PER HOUR
S Wall	(30 ft x 12 ft) − 155	205	3	615
W Wall	(60 ft x 12 ft) − 321	399	2.5	998
Roof	60 ft x 30 ft	1800	11	19,800
Total				21,413

NORMAL TRANSMISSION:

SURFACE	DIMENSIONS	AREA SQ FT	U	TEMP. DIFF. DEG F	BTU PER HOUR
S Glass	2 (2 ft 6 in. x 7 ft) + 2 (10 ft x 6 ft)	155	1.13	15	2,627
Floor	26 ft x 54 ft	1404	0.34	5	2,387
N Partition	30 ft x 12 ft	360	0.34	8	979
Total					5,993

[10]Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27). Reprints of this code are available at $.10 a copy.

SOLAR RADIATION THROUGH GLASS:

W Glass. $A_G = 3\,(14\text{ ft} \times 6\text{ ft}) + (8\text{ ft} \times 6\text{ ft}) + (3\text{ ft} \times 7\text{ ft}) = 321$ sq ft

$H_G = 321 \times 0.28 \times 211 = 18{,}965$ Btu per hour (Equation 2).

OUTSIDE AIR:

$H = \dfrac{Q}{v}\,(h_o - h_i)$ (Equation 3).

$v = v_a + \mu v_{as}$ (Equation 17, Chapter 1).

v_a = specific volume of dry air at 95 F = 13.97 cu ft per pound (Table 6, Chapter 1).

v_{as} = difference between volume of saturated mixture and specific volume of dry air at 95 F = 0.82 cu ft per pound (Table 6, Chapter 1).

μ = per cent saturation at 95 F dry-bulb and 75 F wet-bulb = 38.4 per cent (by calculation, Chapter 1).

$v = 13.97 + (0.384 \times 0.82) = 14.28$ cu ft per pound dry air.

$h_o = h_a + \mu h_{as}$ (Equation 19, Chapter 1).

h_a = specific enthalpy of dry air at 95 F = 22.80 Btu per pound (Table 6, Chapter 1).

h_{as} = difference between enthalpy of saturated mixture and specific enthalpy of dry air at 95 F = 40.25 Btu per pound (Table 6, Chapter 1).

$h_o = 22.80 + (0.384 \times 40.25) = 38.26$ Btu per pound dry air.

μ at 80 F dry-bulb and 67 F wet-bulb = 50.2 per cent (by calculation, Chapter 1).

$h_i = h_a + \mu h_{as} = 19.19 + (0.502 \times 24.32) = 31.40$ Btu per pound dry air (Table 6, Chapter 1).

$H = \dfrac{30{,}000}{14.28}\,(38.26 - 31.40) = 14{,}410$ Btu per hour.

W_o = humidity ratio of outside air at 95 F and 75 F = $0.384 \times 0.03652 = 0.01402$ lb water per pound dry air. (Equation 14, Chapter 1).

W_i = humidity ratio of inside air at 80 F and 67 F = $0.502 \times 0.02221 = 0.01115$ lb water per pound dry air. (Equation 14, Chapter 1),

Weight of water to be removed $= \dfrac{Q}{v}\,(W_o - W_i) = \dfrac{30{,}000}{14.28}\,(0.01402 - 0.01115) =$ 6.03 lb per hour.

OCCUPANTS:

$50 \times 431 = 21{,}550$ Btu per hour.

$50 \times 0.199 = 9.95$ lb water per hour evaporated.

LIGHTS:

$4200 \times 3.413 = 14{,}335$ Btu per hour.

SUMMARY:

Component of Load	Btu per Hour
Combined Normal and Solar Radiation Transmission	21,413
Normal Transmission	5,993
Solar Radiation Through Glass	18,965
Outside Air	14,410
Occupants	21,550
Lights	14,335
Total	96,664

LATENT HEAT:

Outside air 6.03 lb water per hour.
Occupants 9.95 lb water per hour.

15.98 lb water per hour.

$15.98 \times h_{fg} = 15.98 \times 1062.7 = 16{,}980$ Btu per hour.

Chapter 7

COMBUSTION AND FUELS

Principles of Combustion, Classification of Coals, Firing Methods for Coals, Firing Methods for Coke, Dustless Treatment of Coal, Classification of Oils, Combustion of Oil, Classification of Gas, Combustion of Gas

THE data given in the first part of this chapter are of general application to the various fuels used in domestic heating which are coal, coke, oil and gas. The choice of fuel is a question of dependability, cleanliness, fuel availability, economy, operating requirements and control.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Combustion may be defined as the chemical combination of a substance with oxygen with a resultant evolution of heat. The rate of combustion depends partly upon the specific rate of reaction of the combustible substance with oxygen and partly upon the rate at which oxygen is supplied and the surrounding conditions as they define the temperature.

Complete combustion is obtained when all of the combustible elements in the fuel are oxidized with all of the oxygen with which they can combine. All of the oxygen supplied may not be utilized.

Perfect combustion is defined as the result of supplying the required amount of oxygen for combination with all of the combustible elements of the fuel and utilizing all of the oxygen so supplied.

The oxygen required for the process of combustion is obtained from air which is a mechanical mixture of oxygen, nitrogen and small amounts of carbon dioxide, water vapor and inert gases. These inert gases are generally included with the nitrogen, and for engineering purposes the values given herewith may be used.

	By Volume Per Cent	By Weight Per Cent
Oxygen, O_2	20.9	23.15
Nitrogen, N_2	79 1	76.85

The combination of oxygen with the combustible elements and compounds of a fuel is in accordance with fixed laws. In the case of perfect combustion the reactions and resultant combinations are shown in Table 1.

The most important condition governing the process of combustion is temperature. It is necessary to bring a combustible substance to its

Table 1. General Data of Combustible Elements and Compounds

Substance	Molecular Symbol	Chemical Reaction of Combustion	Ignition Temperature[a] Deg F	Calorific Value			Theoretical Oxygen and Air Requirements			
				Btu per pound		Btu per Cubic Foot	Lb per Lb		Cubic Ft per Cubic Ft	
				Higher	Lower	Higher	O_2	Air	O_2	Air
Carbon (to CO)	—	$2C + O_2 = 2CO$	—	4380	—	—	1.333	5.76	—	—
Carbon (to CO_2)	—	$2C + 2O_2 = 2CO_2$	—	14540	—	—	2.667	11.52	—	—
Sulphur	S_2	—	—	—	—	—	1.000	4.32	—	—
Sulphur (to SO_2)	—	$S + O_2 = SO_2$	—	4050	—	—	—	—	—	—
Sulphur (to SO_3)	—	$2S + 3O_2 = 2SO_3$	—	5940	—	—	—	—	—	—
Carbon monoxide	CO	$2CO + O_2 = 2CO_2$	1166-1319	4380	—	342	0.572	2.46	0.5	2.391
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1202-1346	23850	21670	1073	4.000	17.28	2.0	9.564
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	763-824	21460	21020	1590	3.077	13.29	2.5	11.955
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	986-1123	21450	20420	1675	3.429	14.81	3.0	14.346
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	986-1123	22230	20500	1883	3.733	16.13	3.5	16.737
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1063-1166	62000	52920	348	8.000	34.56	0.5	2.391
Hydrogen sulphide	H_2S	$2H_2S + 3O_2 = 2H_2O + 2SO_2$	599-608	—	—	—	1.412	6.10	1.5	7.173

[a]From International Critical Tables, 1927.

ignition temperature before it will unite in chemical combination with oxygen to produce combustion. The ignition temperatures for several of the combustible constituents of fuels are presented in Table 1.

HEAT OF COMBUSTION

As previously stated, the process of combustion results in the evolution of heat. The *heat of combustion*, or calorific value, of a fuel is the amount of heat generated by the complete combustion of a unit of the fuel and is constant for a given combination of combustible elements and compounds. The heat of combustion of the several fuel elements and compounds in their *pure* state is given in Table 1.

The reaction of the carbon in the fuel with oxygen may result in the formation of carbon monoxide or carbon dioxide. In burning to carbon monoxide, the carbon is not completely oxidized and, as shown by the data, the heat produced is considerably less than if it were completely oxidized. This fact is of greatest importance in considering the efficiency of combustion.

The calorific value of a fuel is determined by direct measurement of the heat evolved during combustion in a calorimeter. As the calorific value, on a *moisture and ash free* basis, of coal from a given district or mine remains substantially constant the calculation of the calorific value of a particular lot of coal can be made if the lot is analyzed for moisture and ash. From a known reliable calorific value for coal from the same mine or district the calorific value on a *moisture and ash free* basis, often called the *H* value, is calculated from Formula (1).

$$\text{Calorific value, moisture and ash free} = \frac{100 \times \text{Calorific value (as received)}}{100 - (\text{Moisture} + \text{Ash})} \qquad (1)$$

If a *dry* or *moisture free* analysis is used it is necessary to correct for *ash only* to reduce to moisture and ash free basis. From the value obtained by Formula (1) the calorific value for the sample under consideration can be calculated as follows:

$$\text{Calorific value} = \frac{\text{Calorific value (moisture and ash free)} \times [100 - (\text{Moisture} + \text{Ash})]}{100} \qquad (2)$$

In the above formulae moisture and ash are expressed in per cent.

The *H* values for Illinois coals are published[1] and it is to be expected that more data on *H* values for other coals will be available in the future.

As practically all fuels contain hydrogen they produce a certain amount of water vapor as one of the products of combustion. The amount of water vapor produced increases as the hydrogen content of the fuel increases. When the calorific value of a fuel is determined in a calorimeter the water vapor is condensed and the latent heat of vaporization that is given up during the condensation is reported as a portion of the heat value of the fuel. The heat value so determined is termed the *gross* or *higher* heat value and this is what is ordinarily meant when the heat value of a fuel is specified. In burning the fuel, however, the products of

[1]*State Geological Survey Bulletin*, No. 62, Classification and Selection of Illinois Coals.

combustion are not cooled to the dew-point and the higher calorific value cannot be obtained.

FLAME

The appearance of the flame or products of combustion may serve as an approximate measure of the temperatures developed in the combustion process. The luminosity of a flame is caused by the heating to incandescence of unconsumed particles of combustible matter in the gases and the higher the temperature of these particles the whiter the flame. Table 2 gives some approximate flame temperature data.

AIR AND COMBUSTION

The weight of air required for the perfect combustion of a pound of fuel may be determined by use of the ultimate analysis of the fuel as applied to Formulae 3 to 5. The various elements are expressed in percentages by weight.

TABLE 2. FLAME TEMPERATURE DATA

APPEARANCE OF FLAME	TEMPERATURE DEG F
Red, visible in daylight	975
Light red	1832
Orange-red	2012
Orange-yellow	2192
Yellow-white	2372
Bright white	2550

Solid and Liquid Fuels:

$$\text{Pounds air required per pound fuel} = 34.56 \frac{C}{3} + \left(H + \frac{O}{8}\right) + \frac{S}{8} \tag{3}$$

Gaseous Fuels:

$$\text{Pounds air required per pound fuel} = 2.46\, CO + 34.56\, H_2 + 17.28\, CH_4 + 13.29\, C_2H_2 + 14.81\, C_2H_4 + 16.13\, C_2H_6 + 6.10\, H_2S - 4.32\, O_2 \tag{4}$$

When the analysis is given on a volumetric basis the formula is expressed as follows:

$$\text{Cubic feet air required per cubic foot gas} = 2.39\,(CO + H_2) + 9.56\, CH_4 + 11.98\, C_2H_2 + 14.35\, C_2H_4 + 16.74\, C_2H_6 - 4.78\, O_2 \tag{5}$$

Formulae 6 and 7 may be used as approximate methods of determining the theoretical air requirement for any fuel.

$$\text{Pounds air required per pound fuel} = 0.755 \times \frac{\text{Calorific value (Btu per pound)}}{1000} \tag{6}$$

$$\text{Cubic feet air required per unit fuel} = \frac{\text{Calorific value (Btu per unit)}}{100} \tag{7}$$

Approximate values for the theoretical air required for different fuels are given in Table 3.

It is customary to make use of the analysis of the products of combustion to determine the amount of flue gas produced and the actual

amount of air supplied for combustion. The analysis of flue gases has been well described in various publications of the *Bureau of Mines* and in the literature and the details of Orsat manipulation need not be considered in this discussion. (See Chapter 34.)

The weight of dry flue gas per pound of fuel burned is used in combustion loss calculations and may be determined by Formula 8.

$$\text{Pounds dry flue gas per pound fuel} = \frac{11\ CO_2 + 8\ O_2 + 7\ (CO + N_2)}{3\ (CO_2 + CO)} \times C \qquad (8)$$

Values for CO_2, O_2, CO and N_2 are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

TABLE 3. THEORETICAL AIR REQUIREMENTS

Solid Fuel	Pounds Air Per Pound Fuel
Anthracite	9.6
Semi-bituminous coal	11.2
Bituminous coal	10.3
Lignite	6.2
Coke	11.2
Fuel Oil	**Pounds Air Per Gallon Fuel**
Commercial Standard No. 1	102.6
Commercial Standard No. 2	104.5
Commercial Standard No. 3	106.5
Commercial Standard No. 5	112.0
Commercial Standard No. 6	114.2
Gaseous Fuels	**Cubic Feet Air Per Cubic Foot Gas**
Natural gas	10.0
Mixed, natural and water gas	4.4
Carbureted water gas	4.4
Water gas, coke	2.1
Coke oven gas	5.2

EXCESS AIR

Because the real measure of the efficiency of combustion is the relation existing between the amount of air theoretically required for *perfect* combustion and the amount of air actually supplied a method of determining the latter factor is of value. Formula 9 will give reasonably accurate results, for most solid and liquid fuels, for determining the amount of air supplied per pound of fuel.

$$\text{Pounds dry air supplied per pound of fuel} = \frac{3.036\ N_2}{(CO_2 + CO)} \times C \qquad (9)$$

Values for CO_2, CO and N are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

The relationship of the air supplied, as determined from the previous formula, to the theoretical air required indicates the per cent of excess air supplied.

A formula that may be used to determine directly the per cent of *excess air* is expressed:

$$\text{Per cent excess air} = \frac{100\left(O_2 - \frac{CO}{2}\right)}{N_2 \times 0.264 - \left(O_2 - \frac{CO}{2}\right)} \qquad (10)$$

In this formula the symbols represent volumetric percentages of the flue gas constituents as determined by analysis.

The amount of excess air in its relation to the percentage of CO_2 is shown by the curves in Fig. 1 for several fuels. These are approximate values. It should be noted that in hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

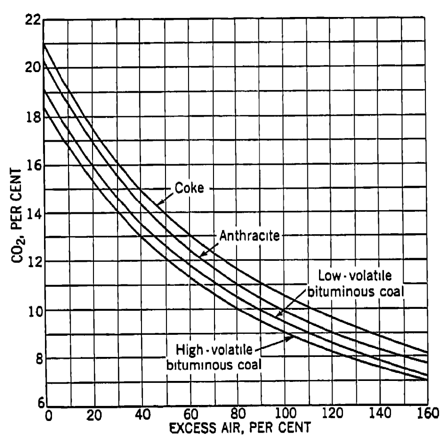


FIG. 1. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

Due to the different carbon-hydrogen ratios of the different fuels the maximum CO_2 attainable varies. Representative values for perfect combustion of several fuels are given in Table 4.

In considering the factor of excess air it should be noted that a deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack. An excess of air is usually required, however, to eliminate combustible losses occasioned by poor mixing of the fuel and air. It is considered good practice, under usual operating conditions, to supply from 25 to 50 per cent excess air, dependent upon the fuel utilized.

SECONDARY AIR

When a solid fuel is hand-fired in a furnace the volatile matter in the fuel distills off leaving coke on the grate. The product of combustion of

the coke is CO_2 and under certain conditions some CO may arise from the bed. The combustion of the volatile matter and the CO may amount to the liberation of from 40 to 60 per cent of the heat in the fuel in the combustion space over the fuel bed.

The air that passes through the fuel bed is called *primary air* and the air that is admitted over the fuel bed in order to burn the volatile matter and CO is called secondary air.

TABLE 4. MAXIMUM CO_2 VALUES

FUEL	PER CENT CO_2
Coke	21.0
Anthracite	20.2
Bituminous coal	18.2
Oil	15.5
Natural gas	12.0
Coke oven gas	11.0

This process of combustion is illustrated in Fig. 2[2]. The free oxygen of the air passes through the grate and the ash above it and burns the carbon in the lower three or four inches of the fuel bed forming carbon dioxide. This layer noted as the oxidizing zone is indicated by the symbols CO_2 and O_2. Some of the carbon dioxide of the oxidizing zone is reduced to carbon monoxide in the upper layer of the fuel bed noted as the reducing zone and indicated by the symbols CO_2 and CO. The gases leaving the fuel

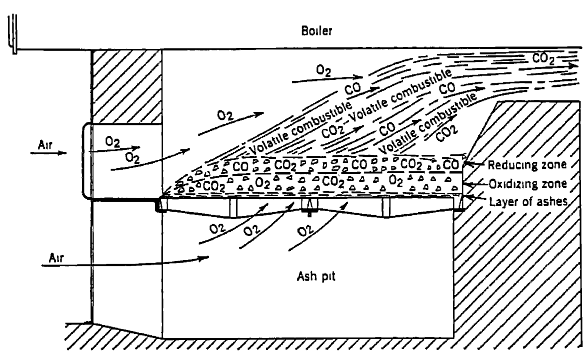


FIG 2. COMBUSTION OF FUEL IN A HAND-FIRED FURNACE

bed are mainly carbon monoxide, carbon dioxide, nitrogen and very little free oxygen. Free oxygen is admitted through the firing door to burn carbon monoxide and the volatile combustible distilled from the freshly fired fuel.

The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size of fuel, depth of fuel bed, and size of fire-pot. The ratio of the secondary to the primary air increases with

[2]From *Bureau of Mines Technical Paper* No. 80.

decrease in the size of the fuel pieces, with increase in the depth of the fuel bed, and with increase in the area of the fire-pot; the ratio also increases with increase in rate of burning.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets through the fuel bed.

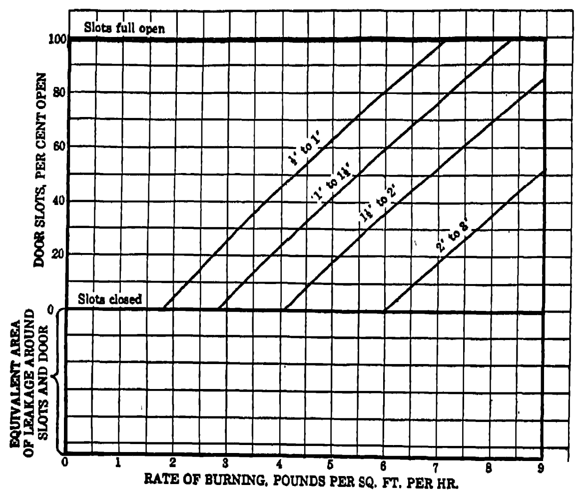


FIG. 3. RELATIVE AMOUNT OF FIRE DOOR SLOT OPENING REQUIRED IN A GIVEN FURNACE TO GIVE EQUALLY GOOD COMBUSTION FOR HIGH TEMPERATURE COKE OF VARIOUS SIZES WHEN BURNED AT VARIOUS RATES

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the effect of slot opening is dependent on whether the ashpit damper is open or closed. It is better to have too much than too little secondary air; the opening is too small if there is a puff of flame when the firing door is opened.

The relationship of the slot opening, for a domestic furnace, to the size of coke and the rate of burning is shown in Fig. 3[3]. These openings are with the ashpit damper wide open, and would be less if the available draft permitted the damper to be partly closed. The same openings are satisfactory for anthracite.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing in order to consume the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is

[3]From *Bureau of Mines Report of Investigations*, No. 2980.

best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial. The following suggestions will be helpful:

1. In cold weather, with high combustion rates, the secondary air damper should be half open all the time.
2. In very mild weather, with a very low combustion rate, the secondary air damper should be closed all the time.
3. For temperatures between very mild and very cold, the secondary air damper should be in an intermediate position.
4. For ordinary house operation, secondary air is needed after each firing for about one hour.

In the field of domestic heating the use of secondary air in the combustion of oil is generally restricted to the larger semi-commercial types of oil burners used in large heating boilers. This factor is discussed in Chapter 9, Automatic Fuel Burning Equipment.

The air that is supplied around the flame in a domestic heating gas burner is considered as secondary air. As it is drawn into the appliance by natural draft action, the need for proper draft control is evident.

Draft Requirements

The draft required to effect a given rate of burning the fuel as measured at the smokehood is dependent on the following factors:

1. Kind and size of fuel.
2. Combustion rate per square foot of grate area per hour.
3. Thickness of fuel bed.
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control by adjustment of the dampers.

The quantity of excess air present has a marked affect on the draft required to produce a given rate of burning. If the excess is caused by holes in the fuel bed or an extremely thin fuel bed it is often possible to produce a higher rate of burning by increasing the thickness of the bed. The thickness of the fuel bed should not, however, be increased too much because the increased draft resistance will reduce the rate of primary air supply and the rate of burning.

DRAFT REGULATION

Because of the varying heating load demands present in most installations it is necessary to vary the rate of fuel burning. The maintenance of the proper air supply for the various rates of burning is accomplished by regulation of the drafts. Correct and incorrect methods of draft regulation are shown in Fig. 4. The air enters through the ashpit, firing door and by leaks in the setting, whereas the gases leave only through the uptake. By throttling the gases with the damper in the uptake all the air entering by each of the three intakes is reduced in the same proportion.

If the ashpit door is closed the air admitted through the ashpit is reduced and increased through the other two intake openings.

Methods of control of draft conditions when burning oil or gas are noted in Chapter 9, Automatic Fuel Burning Equipment.

CLASSIFICATION OF COALS

The complex composition of coal makes it difficult to classify it into clear-cut types. Its chemical composition is some indication but coals having the same chemical analysis may have distinctly different burning characteristics. Users are mainly interested in the available heat per

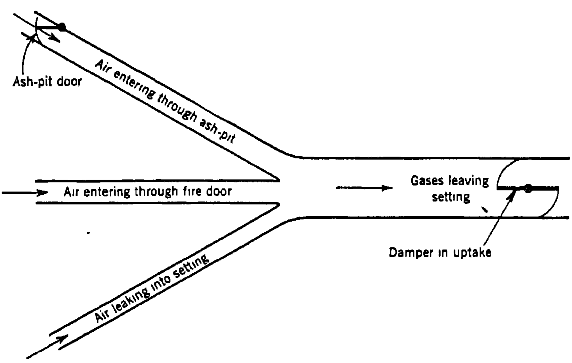


FIG. 4. CORRECT AND INCORRECT METHODS OF DRAFT REGULATION IN A HAND-FIRED FURNACE

pound of coal, in the handling and storing properties, and in the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in *Bureau of Mines Bulletin* No. 276.

The classification of coals by rank involves several of the items indicated in a proximate analysis of the coal. This analysis determines the content of volatile matter, fixed carbon, ash and moisture. The calorific value and sulphur content are often reported with the proximate analysis. Volatile matter is the loss of weight when the coal sample is heated to 1732 F for 7 min. Fixed carbon is the difference from 100 per cent of the sum of other losses, not including sulphur. Ash is the incombustible impurity in the coal and has no heating value. Moisture is the inherent and extraneous water in the fuel.

Other important qualities of coals are the screen sizes, ash softening temperature, friability, caking tendency, and the qualities of the volatile matter. In considering these factors the following points are of interest. The volatile products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash and moisture-free coal, increasing amounts of oils and tars are released. For coals of higher volatile content, the relative quantity of oils and tars decreases, so it

is low in the sub-bituminous coals and in lignite. The percentage of ash and its fusion temperature do not indicate how the ash is distributed or how much of it is less fusible lumps of slate or shale.

A classification of coals is given in Table 5, and a brief description of the kinds of fuels is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other.

Anthracite is a clean, dense, hard coal which creates very little dust in handling. It is comparatively hard to ignite but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires little attention to the fuel bed between firings. It is capable of giving a high efficiency in the common types of hand-fired furnaces. A tabulation of the quality of the various anthracite sizes will be found in *Bureau of Mines Report of Investigations* No. 3283.

Semi-anthracite has a higher volatile content than anthracite. It is not so hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

TABLE 5. CLASSIFICATION OF COALS BY RANK[a]

Legend: F.C. = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units.

Class	Group	Limits of Fixed Carbon or Btu Mineral-Matter-Free Basis	Requisite Physical Properties
I. Anthracite	1. Meta-anthracite	Dry F.C., 98 per cent or more (Dry V.M., 2 per cent or less)	Non-agglomerating[b]
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	
II Bituminous[d]	1. Low volatile bituminous coal	Dry F.C., 78 per cent or more and less than 86 per cent (Dry V.M., 22 per cent or less and more than 14 per cent)	Either agglomerating[b] or non-weathering[f]
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 78 per cent (Dry V.M., 31 per cent or less and more than 22 per cent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 per cent (Dry V.M., more than 31 per cent); and moist[c] Btu, 14,000[e] or more	
	4. High volatile *B* bituminous coal	Moist[c] Btu, 13,000 or more and less than 14,000[e]	
	5 High volatile *C* bituminous coal	Moist Btu, 11,000 or more and less than 13,000[e]	
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[e]	Both weathering and non-agglomerating[b]
	2. Sub-bituminous *B* coal	Moist Btu, 9500 or more and less than 11,000[e]	
	3. Sub-bituminous *C* coal	Moist Btu, 8300 or more and less than 9500[e]	
IV. Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

[a]This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and sub-bituminous ranks. All of these coals either contain less than 48 per cent dry, mineral-matter-free fixed carbon or have more than 15,500 moist, mineral-matter-free Btu.

[b]If agglomerating, classify in low-volatile group of the bituminous class.

[c]Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[d]It is recognized that there may be non-caking varieties in each group of the bituminous class.

[e]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[f]There are three varieties of coal in the High-volatile *C* bituminous coal group, namely, Variety 1, agglomerating and non-weathering; Variety 2, agglomerating and weathering; Variety 3, non-agglomerating and non-weathering.

Adapted from *A.S.T.M.* Standards, 1937, Supplement, p. 145, *American Society for Testing Materials*, Philadelphia.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the more abundant bituminous coals it can be burned with less production of smoke, and it is sometimes called *smokeless coal.*

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which completely melt, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to permit of the screened sizes being delivered free from fines. In general, they ignite easily and burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed.

It is often desirable to learn about the properties of a coal, such as the various items noted in the discussion of proximate analyses. As a guide for the consumer as to the expected characteristics of coals several commercial publications are available and numerous reports of the *Bureau of Mines* discuss the coals produced in individual state areas.

CLASSIFICATION OF COKES

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil.

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke*, which covers the greater part of the coke sold, and *gas-house coke.* The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

Petroleum cokes, which are obtained by coking the residue left from the distillation of petroleum, vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

FIRING METHODS FOR ANTHRACITE[4]

An anthracite fire should never be poked, as this serves to bring ash to the surface of the fuel bed where it melts into clinker.

[4]See reports published by *Anthracite Industries Laboratory*, Primos, Delaware County, Pennsylvania.

Egg size is suitable for large firepots (grates 24 in. and over) if the fuel can be fired at least 16 in. deep. The air spaces between the pieces of coal are large, and for best results this coal should be fired deeply.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The only instructions needed for burning this type of fuel are that the grate should be shaken daily, the fire should never be poked or disturbed, and the fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for fire-pots up to 20 in. in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. It is relatively low in price. When fired carefully, pea coal can be burned on standard grates. It is well to have a small amount of a larger fuel on hand when building new fires, or when filling holes in the fuel bed. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire-door. This keeps a bed of ignited coal in readiness against the time when a sudden demand for heat shall be made on the heater. A very satisfactory method of firing pea coal consists of drawing the red coals toward the front end and piling fresh fuel toward the back of the fire-box.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open and regulating solely by means of the cold air check and the air inlet damper. As a precaution against clinker, it is well to adjust the air inlet damper so that it can never be completely closed under any operating conditions.

Buckwheat size coal for best results requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with pea coal on account of the danger of the fuel falling through the grate. In house heating furnaces the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire it is advisable after coaling to expose a small spot of hot fire by putting a poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent an explosion of gas within the fire-pot, which in some cases depending upon the thickness of the bed of fresh coal is severe enough to blow open the doors and dampers of the furnace. A good draft is required and consequently the fire is best controlled by the air-inlet damper only. Where frequent attention can be given and care exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an

TABLE 6. ANTHRACITE STANDARDS

Classification	Coal Size, Inches	
Egg	Through 3-1/4	Over 2-7/16
Stove	Through 2-7/16	Over 1-5/8
Nut	Through 1-5/8	Over 13/16
Pea	Through 13/16	Over 9/16
Buckwheat	Through 9/16	Over 5/16

easily broken up condition so that it readily can be shaken through the grate.

Forced draft and small mesh grates are frequently used for burning buckwheat anthracite. For best results and a higher degree of convenience, domestic stokers are used.

Buckwheat anthracite No. 2, or rice size, is used principally in stokers of the domestic, commercial and industrial type. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

The *Anthracite Institute* Standards of sizing are shown in Table 6 taken from *Anthracite Industries Manual*, Report No. 2403.

FIRING METHODS FOR BITUMINOUS COAL

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

Air should be admitted over the fire through a special secondary air device, or through a slide in the fire-door or by opening the fire-door slightly. If the quantity of air admitted is too great the gases will be cooled below the ignition temperature and will fail to burn. The fireman can judge the quantity of air to admit by noting when the air supplied is just sufficient to make the gases burn rapidly and smokelessly above the fuel bed.

The red fuel in the fire-box, before firing, excepting only a shallow layer of coke on the grate, should be pushed to one side or forward or backward to form a hollow in which to throw the fresh fuel. Some manufacturers recommend that all red fuel be pushed to the rear of the fire-box and that the fresh fuel be fired directly on the grate and allowed to ignite from the top. The object of this is to reduce the early rapid distillation of gases and to reduce the quantity of secondary air required for smokeless combustion.

It is well to have the bright fuel in the fire-box so placed that the gases from the freshly fired fuel, mixed with the air over the fuel bed, pass over the bed of bright fuel on the way to the flues. The bed of bright fuel then supplies the heat to raise the mixture of air and gas to the ignition temperature, thereby causing the gaseous matter to burn and preventing the formation of smoke.

The importance of firing bituminous coal in small quantities at short intervals is discussed in the *U. S. Bureau of Mines Technical Paper*, No. 80. Better combustion is obtained by this method in that the fuel supply is maintained more nearly proportional to the air supply.

This is demonstrated in Fig. 5 where diagram A shows the air supply and the distillation of the volatile combustible when the firings are 5 min apart; and diagram B indicates the same relationships when the firings are 15 min apart. In both cases the amount of coal fired per hour and the weight of volatile combustible distilled from the coal are the same. This weight of volatile conbustible is represented by the shaded area under the saw-tooth curve. The horizontal dotted lines represent the constant air supply sufficient to burn the volatile matter represented by the shaded areas under each line. The shaded areas above each horizontal line represent for each air supply the loss from incomplete combustion of the volatile matter. The clear area under each horizontal line represents the loss from excessive air. As the air supply increases the loss from incom-

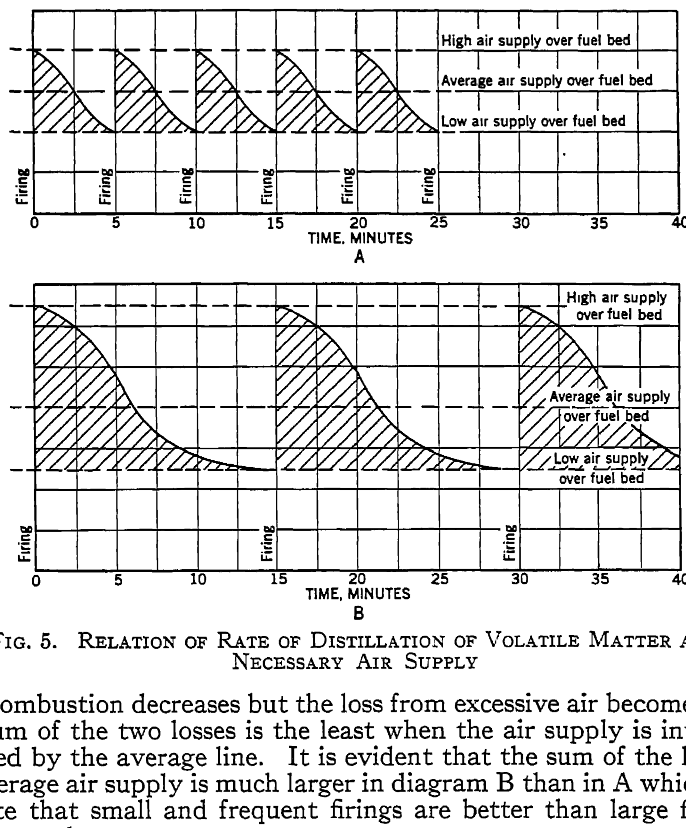


Fig. 5. Relation of Rate of Distillation of Volatile Matter and Necessary Air Supply

plete combustion decreases but the loss from excessive air becomes larger. The sum of the two losses is the least when the air supply is introduced as noted by the average line. It is evident that the sum of the losses for the average air supply is much larger in diagram B than in A which would indicate that small and frequent firings are better than large firings at long intervals.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the fire-box. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be

raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

It is acknowledged that it may be difficult to apply the outlined methods to domestic heating boilers of small size, especially when frequent attendance is impractical. The adherence to these methods insofar as practical, however, will result in better combustion.

The output obtained from any heater with bituminous coal will usually exceed that obtainable with anthracite, since bituminous coal burns more rapidly than anthracite and with less draft. Bituminous coal, however, will require frequent attention to the fuel bed, because it burns unevenly, even though the fuel bed may be level, forming holes in the fire which admit too much air, chilling the gases over the fuel bed and reducing the available draft.

FIRING METHODS FOR SEMI-BITUMINOUS COAL

The *Pocahontas Operators Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

FIRING METHODS FOR COKE

Coke ignites less readily than bituminous coal and more readily than anthracite and burns rapidly with little draft. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds rapidly to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. In order to obtain the same interval of attention as with other fuels a deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. The best size of coke for general use, for small firepots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large firepots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or broken up before using.

PULVERIZED COAL

Although several pulverized coal burning units for domestic heating plant firing have been developed, none has attained extended use. Two general methods of adaptation have been employed, one where the coal is pulverized by the unit at the furnace and one where the coal is delivered to the home in pulverized form.

FURNACE VOLUME

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly, so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires practically no combustion space.

DUSTLESS TREATMENT OF COAL

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with petroleum products, particularly the lighter oils, a solution of calcium chloride or a mixture of calcium and magnesium chlorides. The latter salts are very hygroscopic and their moisture under normal atmospheric conditions keeps the surface of the coal damp, thus reducing the dust during delivery in the cellar, and obviating the necessity of sprinkling the coal in the bin.

The coal is usually treated at the mine, but sometimes by the local distributor just before delivery. The salt solutions are sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size. Oil for the dustless treatment of coal is also applied under high pressure, in concentrations of 1 to 8 qt per ton of coal, depending upon the characteristics of the coal and oil.

CLASSIFICATION OF OILS

The Commercial Standard Specifications for Fuel Oils (CS 12-38) of the *U. S. Department of Commerce* are given in Table 7. These specifications conform with *American Society for Testing Materials* Tentative Specifications for Fuel Oils D396-38T.

The specific gravity of oil is of interest in its relationship to the calorific value and these data are given in Table 8.

COMBUSTION OF OIL

With oil, as with any kind of fuel, efficient heat production requires that all combustible matter in the fuel shall be completely consumed and that it shall be done with a minimum of excess air. The combustion of oil is a rather rapid chemical reaction. Excess air provides an over supply of oxygen so that all of the oil, composed of carbon and hydrogen, will be completely oxidized and thus produce all the heat possible. The use of unreasonable quantities of air in excess of theoretical combustion requirements results in lowered efficiencies due to increased stack losses. Such

TABLE 7. DETAILED REQUIREMENTS FOR FUEL OILS[a]

Grade[b]	Flash Point Deg F		Pour Point Deg F	Water and Sediment Per Cent	Carbon Residue Per Cent	Ash Per Cent	Distillation Temperatures Deg F: 10 Per Cent Point	90 Per Cent Point		End Point	Viscosity Seconds: Saybolt Universal at 100 F		Saybolt Furol at 122 F	
	Min.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Max.	Min.	Max.	Min.
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel.	100 or Legal	165	0[c]	Trace	0.05 on 10% Residuum[d]		410			560[e]				
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel.	110 or Legal	190	10[c]	0.05	0.25 on 10% Residuum[f]		440	600						
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel.	110 or Legal	230	20[c]	0.10	0.15 Straight			675	600[g]		45			
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel.	130 or Legal			1.00		0.10						50	40	
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel.	150			2.00[h]									300	45

[a]Recognizing the necessity for low sulphur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulphur requirement may be specified in accordance with the following table:

Grade of Fuel Oil	Sulphur, Max. Per cent
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No Limit
No. 6	No Limit

Other sulphur limits may be specified only by mutual agreement between the buyer and seller.

[b]It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c]Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any conditions.

[d]For use in other than sleeve type blue flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

[e]The maximum end point may be increased to 590 F when used in burners other than sleeve type blue flame burners.

[f]To meet certain burner requirements the carbon residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

[g]The minimum distillation temperature of 600 F for 90 per cent may be waived if A.P.I. gravity is 26 or lower.

[h]Water by distillation, plus sediment by extraction. Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

losses, if not accompanied by unburned products of combustion (saturated and unsaturated hydrocarbons, hydrogen, etc.) may be offset somewhat by increasing the secondary heating surfaces of the heat absorbing medium boiler or furnace.

Oil is a highly concentrated fuel composed mainly of hydrogen and carbon. In its liquid form oil cannot burn. It must be converted into a gas or vapor by some means. If the excess air is to be kept within efficient limits it means that air must be supplied in carefully regulated quantities. The air and oil vapor must be vigorously mixed to get a rapid and complete chemical reaction. The better the mixing, the less excess air that will be needed. The combustion must take place in a space that maintains the temperatures high so the reaction will not be stopped before completion. When equipped with a means of igniting the oil and safety devices to guard against mishaps, the oil burner possesses all of the elements to be efficient and automatic.

TABLE 8. APPROXIMATE GRAVITY AND CALORIFIC VALUE OF STANDARD GRADES OF FUEL OIL

COMMERCIAL STANDARD NO.	APPROXIMATE GRAVITY, RANGE BAUME	CALORIFIC VALUE BTU PER GALLON
1	38-40	136,000
2	34-36	138,500
3	28-32	141,000
5	18-22	148,500
6	14-16	152,000

CLASSIFICATION OF GAS

Gas is broadly classified as being either *natural* or *manufactured.* Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes. Representative properties of gaseous fuels commonly used in domestic heating are presented in Table 9.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 1000 to 1200 Btu per cubic foot, the majority of natural gases averaging about 1000 Btu per cubic foot. Table 9 shows typical values for the four main oil fields, although values from any one field vary materially.

Table 9 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. However, in any community the variations in gas composition are held within suitable

TABLE 9. REPRESENTATIVE PROPERTIES OF GASEOUS FUELS. BASED ON GAS AT 60 F AND 30 IN. HG.

Gas	Btu per Cu Ft		Specific Gravity, Air = 1.00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature, (Deg Fahr)
					Cubic Feet			Ultimate CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1087	0.67	11.26	1.24	2.24	12.4	12.2	3610
Natural gas—Mid-Continental	967	873	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1.17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1232	1120	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	575	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	588	521	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carbureted water gas	536	496	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	308	281	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	134	124	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

limits so that the performance of approved gas appliances will not be adversely affected.

COMBUSTION OF GAS

The majority of gas burners utilized in central domestic heating plants are of the Bunsen type and operate with a non-luminous flame. In this type of burner part of the air required for combustion is mixed with the gas as primary air, the air and gas mixture being fed to the burner ports. Additional secondary air is introduced around the flame by draft inspiration. In the luminous flame burner, which is sometimes used, all of the air for combustion is brought in contact with the flame as secondary air. The importance of bringing the secondary air into intimate contact with the gas is noted.

Some makes of burners use radiants or refractories to convert some of the energy in the gas to radiant heat. The radiants also serve as baffles in directing the flow of the products of combustion.

The quantity of air given in Table 9 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. The division of the air into primary and secondary

is a matter of burner design and the pressure of gas available, and also of the type of flame desired.

The air gas ratio has a decided effect upon flame propagation. It is necessary that the gas will flow out of the burner ports fast enough so that the flame cannot travel back into the burner head, i.e. *flash back*, but the velocity must not be so high that it blows the flame away from the port.

The maximum and minimum flow speeds from burner ports which may be permitted are known to be very close together when air-gas mixtures in theoretical proportions are being supplied to the burner. As the air-gas ratio is lowered, and the mixture becomes more *gas rich*, the limiting speeds become further apart, until with 100 per cent gas, in an all-yellow flame, flash back cannot occur and a much higher velocity is needed to blow off the flames.

SOOT

The deposit of soot on the flue surfaces of a boiler or heater acts as an insulating layer over the surface and reduces the heat transmission to the water or air; the *Bureau of Mines Report of Investigations* No. 3272 shows that the loss of seasonal efficiency is not as great as has been believed and usually is not over 6 per cent because the greater part of the heat is transmitted through the combustion chamber surfaces. The soot accumulation clogs the passages and reduces the draft; the loss of efficiency from this action may be considerably greater than from the reduction in heat transfer.

REFERENCES

Fuels and Their Combustion, Haslam and Russell (McGraw Hill Co., 1926).

Principles of Combustion in the Steam Boiler Furnace, Arthur D. Pratt, (Babcock and Wilcox Co.).

Smoke-Producing Tendencies in Coals of Various Ranks, by H. J. Rose and F. P. Lasseter (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 329).

Bureau of Mines Publications:

Bulletin No. 97, Sampling and Analyzing Flue Gases, by Henry Kreisinger and F. K. Ovitz.

Report of Investigations (R.I. 2980), Coke as a Domestic Heating Fuel, by P. Nicholls and B. A. Landry.

Bulletin No. 276, Five Hundred Tests of Various Coals in House-heating Boilers, by P. Nicholls, S. B. Flagg and C. E. Augustine.

Report of Investigations (R.I. 3283), Quality of Anthracite as Prepared at Breakers, 1935.

Technical Paper No. 80, Hand Firing Soft Coal under Power-Plant Boilers, by Henry Kreisinger.

Technical Paper No. 303, Value of Coke, Anthracite, and Bituminous Coal for Generating Steam in a Low-pressure Cast-iron Boiler, by John Blizard, James Neil and F. C. Houghten.

Anthracite Industries Laboratories Publications:

Report 2015, Comparison of Sizes, Egg, Stove and Chestnut Anthracite.

Report 2018, Domestic Survey.

Report 2062, Utilization of Anthracite for Domestic Heating.

Report 2204, The Crater Method of Firing.

Report 2403, Anthracite Industries Manual.

An Experimental Investigation of the Use of Oil for the Treatment of Coal, by Ralph A. Sherman and J. M. Pilcher (*A.S.M.E. Transactions*, February, 1938).

Handbook of Oil Burning, by Harry F. Tapp, *American Oil Burner Association.*

Industrial Gas Series, Combustion, *American Gas Association.*

Comfort Heating, *American Gas Association.*

Tests of Gas Home-Heating Equipment, by R. B. Leckie and C. H. B. Hotchkiss (Purdue University, *Engineering Bulletin Research Series* No. 36).

Chapter 8

CHIMNEYS AND DRAFT CALCULATIONS

Natural Draft, Mechanical Draft, Draft Control, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Domestic Chimneys, Construction Details, Chimneys for Gas Heating

DRAFT, in general, may be defined as the pressure difference between the atmospheric pressure and that at any part of an installation through which the gases flow. Since a pressure difference implies a head, draft is a static force. While no element of motion is inferred, yet motion in the form of circulation of gases throughout an entire boiler plant installation is the direct result of draft. This motion is due to the pressure difference, or unbalanced pressure, which compels the gases to flow. Draft is often classified into two kinds according to whether it is created thermally or artificially, such as, (1) natural or thermal draft, and (2) artificial or mechanical draft.

Natural Draft

Natural draft is the difference in pressure produced by the difference in weight between the relatively hot gases inside a natural draft chimney and an equivalent column of the cooler outside air, or atmosphere. Natural draft, in other words, is an unbalanced pressure produced thermally by a natural draft chimney as the pressure transformer and a temperature difference. The intensity of natural draft depends, for the most part, upon the height of the chimney above the grate bar level and also the temperature difference between the chimney gases and the atmosphere.

A typical natural draft system consists essentially of a relatively tall chimney built of steel, brick, or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all the heat has not been extracted. Hot gases are an essential element in the operation of a natural draft system, although inherently a heat balance loss.

A natural draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a nuisance.

Natural draft is quite advantageous in installations where the total loss of draft due to resistances is relatively low and also in plants which have practically a constant load and whose boilers are seldom operated above their normal rating. Natural draft systems have been, and are still being,

employed in the operation of large plants during the periods when the boilers are operated only up to their normal rating. When the rate of operation is increased above the normal rating, some form of mechanical draft is employed as an auxiliary to overcome the increased resistances or draft losses. Natural draft systems are used almost exclusively in the smaller size plants where the amount of gases generated is relatively small and it would be expensive to install and operate a mechanical draft system.

The principal advantages of natural draft systems may be summarized as follows: (1) simplicity, (2) reliability, (3) freedom from mechanical parts, (4) low cost of maintenance, (5) relatively long life, (6) relatively low depreciation, and (7) no power required to operate. The principal

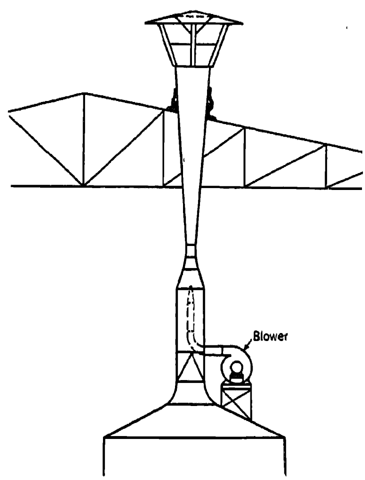


Fig. 1. Diagram of Venturi Ejector

disadvantages are: (1) lack of flexibility, (2) irregularity, (3) affected by surroundings, and (4) affected by temperature changes.

Mechanical Draft

Artificial draft, or mechanical draft, as it is more commonly called, is a difference in pressure produced either directly or indirectly by a forced draft fan, an induced draft fan, or a Venturi chimney as the pressure transformer. The intensity of mechanical draft is dependent for the most part upon the size of the fan and the speed at which it is operated. The element of temperature does not enter into the creation of mechanical draft and therefore its intensity, unlike natural draft, is independent of the temperature of the gases and the atmosphere. The purpose of any mechanical draft system is to produce a difference in pressure between the point at which the air for combustion enters the boiler and the point at which the products of combustion leave the boiler. Such systems include the blower or fan type which produces a plenum or pressure above that

of the atmosphere under the fire and the exhaust fan and Venturi types which produce a partial vacuum that is minimum under the fire, and maximum at the point of exit of the products of combustion from the boiler. The latter types are known as induced draft systems. A mechanical draft system called a Venturi ejector[1] is illustrated in Fig. 1, in which the blower forces air, taken from the outside, through a Venturi tube which draws the gases from the furnace, boiler or hood. With this system, the hot or corrosive gases do not come in contact with the blower. A mechanical draft system may be used either in conjunction with, or as an adjunct to, a natural draft system.

Draft Control

To obtain the maximum efficiency of combustion, a definite minimum supply of air to the combustion chamber must be maintained. To pro-

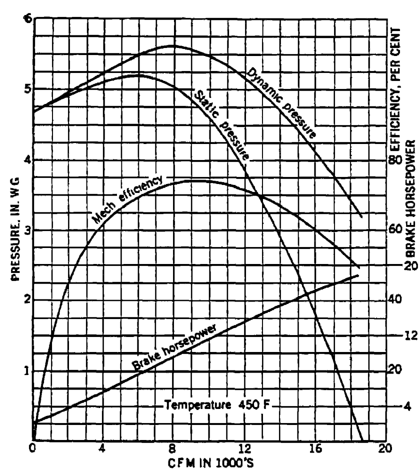


FIG. 2. GENERAL OPERATING CHARACTERISTICS OF TYPICAL INDUCED DRAFT FAN

vide this condition, it is necessary to have some automatic mechanical means of draft control or adjustment, because of variable wind velocities, fluctuations in atmospheric temperatures and barometric pressures, each of which has an effect upon draft.

For this purpose there are various mechanical devices which automatically control the volume of air admitted to the combustion chamber. Mechanical draft regulators designed to control or adjust draft should not be confused with mechanical draft systems that *create* draft mechanically, but which must also be automatically controlled.

The use of such a device, to provide a more uniform and dependable control of draft than could be maintained by manually operated dampers, will produce better combustion of fuel. This higher efficiency of combus-

[1]The Venturi Ejector for Handling Air, by F. F. Kravath (*Heating and Ventilating*, June, p. 17, August, p. 46, 1940).

tion, together with the reduced heat losses up the chimney by reason of decreased gas velocity, results in fuel economy, with consequent lower costs of plant operation.

CHARACTERISTICS OF CHIMNEYS

In order to analyze the performance of a natural draft chimney, it may be advantageous to compare its general operating characteristics with those of a centrifugal pump and also of a centrifugally-induced draft fan, there being a similarity among the three. Figs. 2, 3 and 4 show the general operating characteristics of a typical centrifugally-induced draft fan, a typical centrifugal pump, and a typical natural draft chimney, respectively. The draft-capacity curve of the chimney corresponds to

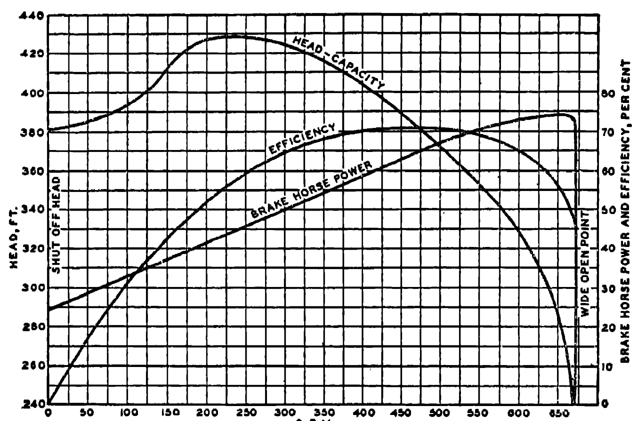


FIG. 3. OPERATING CHARACTERISTICS OF TYPICAL CENTRIFUGAL PUMP

the head-capacity curve of the pump and also to the dynamic-head-capacity curve of the fan.

When the gases in the chimney are stationary, the draft created is termed the *theoretical draft.* When the gases are flowing, the theoretical intensity is diminished by the draft loss due to friction, the difference between the two being termed the total *available draft.*

If pressures at the bases of a column of air and a column of chimney gas, each of height H feet; and d_o and d_c represent the respective densities of the air and the gas in pounds per cubic foot, then the theoretical draft D_t in pounds per square foot is:

$$D_t = d_oH - d_cH$$

Expressing the densities under standard conditions of pressure and temperature, and assuming that the absolute pressure of the gas is the same as that of the air, the theoretical draft becomes:

$$D_t = 15.36\ HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right)$$

Expressed in inches of water this is:

$$D_t = 2.96\ HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right)$$

The friction loss in the chimney may be determined from the Fanning equation:

$$\text{Head lost in feet of fluid} = \frac{fRL}{A} \times \frac{V^2}{2g}$$

in which R is the inside perimeter of the cross-section in feet, A the cross-section area in square feet, and V the velocity of fluid in feet per second.

Substituting for V its value in terms of W and the cross-section area, and expressing the loss of head in inches of water this becomes:

For cylindrical stacks

$$h_L = 0.01936 \frac{fLW^2}{D^5 d_c}$$

and for a rectangular stack of sides x and y in feet,

$$h_L = 0.00597 \frac{fLW^2 (x + y)}{\overline{xy^3}\, d_c}$$

Substituting for d_c its value:

$$\frac{460\, B_o\, W_c}{29.92\, T_c}$$

gives for a cylindrical stack,

$$h_L = 0.00126 \frac{W^2\, T_c fL}{D^5\, B_o\, W_c}$$

and for a rectangular stack,

$$h_L = 0.000388 \frac{W^2\, T_c fL\, (x + y)}{\overline{xy^3}\, B_o\, W_c}$$

The available draft then is, for a cylindrical stack:

$$D_a = 2.96HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126W^2T_cfL}{D^5B_oW_c} \qquad (1)$$

and for a rectangular stack:

$$D_a = 2.96\, HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.000388\, W^2\, T_c fL\, (x + y)}{\overline{xy^3}\, B_o\, W_c} \qquad (2)$$

where

D_a = available draft, inches of water.
H = height of chimney above grate bars, feet.
B_o = barometric pressure corresponding to altitude, inches of mercury.
W_o = unit weight of a cubic foot of air at 0 F and sea level atmospheric pressure, pounds per cubic foot.
W_c = unit weight of a cubic foot of chimney gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
T_o = absolute temperature of atmosphere, degrees Fahrenheit.
T_c = absolute temperature of chimney gases, degrees Fahrenheit.
W = weight of gases generated in the combustion chamber of the boiler and passing through the chimney, pounds per second.
f = coefficient of friction.
L = length of friction duct of the chimney, feet.
D = minimum diameter of chimney, feet.

The first term of the right hand expression of Equation 1 represents the theoretical draft intensity, and the second term, the loss due to friction.

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_0 = 29.92 in. of mercury; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Substituting these values in Equation 1 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^5 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 4 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the

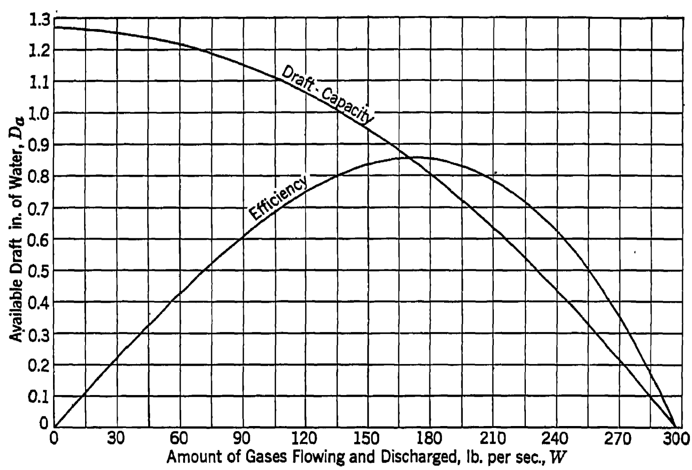


FIG. 4. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available intensity decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The draft-capacity curve corresponds to the head-capacity curve of centrifugal pump characteristics and the dynamic-head-capacity curve of a fan. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 1 and then plotting the results in the manner shown in Fig. 4.

In substituting the values for the various factors in Equation 1, care should be exercised that the selections be as near the actual conditions as

is practically possible. The following notes will serve as a guide for these selections:

1. The *barometric pressure*, represented by B_0, is the actual pressure at the site of the chimney and not the pressure reduced to sea level datum.

In general, the barometric pressure decreases approximately 0.1 in. of mercury per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_c = 0.131 CO_2 + 0.095\, O_2 + 0.083\, N_2 \tag{3}$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_c may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper are disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* decreases from the breeching connection to the top of the stack. This drop in temperature depends upon the material and construction of the stack, its tightness or freedom from leaks, its area, its height, and the velocity of the gases through it. The same chimney will suffer different temperature losses depending upon the capacity under which it is working and the variable atmospheric conditions. No general equation covering all these variables has been suggested, but from observations on chimneys varying in diameter from 3 to 16 ft and in height from 100 to 250 ft the following equation was deduced[2];

$$T_c = \frac{3.13\, T_1 \left[\left(\frac{H_b}{3}\right)^{0.96} - 1\right]}{H_b - 3} \tag{4}$$

where

T_1 = absolute temperature at the center of the connection from the breeching, degrees Fahrenheit.

H_b = the height of the stack above center line connection to breeching, feet.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

In important chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of Reynolds number[3] in determining the friction factor, *f*.

6. The *length of the friction duct* is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

7. Assuming no air infiltration the *amount of gases flowing and being discharged* is, of course, equal to the amount of gases generated in the combustion chamber of the

[2]Notes on Power Plant Design, by E. F. Miller and James Holt (*Massachusetts Institute of Technology*, 1930).

[3]For more complete discussion see Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering*, August, 1933).

boiler. The total products of combustion in pounds per second for a grate-fired boiler may be computed from the equation:

$$W = \frac{C_g G W_{tp}}{3600} \tag{5}$$

where

C_g = pounds of fuel burned per square foot of grate surface per hour.
G = total grate surface of boilers, square feet.
$C_g \times G$ = total weight of fuel burned per hour.
W_{tp} = total weight of products of combustion per pound of fuel.

A similar computation may be made in the case of gas, oil, or stoker-fired fuel.

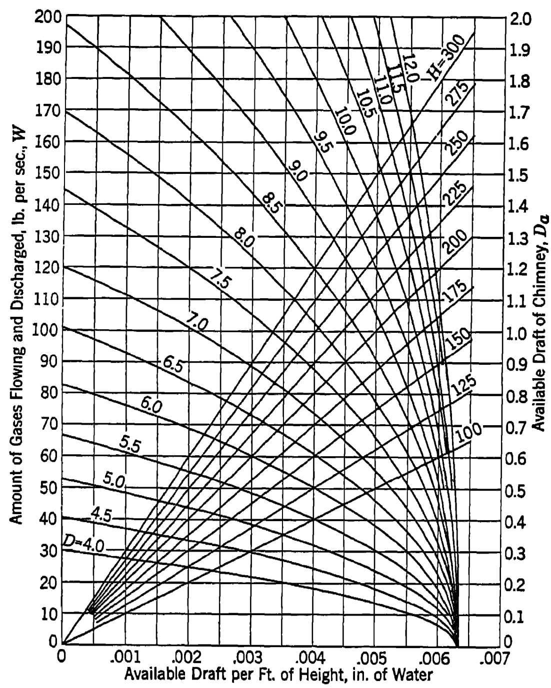


FIG. 5. CHIMNEY PERFORMANCE CHART[a]

[a]*To solve a typical example:* Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale. Starting from a point of Available Draft, take steps in reverse order.

Fig. 5 is a typical chimney performance chart giving the available draft intensities for various amounts of gases flowing and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific conditions, a new chart should be prepared from Equation 1.

It has been the usual custom, and still is to a lamentably great extent, to select the required size of a natural draft chimney from a table of

chimney sizes based only on boiler horsepowers. After the ultimate horsepower of the projected plant had been determined, the chimney size in the table corresponding to this figure was then selected as the proper size required. Generally, no further attempt was made to determine if the height thus selected was sufficient to help create the required draft demanded by the entire installation, or the diameter sufficiently large to enable the chimney quickly, efficiently, and economically to dispose of the gases. Since the operating characteristics of a natural draft chimney are similar in all respects to those of a centrifugal pump, or a centrifugal fan, it is no more possible to select a proper size chimney from such a table, even with correction factors appended, than it is to select the proper size pump from tables based only on the amount of water to be delivered.

DETERMINING CHIMNEY SIZES

The required diameter and height of a natural draft cylindrical chimney are given by the following equations:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{6}$$

The weight of gas per second,

$$W = 12.075\frac{D^2\, VB_o\, W_c}{T_c}$$

from which

$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \tag{7}$$

where

H = required height of chimney above grate bar level, feet.
D = required minimum diameter of chimney, feet (constant for entire height).
V = chimney gas velocity, feet per second.
D_r = total required draft demanded by the entire installation outside of the chimney, inches of water.

Equations 6 and 7 give the required size of a natural draft chimney with all of the operating factors taken into consideration. Values for all of the factors with the exception of the chimney gas velocity may be either observed or computed. It is, of course, necessary to assume an arbitrary value for the velocity in order to arrive at some definite size. For any one set of operating conditions there will be as many sizes of chimney as there are values of reasonable velocities to assume. Of the number of sizes corresponding to the various assumed velocities, there is one size which will be least expensive. Since the cost of a chimney structure, regardless of the kind of material used in the construction, varies as the volume of material in the structure, the cost criterion then may be represented by the approximate equation:

$$Q = \pi tHD \tag{8}$$

where

Q = volume of material, cubic feet.
t = average wall thickness, feet.

For all practical purposes, the value of πt may be taken as a constant regardless of the size of the structure. Hence, in general, the volume, and consequently the cost, of a chimney structure may be based on the factor *HD* as a criterion. Therefore, the value of the chimney gas velocity which will result in the least value of *HD* for any one set of operating conditions will produce a structure which will be the most economical to use, because its cost will be least.

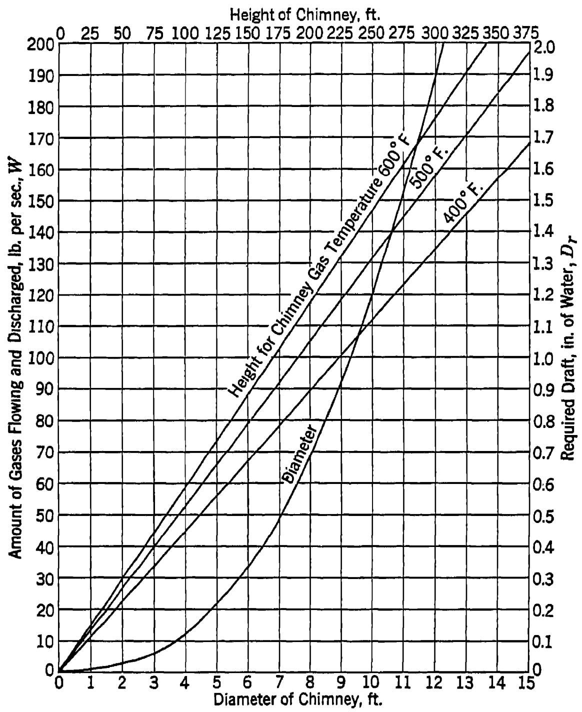


FIG. 6. ECONOMICAL CHIMNEY SIZES[a]

[a]Diameter values also for gas temperatures of 400, 500 and 600 F

The problem is to deduce an equation for the chimney gas velocity which will result in a combination of a height and a diameter whose product *HD* will be least. The solution is obtained by equating the product of Equations 6 and 7 to *HD*, differentiating this product with respect to *V* and equating the resulting expression to zero. This procedure results in the following expression:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{W T_c}{B_o W_c}}}{f W_c} \right)^{2/5} \tag{9}$$

where V_e = economical chimney gas velocity, feet per second.

Equation 9 gives the economical velocity of the chimney gases for any set of operating conditions, and represents the velocity which will

result in a chimney the size of which will cost less than that of any other size as determined by any other velocity for the same operating conditions. After the value of the economical velocity has been determined, the corresponding height and diameter can then be determined from Equations 6 and 7, respectively, and the economical size will then be attained. Equations 6, 7 and 9 may be simplified considerably for average operating conditions in an average size steam plant by assuming typical conditions.

Average chimney gas temperature, 500 F .... $T_c = 960$
Mean atmospheric temperature. 62 F .... $T_o = 522$
Average coefficient of friction, 0.016 .... $f = 0.016$
Average chimney gas density, 0.09 .... $W_c = 0.09$
Sea level elevation, with barometer of 29.92 .... $B_o = 29.92$

Substituting these values in Equations 9, 7 and 6, respectively, and reducing, the results are substantially:

$$V_e = 13.7W^{1/5} \tag{10}$$

$$D = 1.5W^{2/5} \tag{11}$$

$$H = 190D_r \tag{12}$$

Fig. 6 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities as computed from Equations 10, 11 and 12. They are based on the operating factors used in reducing Equations 6, 7 and 9 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 6, 7 and 9. The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

GENERAL EQUATION

The general draft equation for a steam producing plant may be stated as follows:

$$D_t - h_f = h_F + h_B + h_{Bd} + h_C + h_{Br} + h_V + h_O + h_E + h_R \tag{13}$$

where

D_t = theoretical draft intensity created by pressure transformer, inches of water.
h_f = draft loss due to friction in pressure transformer, inches of water.
h_F = draft loss through the fuel bed, inches of water.
h_B = draft loss through the boiler and setting, inches of water.
h_{Br} = draft loss through the breeching, inches of water.
h_V = draft loss due to velocity, inches of water.
h_{Bd} = draft loss due to bends, inches of water.
h_C = draft loss due to contraction of opening, inches of water.
h_O = draft loss due to enlargement of opening, inches of water.
h_E = draft loss through the economizer, inches of water.
h_R = draft loss through recuperators, regenerators, or air heaters, inches of water.

The left hand member of Equation 13 represents the total amount of available draft created by the pressure transformer, that is, the natural draft chimney, Venturi chimney, or fan, and is equal to the theoretical intensity less the internal losses incidental to operation. The right hand member represents the sum of all of the various losses of draft throughout the entire boiler plant installation outside of the pressure transformer itself. The left hand member expresses the available intensity and is analogous to the head developed by a centrifugal pump in a water works system, while the right hand member expresses the required draft in-

Table 1. Recommended Minimum Chimney Sizes for Heating Boilers and Furnaces[a]

Warm Air Furnace Capacity in Sq In. of Leader Pipe	Steam Boiler Capacity Sq Ft of Radiation	Hot Water Heater Capacity Sq Ft of Radiation	Nominal Dimensions of Fire Clay Lining in Inches	Rectangular Flue		Round Flue		Height in Ft Above Grate
				Actual Inside Dimensions of Fire Clay Lining in Inches	Actual Area Sq In.	Inside Diameter of Lining in Inches	Actual Area Sq In.	
790	590	973	8½ x 13	7 x 11½	81			35
1000	690	1,140				10	79	
	900	1,490	13 x 13	11¼ x 11¼	127			
	900	1,490	8½ x 18	6¾ x 16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13 x 18	11¼ x 16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18 x 18	15¾ x 15¾	248			
	2,480	4,090	20 x 20	17¼ x 17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20 x 24	17 x 21	357			
	5,000	8,250	24 x 24	21 x 21	441			55
	5,570	9,190		24 x 24[b]	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24 x 28[b]	672			
	8,700	14,400		28 x 28[b]	784			
	9,380	15,500				27	573	
	10,150	16,750		30 x 30[b]	900			
	10,470	17,250		28 x 32[b]	896			

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

[b]Dimensions are for unlined rectangular flues.

tensity and is analogous to the total dynamic head in a water works system. For a general circulation of gases

$$D_a = D_r \tag{14}$$

where

D_a = available draft intensity, inches of water.
D_r = required draft, inches of water.

The *draft loss through the fuel bed* (h_F), or the amount of draft required to effect a given or required rate of combustion, varies between wide limits and represents the greater portion of the required draft. In coal-fired

installations, the draft loss through the fuel bed is dependent upon the following factors: (1) character and condition of the fuel, clean or dirty; (2) percentage of ash in the fuel; (3) volume of interstices in the fuel bed, coarseness of fuel; (4) thickness of the fuel bed, rate of combustion; (5) type of grate or stoker used; (6) efficiency of combustion.

There is a certain intensity of draft with which the best results will be obtained for every kind of coal and rate of combustion. Fig. 7 gives the intensity of draft, or the vacuum in the combustion chamber required to burn various kinds of coal at various rates of combustion. Expressed in other words, these curves represent the amount of draft required to force the necessary amount of air through the fuel bed in order to effect various rates of combustion. It will be noted that the amount of draft increases as the percentage of volatile matter diminishes, being comparatively low

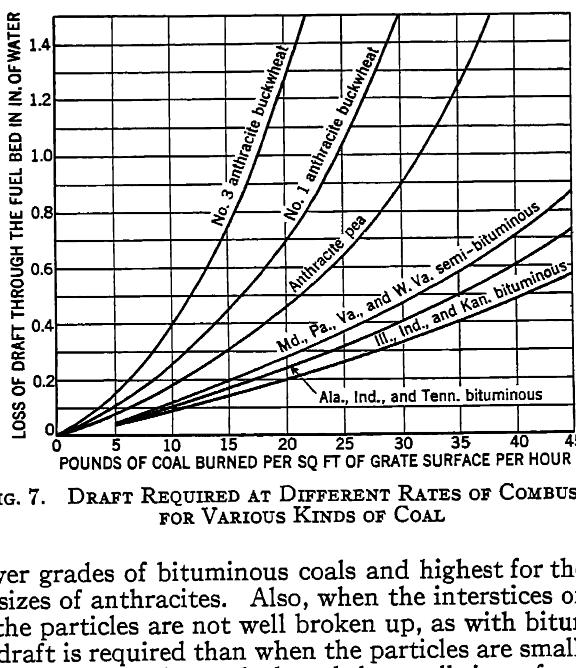


FIG. 7. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

for the lower grades of bituminous coals and highest for the high grades and small sizes of anthracites. Also, when the interstices of the coal are large and the particles are not well broken up, as with bituminous coals, much less draft is required than when the particles are small and are well broken up, as with bituminous slack and the small sizes of anthracites. In general, the draft loss through the fuel bed increases as: (1) the percentage of volatile matter diminishes; (2) the percentage of fixed carbon increases; (3) the thickness of the bed increases; (4) the percentage of ash increases; (5) the volume of the interstices diminishes.

In making the preliminary assumptions for the draft loss through the fuel bed, due allowances should be made for a possible future change in the grade of fuel to be burned and also in the rate of combustion. A value should be selected for this loss which will represent not only the highest rate of combustion which will be encountered, but also the grade of coal which has the greatest resistance through the fuel bed and which may be burned at a later date.

In powdered-fuel and oil-fired installations, there will be no draft loss through the fuel bed since there is none and, consequently, this factor becomes zero in the general draft equation. All other factors being constant, the height of the chimney in installations of this character will be less than the height in coal-fired installations, and in the case of mechanical draft installations the driving units need not be as large since the head against which the fan is to operate is not as great in the former as in the latter.

The *draft loss through the boiler and setting* (h_b) also varies between wide limits and, in general, depends upon the following factors: (1) type of boiler, (2) size of boiler, (3) rate of operation, (4) arrangement of tubes, (5) arrangement of baffles, (6) type of grate, (7) design of brickwork setting, (8) excess air admitted, and (9) location of entrance into breeching.

Curves showing the draft loss through the boiler are usually based on the load or quantity of gases passing through the boiler, expressed in terms of percentage of normal rate of operation. Owing to the great variety of boilers of different designs and the various schemes of baffling, it is impossible to group together a set of curves for the draft loss through the boiler which may even be used generally. It is therefore necessary to secure this information from the manufacturer of the particular type of boiler and baffle arrangement under consideration.

When a boiler is installed and in operation, the draft loss depends upon the amount of gases flowing through it. This, in turn, depends upon the proportion of excess air admitted for combustion. Primarily, the amount of excess air is measured by the CO_2 content; the less the amount of CO_2, the greater the amount of excess air and hence the greater the draft loss.

The loss of draft through the boiler will vary directly as the size of the boiler and the length of the gas passages within. The loss also varies as the number of tubes high, but not in a direct ratio inasmuch as the loss due to the reversal of flow at the ends of the baffles remains constant regardless of the height of the boiler. The arrangement of the tubes, whether the gases flow parallel to or at right angles to the tubes, has an appreciable effect on the loss. The arrangement of the baffles influences the draft loss greatly, the loss through a boiler with five passes being greater than the loss through one of three or four passes. A poor design and a rough condition of the brickwork will increase the loss greatly, whereas a proper design and a smooth condition will keep the loss at a minimum. The loss through the boiler will be less when the breeching entrance is located at or near the top of the boiler than when it is located at or near the bottom since the gases have a shorter distance to travel in the former instance.

The *draft loss through the breeching* (h_{Br}) may be found by applying the last term on the right, with the sign changed, of Equation 1 or 2 depending upon whether the breeching is cylindrical or rectangular and observing the following changes in the symbols:

T_C = absolute temperature of breeching gases, degrees Fahrenheit.
f = coefficient of friction for the breeching.
L = length of breeching, feet.
D = diameter of cylindrical breeching, feet.
x and y = sides of breeching, if rectangular, feet.

It has been the general custom to *lump off* the intensity of the breeching loss at 0.10 in. of water per 100 ft of breeching length regardless of its size or shape or the amount and temperature of the gases flowing through it. This practice is hazardous and has no more foundation in fact than that of determining the friction head in a water works system without taking into consideration the size of the pipe or the amount of water flowing through it. When the length of the breeching is relatively short, any variation in any one of the factors in the equation will have no appreciable effect on the draft loss. However, when the breeching is relatively long, the draft loss is affected greatly by the various factors, particularly by the size and shape as well as by the weight of gases flowing.

The *draft loss due to velocity* (h_V) is given by the equation

$$h_V = \frac{0.000194 W^2 T_c}{A^2 B_O W_c} \tag{15}$$

where

A = cross-section area at the top of the stack, square feet.

The *draft loss due to bends in the breeching* (h_{Bd}) is dependent upon the center line radius of curvature of the bends and the form of the cross-section. This loss is expressed in terms of the velocity head. (See Fig. 4, Chapter 31.)

The *draft loss due to sudden contraction of an area* (h_C) is given by the equation:

$$h_C = \frac{0.000194 K_c W^2 T_c}{A_s^2 B_O W_c} \tag{16}$$

where

K_c = coefficient of sudden contraction based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= 0.5\left(1 - \frac{A_s}{A_1}\right)$

A_s = area of the smaller section.

When the flue or passage through which the gases flow is suddenly contracted, a considerable portion of the static head in the larger section is converted into velocity head and a draft loss of some consequence, particularly in a short breeching, takes place. A sudden contraction should always be avoided where possible. At times, however, due to obstructions or limited head-room, it is necessary to alter the size of the breeching, but a sudden contraction may be avoided by gradually decreasing the area over a length of several feet.

The *draft loss due to a sudden enlargement of an area* (h_O) is given by the equation:

$$h_O = \frac{0.000194 K_o W^2 T_c}{A_s^2 B_O W_c} \tag{17}$$

where

K_o = coefficient of sudden enlargement based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= \left(1 - \frac{A_s}{A_1}\right)^2$

When the flue or passage through which the gases flow is suddenly enlarged, a portion of the velocity head is converted into static head in the

larger section and, like the loss due to sudden contraction, a loss of some consequence, particularly in short breechings, takes place. A sudden enlargement in a breeching may be avoided by gradually increasing the area over a length of several feet. In large masonry chimneys, the area of the flue at the region of the breeching entrance is considerably larger than the area of the breeching at the chimney, and a sudden enlargement exists.

The *draft loss through the economizer* (h_E) should be obtained from the manufacturer but for general purposes it may be computed from the following general equation:

$$h_E = \frac{6.6 W_n^2 N T_c}{10^{12}} \tag{18}$$

where

W_n = pounds of gases flowing per hour per linear foot of pipe in each economizer section.

N = number of economizer sections.

An economizer in a steam plant affects the draft in two ways, (1) it offers a resistance to the flow of gases, and (2) it lowers the average chimney gas temperature, thereby decreasing the available intensity. In the case of a natural draft installation, both of these factors result in a relative increase in the height of the chimney and, in the case of a large plant, they may add as much as 20 or 30 ft to the height. The decrease in the temperature of the gases after they have passed through the economizer has an extremely important effect on the performance of a natural draft chimney and also upon the performance of a fan.

GENERAL CONSIDERATIONS FOR DOMESTIC CHIMNEYS

The draft of domestic chimneys may be subject to a variety of influences not usually encountered in power chimneys, such as buildings in the immediate vicinity which may be higher than the chimney, trees, and even hills as well as the shape of the roof of the building which the chimney serves.

Horizontal winds have an aspirating effect as they pass across the chimney and are an aid to draft providing they remain horizontal at the chimney top. However, surrounding objects, such as trees or other buildings, may greatly affect the direction of the wind at the chimney top and may even direct it down the chimney, tending to reduce the draft or even to cause it to be negative. Although the chimney should, in general, extend well above the highest part of the roof, it is impracticable to carry it much beyond this point.

It is also important to consider the source of the air supply for proper combustion. Usually the boiler or furnace is located in the basement or cellar and perhaps, as a general thing, when the furnace room has windows or doors opening to the outside on two or more sides of the house, the leakage of air around the windows and doors will be sufficient for combustion, even though the windows and doors may be shut. However, if the leakage is not sufficient to prevent an appreciable drop of pressure in the furnace room below that of the air outside, the chimney draft will be reduced by the difference between the atmospheric pressure outside and

that inside the boiler room. In case the boiler room is fairly tight against leaks, as is desirable from the standpoint of dust, and is open to the outside on only one side of the house, then the draft will be affected in windy weather even with windows or doors open. If the wind is blowing toward the boiler room the draft will be increased, but if blowing in the opposite direction the draft may be seriously decreased.

It is not to be assumed that increasing the cross-section area of a chimney will always effect a cure for poor draft. The opposite result may be experienced because of the cooling effect of the larger area. This reduces the theoretical draft and the velocity of the gases, and affords a greater opportunity for counter currents in the chimney. The only practical remedy for a chimney with bad draft, when the chimney is of the proper size and is affected by conditions beyond the control of the owner, is to resort to mechanical draft. This can usually be done at small expense if operated only when necessary.

CHIMNEYS FOR GAS HEATING

The burning of gas differs from the burning of coal in that the force which supplies the air for combustion of the gas comes largely from the pressure of the gas in the supply pipe, whereas air is supplied to a bed of burning coal by the force of the chimney draft. If, with a coal-burning boiler, the draft is poor, or if the chimney is stopped, the fire is smothered and the combustion rate reduced. In a gas boiler or furnace such a condition would interfere with the combustion of the gas, but the gas would continue to pass to the burners and the resulting incomplete combustion would produce a dangerous condition. In order to prevent incomplete combustion from insufficient draft, all gas-fired boilers and furnaces should have a back-draft diverter in the flue connection to the chimney.

A study of a typical *back-draft diverter* shows that partial or complete chimney stoppage will merely cause some of the products of combustion to be vented out into the boiler room, but will not interfere with combustion. In fact, gas-designed appliances must perform safely under such a condition to be approved by the *American Gas Association* Laboratory. Other functions of the back-draft diverter are to protect the burner and pilot from the effects of down-drafts, and to neutralize the effects of variable chimney drafts, thus maintaining the appliance efficiency at a substantially constant value.

As is the case with the complete combustion of almost all fuels, the

TABLE 2. MINIMUM ROUND CHIMNEY DIAMETERS FOR GAS APPLIANCES (INCHES)

Height of Chimney Feet	Gas Consumption in Thousands of Btu per Hour								
	100	200	300	400	500	750	1000	1500	2000
20	4.50	5.70	6.60	7.30	8.00	9.40	10.50	12.35	13.85
40	4.25	5.50	6.40	7.10	7.80	9.15	10.25	12.10	13.55
60	4.10	5.35	6.20	6.90	7.60	8.90	10.00	11.85	13.25
80	4.00	5.20	6.00	6.70	7.35	8.65	9.75	11.50	12.85
100	3.90	5.00	5.90	6.50	7.20	8.40	9.40	11.00	12.40

products of combustion for gas are carbon dioxide (CO_2) and water vapor with just a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron oxide catalyst. The volume of water vapor in the flue products is about twice the volume of the carbon dioxide when coke oven or natural gas is burned. Because of the large quantity of water vapor which is formed by the burning of gas, it is quite important that all gas-fired central heating plants be connected to a chimney having a good draft. Lack of chimney draft causes stagnation of the products of combustion in the chimney and results in the condensation of a large amount of the water vapor. A good chimney draft draws air through the openings in the back-draft diverter, lowers the dew-point of the mixture, and reduces the tendency of the water vapor to condense.

The flue connections from a gas-fired boiler or furnace should be of a non-corrosive material. The material used for the flue connection should not only be resistant to the corrosion of water but should resist the corrosion of dilute solutions of sulphur trioxide. Local practice should be followed in the selection of the most appropriate flue materials.

When condensation in a chimney proves troublesome, it may be necessary to provide a drain to a dry well or sewer. The cause of the excessive condensation should be investigated and remedied if possible. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt-chromate provides an excellent protection.

A chimney for a gas-fired boiler or furnace should be constructed similarly to the principles applicable to other boilers. Table 2 gives the minimum cross-sectional diameters of round chimneys (in inches) for various amounts of heat supplied to the appliance, and for various chimney heights. This is in accordance with *American Gas Association* recommendations.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Standard Ordinance for Chimney Construction of the *National Board of Fire Underwriters*. Briefly summarized, these provisions are as follows for heating boilers and furnaces:

The walls of brick chimneys shall be not less than 3¾ in. thick (width of a standard size brick) and shall be lined with fire-clay flue lining meeting the standard specification of the *Eastern Clay Products Association*. The flue sections shall be set in special mortar, and shall have the joints struck smooth on the inside. The masonry shall be built around each section of lining as it is placed, and all spaces between masonry and linings shall be completely filled with mortar. No broken flue lining shall be used. Flue lining shall start at least 4 in. below the bottom of smoke-pipe intakes of flues, and shall be continued the entire heights of the flues and project at least 4 in. above the chimney top to allow for a 2 in. projection of lining.

Flue lining may be omitted in brick chimneys, provided the walls of the chimneys are not less than 8 in. thick, and that the inner course shall be a refractory clay brick. All brickwork shall be laid in spread mortar, with all joints push-filled. Exposed joints both inside and outside shall be struck smooth. No plaster lining shall be permitted.

Chimneys shall extend at least 3 ft above flat roofs and 2 ft above the ridges of peak roofs when such flat roofs or peaks are within 30 ft of the chimney. The chimney shall be high enough so that the wind from any direction shall not strike the top of the chimney from an angle above the horizontal. The chimney shall be properly capped, but no such cap or coping shall decrease the flue area.

Chapter 9

AUTOMATIC FUEL BURNING EQUIPMENT

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Classification of Oil Burners, Combustion Chamber Design, Classification of Gas-Fired Appliances

AUTOMATIC mechanical equipment for the combustion of solid, liquid and gaseous fuels is considered in this chapter.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed and positive control of the air supplied for combustion.

Stokers may be divided into four types according to their construction, namely, (1) overfeed flat grate, (2) overfeed inclined grate, (3) underfeed side cleaning type, and (4) underfeed rear cleaning type.

Overfeed Flat Grate Stokers

This type is represented by the various chain- or traveling-grate stokers. These stokers receive fuel at the front of the grate in a layer of uniform thickness and move it back horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the stoker to maintain ignition of the incoming fuel. Frequently, a rear combustion arch is required to maintain ignition until the fuel is fully consumed. A typical traveling-grate stoker is illustrated in Fig. 1.

Another and distinct type of overfeed flat-grate stoker is the spreader (Figs. 2 and 3) or sprinkler type in which coal is distributed either by rotating paddles or by air over the entire grate surface. This type of

stoker has a wide application on small sized fuels and on fuels such as lignites, high-ash coals, and coke breeze.

Overfeed Inclined Grate Stokers

In general the combustion principle is similar to the flat-grate stoker, but this stoker (Fig. 4) is provided with rocking grates set on an incline to advance the fuel during combustion. Also this type is provided with an

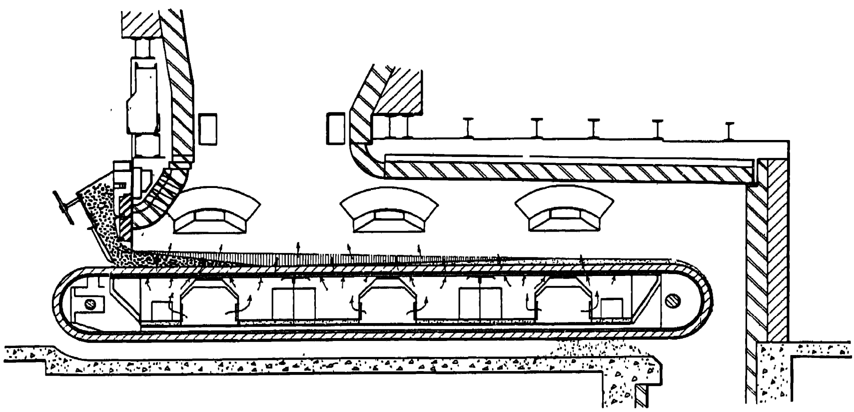


FIG. 1. OVERFEED TRAVELING-GRATE STOKER

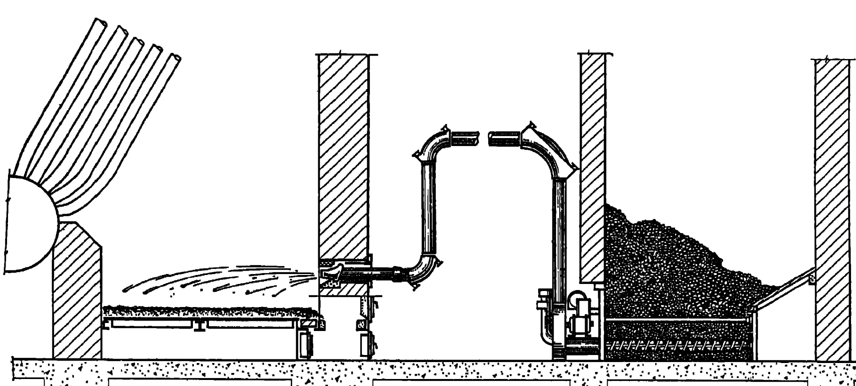


FIG. 2. OVERFEED SPREADER STOKER, PNEUMATIC TYPE

ash plate where ash is accumulated and from which it is dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action has the tendency to keep the fuel bed well broken up thereby allowing for free passage of air. Because of its agitating effect on the fuel it is not so desirable for badly clinkering coals. Furthermore, it should usually be provided with a front arch to care for the volatile gases.

Underfeed Side Cleaning Stokers

In this type (Fig. 5), the fuel is introduced at the front of the furnace to one or more retorts, and is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of

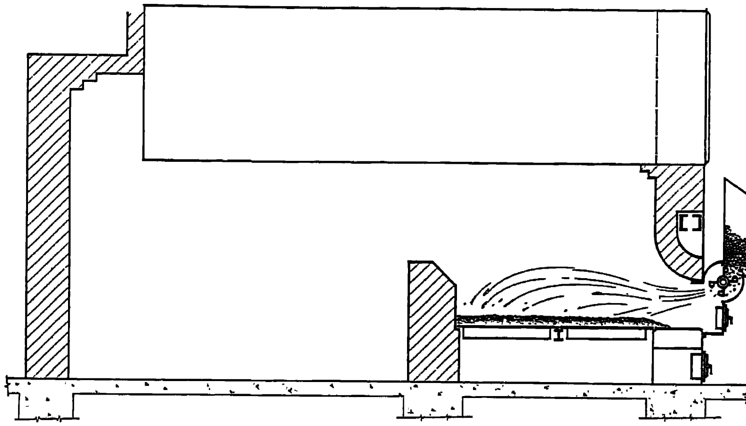


FIG. 3. OVERFEED SPREADER STOKER, ROTOR TYPE

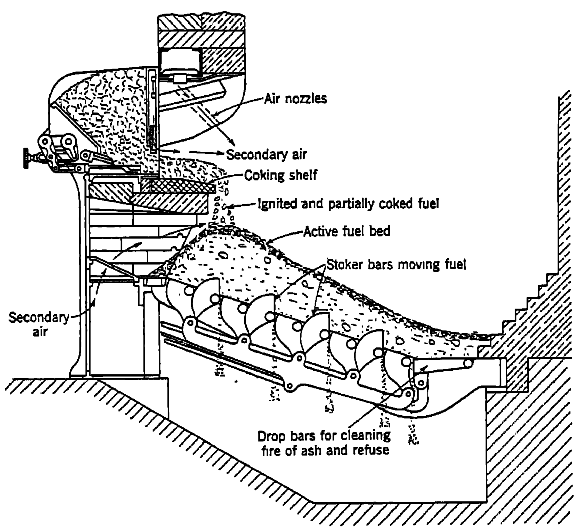


FIG. 4. OVERFEED INCLINED GRATE STOKER

stoker is suitable for all bituminous coals while in the smaller sizes it is suitable for small sizes of anthracite. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously

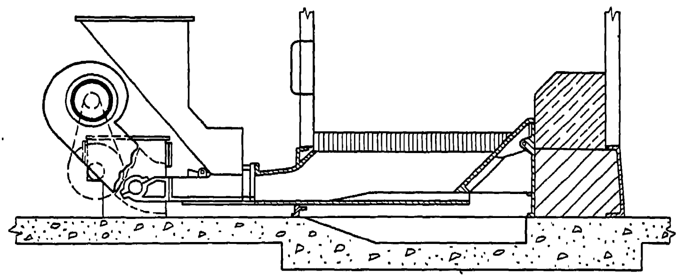


FIG. 5. UNDERFEED SIDE CLEANING STOKER

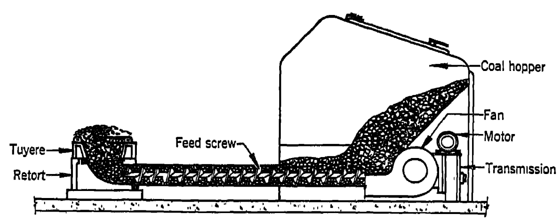


FIG. 6. UNDERFEED STOKER, HOPPER TYPE, CLASS 1

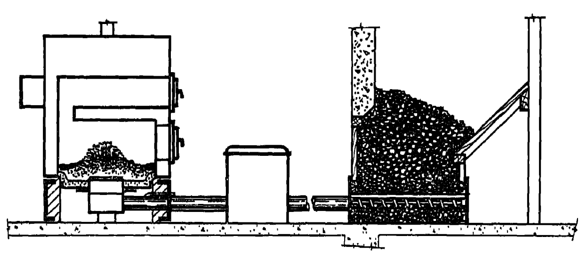


FIG. 7. UNDERFEED STOKER, BIN FEED TYPE, CLASS 1

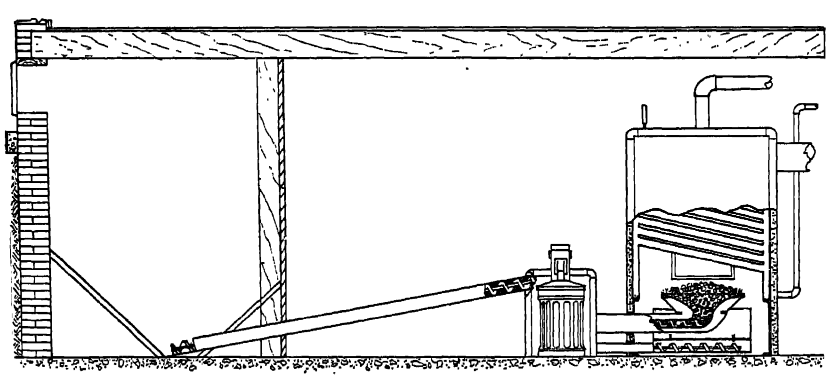


FIG. 8. UNDERFEED ANTHRACITE STOKER WITH AUTOMATIC ASH REMOVAL, BIN TYPE

discharged as in the small stoker or may be accumulated and periodically discharged. This stoker requires no arch as it automatically provides for the combustion of the volatile gases.

Underfeed Rear Cleaning Stokers

This type of stoker accomplishes combustion in much the same manner as the side cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed.

CLASSIFICATION OF STOKERS ACCORDING TO CAPACITY

Stokers may be classified according to their capacity or coal feeding rates. The following classification has been made by the *U. S. Department of Commerce*, in cooperation with the *Stoker Manufacturers Association*.

Class 1. Capacity under 61 lb of coal per hour.
Class 2. Capacity 61 to 100 lb of coal per hour.
Class 3. Capacity 101 to 300 lb of coal per hour.
Class 4. Capacity 300 to 1200 lb of coal per hour.
Class 5. Capacity 1200 lb of coal per hour and over.

Class 1 Stokers

These stokers are used primarily for home heating and are, therefore, designed for quiet, automatic operation. Simple, trouble-free construction and attractive appearance are very desirable characteristics on these small units. Equipment of this capacity is also used extensively for commercial and industrial applications requiring the burning of small quantities of fuel under automatic control.

A common type of stoker in this class (Fig. 6) consists essentially of a coal reservoir or hopper, a screw for conveying the coal from the reservoir to the burner head or retort, a fan which supplies the air for combustion, a transmission for driving the coal feed worm, and an electric motor or motors for supplying the motive power for both coal feed and air supply.

Air for combustion is admitted to the fuel through tuyeres at the top of the retort and in this class, the tuyeres and retort are usually round, although they may be either round or rectangular. Stokers in this class are made for burning anthracite, bituminous, semi-bituminous, and lignite coals and coke. The *U. S. Department of Commerce* has issued commercial standards for household anthracite stokers[1].

Units are available in either the hopper type, as shown in Fig. 6, or in the type as shown in Figs. 7 and 8, which feeds the coal directly to the furnace from the coal bin. Some stokers, particularly those designed for use with anthracite coal, have equipment for automatically removing the ash from the ash pit and depositing it in an ash receptacle outside of the furnace, as shown in Fig. 8. Most of the bituminous models, however, operate on the principle of removing the ash from the fuel bed after it is fused into a clinker at the outer periphery of the tuyere.

Most of the stokers in this class feed coal to the furnace intermittently

[1]Household Anthracite Stoker Standards (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS48-40).

in accordance with temperature or pressure demands. A small amount of heat is also released from the fuel bed during the inoperative period of the stoker. Through the use of automatic controls (see Chapter 33), it is possible to maintain temperatures or pressures within very close limits when using stoker firing equipment.

Stoker-Fired Boiler and Furnace Units

Boilers, air conditioners, and space heaters especially designed for stokers are now available having certain design features closely coordi-

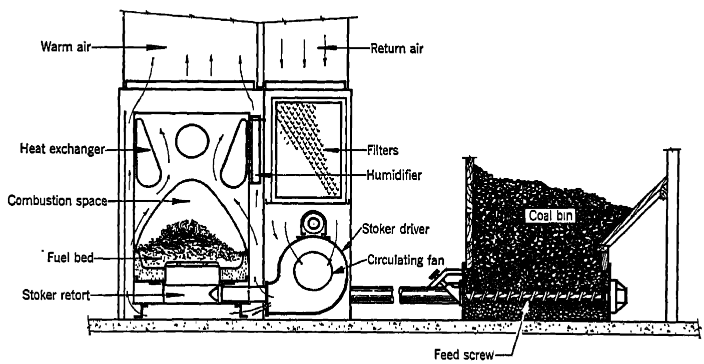


FIG. 9. STOKER-FIRED WINTER AIR CONDITIONING UNIT

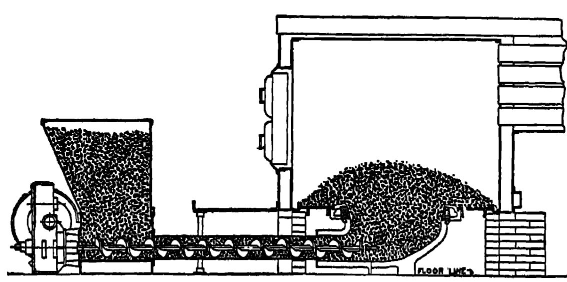


FIG. 10. UNDERFEED SCREW STOKER, HOPPER TYPE, CLASS 2, 3 OR 4

nating the heat absorber and the stoker. Although very efficient and satisfactory performance can be obtained from the application of automatic stokers to existing boilers and furnaces, some of the combination stoker fired units (Fig. 9) are more compact and attractive in appearance in addition to other design features.

Class 2 and 3 Stokers

Stokers in this class are usually of the screw feed type without auxiliary plungers or other means of distributing the coal. They are used extensively for heating plants in apartments and hotels, also, for industrial plants, such as laundries, bakeries, and creameries.

They are primarily of the underfeed type and are available in both the hopper type, as illustrated in Fig. 10, and also, the bin feed type, which

delivers the coal directly to the furnace from the coal bin, as illustrated in Fig. 11. These units are also built in a plunger feed type and the drive for the coal feed may be an electric motor or a steam or hydraulic cylinder.

Stokers in this class are available for burning all types of anthracite, bituminous and lignite coals. The tuyere and retort design varies widely according to the fuel and load conditions. On the bituminous models, the grates are normally of the stationary type and the ash accumulates on the grates surrounding the retort. With the average bituminous coal, the ash then fuses into a clinker which is removed periodically.

The anthracite stokers in this class are normally equipped with moving grates which discharge the ash into a pit below the grate. This ash pit may be located on one or both sides of the grate and on some installations is made of sufficient capacity to hold the ash from several days or weeks operation.

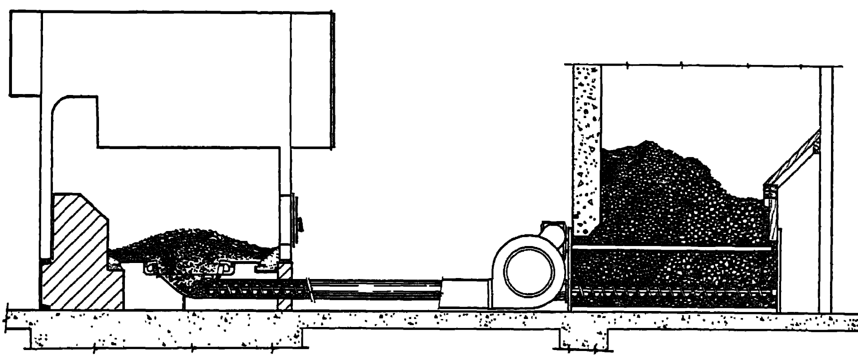


FIG. 11. UNDERFEED SCREW STOKER, BIN TYPE, CLASS 2, 3 OR 4

Class 4 Stokers

Stokers in this group vary widely in details of mechanical design and the several methods of feeding coal previously described are employed. The underfeed stoker is the most widely used, although a number of the overfeed types are also used in the larger sizes of this class. Bin feed, as well as hopper models, are available in both the underfeed and overfeed types.

Class 5 Stokers

The prevalent stokers in this field are: (*a*) overfeed flat grate, (*b*) overfeed inclined grate, (*c*) underfeed side cleaning, and (*d*) underfeed rear cleaning.

The rear cleaning underfeed stoker is usually of the multiple retort design and is used in some of the largest industrial plants and central power stations. In some instances, zoned air control has been applied on these stokers, both longitudinally and transversely of the grate surface.

Underfeed side cleaning stokers are made in sizes up to approximately 500 boiler horsepower. They are not so varied in design as those in the

smaller classes, although the principle of operation is much the same. The overfeed spreader type stoker (Figs. 2 and 3) is adaptable to a wide variety of coals and is being extensively used in capacities up to 1000 boiler horsepower.

Combustion Process

Due to the marked differences in design and operating characteristics of stokers and the widely different characteristics of stoker coals, it is difficult to generalize on the subject of combustion in automatic stokers.

In anthracite stokers of the small Class 1 overfeed type, burning takes place entirely within the stoker retort and tuyere. The ash and refuse of combustion spills over the edge of the tuyere into an ash pit or receptacle from which it may be removed either manually or automatically (Fig. 8).

The larger underfeed anthracite stokers operate on the same principle except that the retort is rectangular and the refuse spills over only one or two sides of the grate. Anthracite for stoker firing is usually supplied in the No. 1 buckwheat or No. 2 buckwheat size.

Since the majority of the smaller bituminous coal stokers operate on the underfeed principle, a general description of their operation will be given. When the coal is fed from the hopper or bin into the retort, it moves upward toward the zone of combustion and is heated by conduction and radiation from the burning fuel in the combustion zone. As the temperature of the coal rises, it gives off moisture and occluded gases, which are largely non-combustible. When the temperature increases to around 700 or 800 F the coal particles become plastic, the degree of plasticity varying with the type of coal.

A rapid evolution of the combustible volatile matter occurs during and directly after the plastic stage of the coal. The distillation of volatile matter continues above the plastic zone where the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal used. While some of the ash fuses into particles in the surface of the coke as it is released, most of it remains on the hearth or grates and as this ash layer becomes thicker with time, that portion exposed to the higher temperatures surrounding the retort normally fuses into a clinker. The temperature attained in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating are factors which govern the degree of fusion.

Most bituminous coal stokers of Classes 1, 2 and 3 operate on the principle of the removal of the ash in this clinker form. Clinker tongs are provided to facilitate removal of the clinker on the smaller models.

There are a number of factors which materially affect the burning rate and the combustion results obtained with automatic stokers, the most important of these being the type and design of stoker, the characteristics of the fuel, and the manner in which the stoker is installed and operated.

Furnace Design

Due to the wide differences in stoker, boiler and furnace design, and fuel burning characteristics, it has not been found practical to establish fixed rules for the proportioning of furnaces for automatic coal stokers.

It is, therefore, essential that an experienced stoker installer give careful consideration to these factors when applying the equipment.

The *Stoker Manufacturers Association* has published standard recommendations on setting heights for stokers having capacities up to 1200 lb of coal per hour. Due to the variable conditions encountered, these data could not be prepared to apply to all equipment or fuels, but are intended merely as a guide for *average* minimum setting heights.

Combustion Adjustments

The coal feeding rate and air supply to the stoker should be regulated so as to maintain as nearly as possible an ideal balance between the load demand and the heat liberated by the fuel. Under such conditions no manual attention to the fuel bed should be required, other than the removal of clinker in stokers which operate on this principle of ash removal.

As in all combustion processes, the problem of maintaining the correct proportions of air and fuel is of utmost importance. It is desirable to supply a minimum amount of air required to properly burn the fuel at the rate it is being fed to the furnace.

While there may be only slight variations in the specified rate at which the coal is being fed to the furnace, due to variations in the size or density of the coal being used, there may be wide variations in the rate of air supplied as the result of changes in fuel bed resistance. These changes in resistance may be caused by changes in the porosity of the fuel bed due to variations in size or friability of the coal, ash and clinker accumulation, and variations in depth of the fuel bed. Because of this variable fuel bed resistance, many of the bituminous stokers, even in the smaller domestic sizes, incorporate air controls which automatically compensate for these changes in resistance and maintain a constant air fuel ratio.

It is also desirable on most stoker installations to provide automatic draft regulation in order to reduce air infiltration and provide better control during the banking or off periods of the stoker. The efficiency of combustion may be determined by analyzing with an Orsat apparatus the gases formed by the combustion process. With this equipment the percentage by volume of carbon dioxide (CO_2), oxygen (O_2), and carbon monoxide (CO) in the flue gases may be obtained. The percentage of CO_2 indicates the amount of excess air supplied. The presence of CO indicates a loss due to incomplete combustion, as the result of a deficiency of air or the improper mixing of air in the gases of combustion.

Sizing Stokers and Stoker Ratings

The *Stoker Manufacturers Association* has adopted a uniform method of rating stokers which includes convenient tables and charts for selecting the size of stoker required[2].

Controls

The heat delivery from the stoker of the smallest household type to the largest industrial unit can be accurately regulated with fully automatic

[2]Uniform Stoker Rating Code. Copies of this standard may be obtained from the *Stoker Manufacturers Association*, 307 North Michigan Ave., Chicago, Ill.

controls. The smaller heating applications are normally controlled by a thermostat placed in the building to be heated. Limit controls are supplied to prevent excessive temperature or pressure being developed in the furnace or boiler and refueling controls are used to maintain ignition during periods of low heat demand. Automatic low water cut-outs are recommended for use with all automatically fired steam boilers. See Chapter 33.

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically and safely from liquid fuels. This heat is produced in the furnace or fire-pot of hot water or steam boilers or warm air furnaces and is absorbed by the boiler, and thus made available for distribution to the house through the heating system.

The number of combinations of the characteristic elements of domestic oil burners is rather large and accounts for the variety of burners found in actual practice. Domestic oil burners may be classified as follows:

1. AIR SUPPLY FOR COMBUSTION
   - *a. Atmospheric*—by natural chimney draft.
   - *b. Mechanical*—electric-motor-driven fan or blower.
   - *c. Combination of* (*a*) *and* (*b*)—primary air supply by fan or blower and secondary air supply by natural chimney draft.
2. METHOD OF OIL PREPARATION
   - *a. Vaporizing*—oil distills on hot surface or in hot cracking chamber.
   - *b. Atomizing*—oil broken up into minute globules.
     - (1) Centrifugal—by means of rotating cup or disc.
     - (2) Pressure—by means of forcing oil under pressure through a small nozzle or orifice.
     - (3) Air or steam—by high velocity air or steam jet in a special type of nozzle.
     - (4) Combination air and pressure—by air entrained with oil under pressure and forced through a nozzle.
   - *c. Combination of* (*a*) *and* (*b*).
3. TYPE OF FLAME
   - *a. Luminous*—a relatively bright flame. An orange-colored flame is usually best if no smoke is present.
   - *b. Non-luminous*—Bunsen-type flame (*i.e.*, blue flame).
4. METHODS OF IGNITION
   - *a. Electric.*
     - (1) Spark—by transformer producing high-voltage sparks. Usually shielded to avoid radio interference. May take place continuously while the burner is operating (continuous ignition) or just at the beginning of operation (intermittent ignition).
     - (2) Resistance—by means of hot wires or plates.
   - *b. Gas.*
     - (1) Continuous—pilot light of constant size.
     - (2) Expanding—size of pilot light expanded temporarily at the beginning of burner operation.
   - *c. Combination*—electric sparks light the gas and the gas flame ignites the oil.
   - *d. Manual*—by manually-operated gas torch for continuously operating burners.

5. MANNER OF OPERATION

a. *On and off*—burner operates at fixed firing rate for period determined by load demand.

b. *High and low*—burner operates continuously but varies from a high to a low flame.

c. *Graduated*—burner operates continuously but flame is graduated according to needs by regulating both air and oil supply.

A trade classification of domestic oil burners consists of the following general types: (*a*) gun or pressure atomizing, (*b*) rotary and (*c*) pot or vaporizing. These are further classified as mechanical draft and natural draft based on method used to supply the air for combustion.

The gun type, illustrated in Fig. 12, is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes are employed at the end of the air tube to

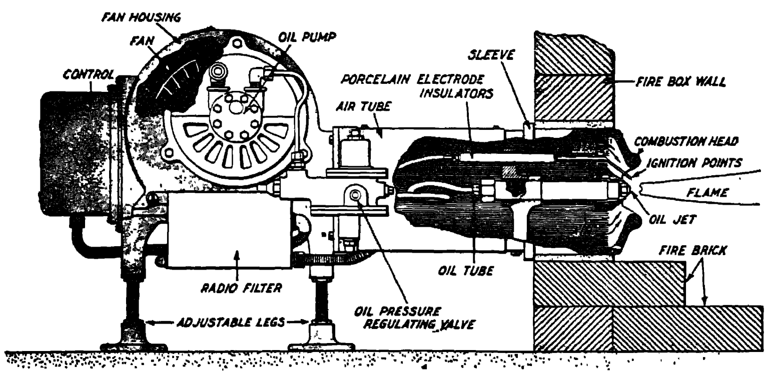


FIG. 12. GUN TYPE PRESSURE ATOMIZING OIL BURNER

influence the direction and speed of the air and thus the effectiveness of the mixing process.

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types: the center flame and wall flame. In the former type (Fig. 13), the oil is atomized by being thrown from the rim of a revolving disc or cup and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 14) differs in that combustion takes place in a ring of refractory material, which is placed around the hearth. These types of burners are further characterized by their installation within the ash pit of the boiler or furnace.

The pot type burner (Fig. 15) can be identified by the presence of a metal structure, called a pot or retort, in which combustion takes place.

When gun type (pressure atomizing) or horizontal rotary burners are used the combustion chamber is usually constructed of firebrick or other suitable refractory material, such as stainless steel, and is part of the installation procedure.

Most oil burners are operated by a small electric motor which pumps the oil and some or all of the air required. The smallest sizes can generally burn not much less than from 0.50 to 0.75 gal of oil per hour. The grade of oil burned ranges from No. 1 to No. 3. No. 3 oil is the heaviest and most viscous of the various grades mentioned. An oil burner satisfactory for No. 3 oil can burn any of the lighter grades easily but an oil burner recommended for No. 2 oil should never be supplied with the heavier grades. While the heavier grades of oil have a smaller heat value per pound, they have, due to greater density, a larger heat value per gallon.

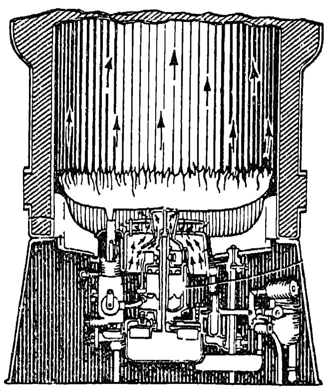


FIG. 13. CENTER FLAME VERTICAL ROTARY BURNER

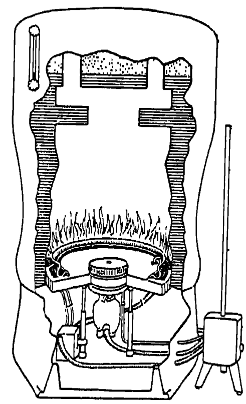


FIG. 14. WALL FLAME VERTICAL ROTARY BURNER

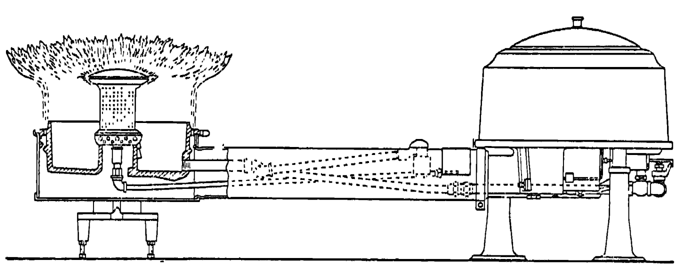


FIG. 15. POT TYPE VAPORIZING BURNER

The relative economy of the various grades must be based upon price and the amount of excess air required for clean and efficient combustion.

Operating Requirements for Mechanical Draft Oil Burners

The *U. S. Department of Commerce* in conjunction with the oil burner industry has established commercial standards for automatic mechanical draft oil burners for domestic installations which cover installation requirements and performance tests[3].

[3]Automatic Mechanical Draft Oil Burners Designed for Domestic Installations (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS75-39).

Oil-Fired Boiler and Furnace Units

Boilers and furnaces especially designed for oil burners are available. This type of equipment usually has more heating surface than the older coal-burning designs. Flue proportions and gas travel have been changed with beneficial results.

COMMERCIAL OIL BURNERS

Liquid fuels are used for heating apartment buildings, hotels, public and office buildings, schools, churches, hospitals, department stores, as well as industrial plants of all kinds. Contrary to domestic heating, convenience seldom is a dominating factor, the actual net cost of heat production usually controlling the selection of fuel. Some of the largest office buildings have been using oil for many years. Many department stores have found that floor space in basements and sub-basements can be used

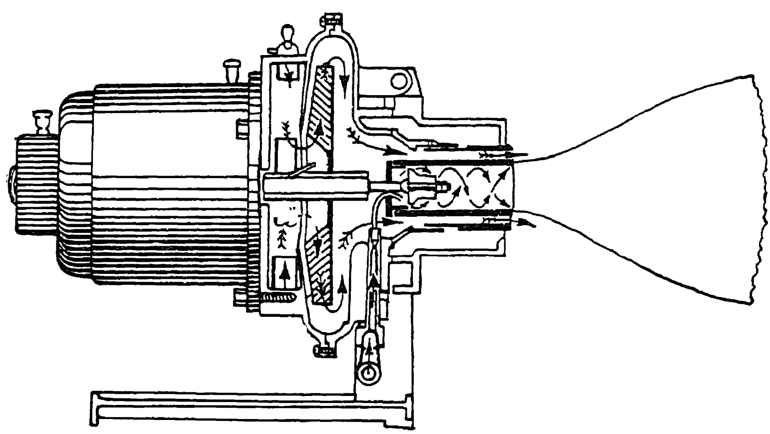


FIG. 16. HORIZONTAL ROTATING-CUP OIL BURNER

to better advantage for merchandising wares, and credit the heat producing department with this saving.

Wherever possible, the boiler plant should be so arranged that either oil or solid fuel can be used at will, permitting the management to take advantage of changes in fuel costs if any occur. Each case should be considered solely in the light of local conditions and prices.

Burners for commercial heating may be either large models of types used in domestic heating, or special types developed to meet the conditions imposed by the boilers involved. Generally speaking, such burners are of the mechanical or pressure atomizing types, the former using rotating cups which throw the oil from the edge of the cup at high velocity into the surrounding stream of air delivered by the blower (Fig. 16). As much as 350 gal of oil per hour can be burned in these units, and frequently they are arranged in multiple on the boiler face, from two to five burners to each boiler.

The larger installations are nearly always started with a hand torch, and are manually controlled, but the use of automatic control is increasing,

and completely automatic burners are now available to burn the two heaviest grades of oil. Nearly all of the smaller installations, in schools, churches, apartment houses and the like, are fully automatic.

Because of the viscosity of the heavier oils, it is customary to heat them before transferring by truck tank. It also has been common practice to preheat the oil between the storage tank and the burner, as an aid to movement of the oil as well as to atomization. This heating is accomplished by heat-transfer coils, using water or steam from the heating boiler.

Unlike the domestic burner, units for large commercial applications frequently consist of atomizing nozzles or cups mounted on the boiler front with the necessary air regulators, the pumps for handling the oil and the blowers for air supply being mounted in sets adjacent to the boilers. In such cases, one pump set can serve several burner units, and common prudence dictates the installation of spare or reserve pump sets. Pre-heaters and other essential auxiliary equipment also should be installed in duplicate.

Boiler Settings

As the volume of space available for combustion is the determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ash pit volume; in new installations the boiler should be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2.5 lb of oil per hour can properly be burned. This corresponds to an average liberation of about 38,000 Btu per cubic foot per hour. There are indications that at times much higher fuel rates may be satisfactory. For best results, care should be taken to keep the gas velocity below 40 ft per second. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or fire brick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

Combustion Process

Efficient combustion must produce a clean flame and must use relatively small excess of air, *i.e.*, between 25 and 50 per cent. This can be done only by vaporizing the oil quickly and completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil, for combustion, by transforming the liquid fuel to the gaseous state through the application of heat. This is accomplished before the oil vapor mixes with air to any extent and if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced. There may be a deficiency of air or an excessive supply

of air, depending upon burner adjustment, without altering the clean, blue appearance of the flame.

An atomizing burner *i.e.*, gun and rotary types is so named because the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization proceeds quickly. The result of such practice is the ability to burn more and heavier oil within a given combustion space or furnace volume. Since the air enters the fire pot with the liquid fuel particles, it follows that mixing, vaporization and burning are all occurring at once in the same space. This produces a luminous instead of a blue or non-luminous flame. In this case a deficient amount of air is indicated by a dull red or dark orange flame with smoky flame tips.

An excessive supply of air may produce a brilliant white flame in some cases or, in others, a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be easily detected, it is generally not possible to distinguish, by the eye alone, the finer adjustment which competent installation requires.

Tests indicate that there is no difference in economy between a blue flame and a luminous flame if the position, shape and the per cent of excess air of both flames are proper for each type.

Furnace or Combustion Chamber Design

With burners requiring a refractory combustion chamber the size and shape should be in accordance with the manufacturer's instructions. It is important that the chamber shall be as nearly air tight as is possible, except when the particular burner requires a secondary supply of air for combustion.

It is evident that the atomizing burner is dependent upon the surrounding heated refractory or fire brick surfaces to vaporize the oil and support combustion. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the fire brick surface, a carbon deposit will result. The combustion chamber should enclose a space having a shape similar to the flame but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray and the back wall rounded. A plan view of the combustion chamber thus resembles in shape the outline of the flame. In this way as much fire brick as possible is close to the flame so it may be kept quite hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead or inactive spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. With some of the vertical rotary burners considerable care must be exercised in definitely following

the manufacturer's instructions when installing the hearth as in this class successful performance depends upon this factor.

Combustion Adjustments

Where adjustments of oil and air have been made which give efficient combustion, the problem of maintaining the adjustments constant becomes an important one. Particularly is this true when the change causes the per cent of excess air to decrease below allowable limits of the burner. A decrease in air supply while the oil delivery remains constant or an increase in oil delivery while the air supply remains constant will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment (*i.e.*, 25 per cent excess air) the more critical it will be of variations. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (*a*) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (*b*) erosion of atomizing nozzle, (*c*) fluctuations in by-pass relief pressures and (*d*) possible variations in methods 2*b* (3) and 2*b* (4) listed in the previous classification table. Note that any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy and possibly a complete interruption of service but usually no soot will form.

The following factors may influence the air supply: (*a*) changes in combustion draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney and changes in draft resistance of boiler due to partial stoppage of the flues), and (*b*) changes in air inlet adjustments to the fan.

It is recognized that a secondary source of air due to leakage in the boiler setting is present in many installations and it is highly desirable that this leakage be reduced to a minimum. Obviously the amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases of combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02-0.05 in. water. An automatic draft regulator is very helpful in maintaining such values.

Even though a fan is generally used to supply the air for combustion, in most oil burners the importance of a proper chimney should not be overlooked. The chimney should have sufficient height and size to insure that the draft will be uniform within the limits given above if maximum efficiency throughout the whole heating season is to be maintained.

Measurement of the Efficiency of Combustion

Efficient combustion being based upon a clean flame and certain proportions of oil and air employed, it is possible to determine the results by analyzing the gases formed by the combustion process. Except in the case of a non-luminous flame it is usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates the best adjustment if the flame is clean. Most of the good installations at the

present time show from 8 to 10 per cent CO_2. Taking into account the potential hazard of oil or air fluctuations with low excess air (high CO_2) a setting to give 10 per cent CO_2 constitutes a reasonable standard for most oil burners.

Controls

Controls for oil burner operation, including devices for the safety and protection of a boiler or furnace, are fully described in Chapter 33.

GAS-FIRED APPLIANCES

The increased use of gas for house heating purposes has resulted in the production of such a large number of different types of gas heating systems and appliances that today there is probably a greater variety of them than there is for any other kind of fuel.

Gas-fired heating systems may be classified as follows:

- I. Gas-Designed Heating Systems.
  - A. Central Heating Plants.
    1. Steam, hot water, and vapor boilers.
    2. Warm air furnaces.
  - B. Unit Heating Systems.
    1. Warm air floor furnaces.
    2. Industrial unit heaters.
    3. Space heaters.
    4. Garage heaters.
- II. Conversion Heating Systems.
  - A. Central Heating Plants.
    1. Steam, hot water and vapor boilers.
    2. Warm air basement furnaces.

These systems are supplied with either automatic or manual control. Central heating plants, for example, whether gas designed or conversion systems, may be equipped with room temperature control, push-button control, or manual control.

Gas-Fired Boilers and Furnaces

Specially designed boilers are available for gas-firing such as shown in Fig. 17. Additional information on gas-fired boilers will be found in Chapter 11. Either snap action or throttling control is available for gas boiler operation. Throttling control is especially advantageous in steam systems because steam pressures can be maintained at desired points, while at the same time complete cut-off of gas is possible when the thermostat calls for it.

Warm air furnaces are variously constructed of cast-iron, sheet metal and combinations of the two materials. If sheet metal is used, it must be of such a character that it will have the maximum resistance to the corrosive effect of the products of combustion. With some varieties of manufactured gases, this effect is quite pronounced. Warm air furnaces are obtainable in sizes from those sufficient to heat the largest residence

down to sizes applicable to a single room. The practice of installing a number of separate furnaces to heat individual rooms is peculiar to mild climates. Small furnaces, frequently controlled by electrical valves actuated by push-buttons in the room above, are often installed to heat rooms where heat may be desired for an hour or so each day. These furnaces are used also for heating groups of rooms in larger residences. In a system of this type each furnace should supply a group of rooms in which the heating requirements for each room in the group are similar.

The same fundamental principle of design that is followed in the construction of boilers, that is, breaking the hot gas into fine streams so that all particles are brought as close as possible to the heating surface, is equally applicable to the design of warm air furnaces.

Codes for proportioning warm air heating plants, such as that formulated by the *National Warm Air Heating and Air Conditioning Association*

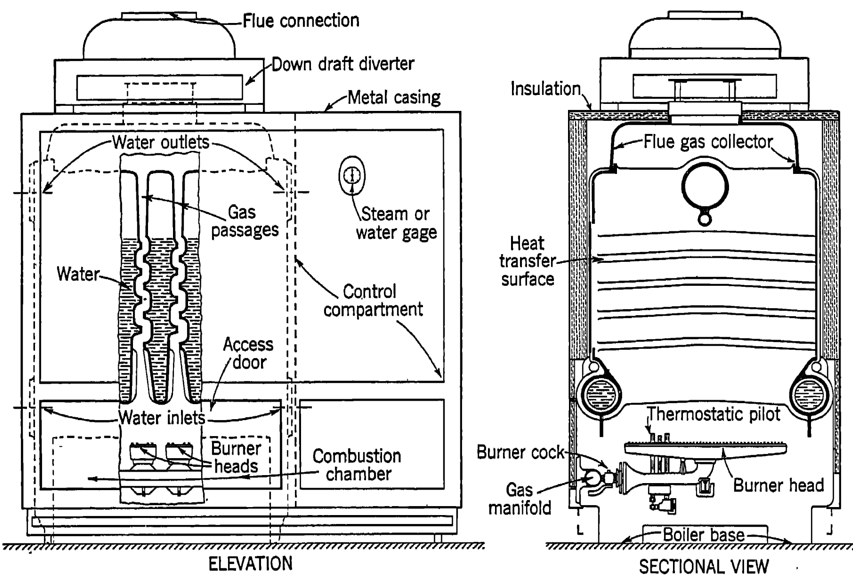


FIG. 17. COMBINATION GAS-FIRED BOILER

are equally applicable to gas furnaces and coal furnaces. Recirculation should always be practiced with gas-fired warm air furnaces. It not only aids in heating, but is essential to economy. Where fans are used in connection with warm air furnaces for residence heating, it is well to have the control of the fan and of the gas so coordinated that there will be sufficient delay between the turning on of the gas and the starting of the fan to prevent blasts of cold air being blown into the heated rooms. An additional thermostat in the air duct easily may be arranged to accomplish this.

Warm air floor furnaces are well adapted for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility as any number of rooms may be heated without heating the others. With the usual type the

register is installed in the floor, the heating element and gas piping being suspended below.

Space Heaters

Space heaters are generally used for auxiliary heating, but may be, and are in many cases, installed for furnishing heat to entire buildings. With the exception of wall heaters, they are portable, and can be easily removed and stored during the summer season. Although it is generally desirable to connect space heaters with solid piping, use of flexible gas tubing, semi-rigid tubing, or flexible metal hose is frequently resorted to, particularly in the connection of portable types. Where flexible gas tubing is used, a gas shut-off on the heater is not permitted and only *American Gas Association* certified tubing should be used.

Parlor furnaces or *circulators* are usually of the cabinet type. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base and passes up inside the jacket around a drum or heating section, and out of the heater at or near the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burner or burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass around baffles within the heating element or drum, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters give off considerable portion of their heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. Some have sheet-iron bodies finished in enamel or brass while others have cast-iron or brass frames with heavy fire-clay bodies. An atmospheric burner is supported near the center of the base, usually by set screws at each end. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are supported on legs and are portable; however, there are also types which are encased in a jacket which fits into the wall with a grilled front, similar to a wall register.

Gas-fired steam and hot water radiators are popular types of room heating appliances. They provide a form of heating apparatus for intermittently heated spaces such as stores, small churches and some types of offices and apartments. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to steam or hot water radiators. They are usually constructed of pressed steel or sheet metal hollow sections. The hot products of combustion circulate through the sections and are discharged from a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Garage heaters are usually similar in construction to the cabinet circulator space heaters, except that safety screens are provided over all

openings into the combustion chamber to prevent any possibility of explosion from gasoline fumes or other gases which might be ignited by an open flame. They are usually provided with automatic room temperature controls and are well suited for heating either residence or commercial garages.

Conversion Burners

Residence heating with gas through the use of conversion burners installed in coal-designed boilers and furnaces represents a common type of gas-fired house heating system. In many conversion burners radiants or refractories are employed to convert some of the energy in the gas to radiant heat. Others are of the blast type, operating without refractories.

Many conversion units are equipped with a sheet metal secondary air duct which is inserted through the ashpit door. The duct is equipped with automatic air controls which open when the burners are operating and close when the gas supply is turned off. This prevents a large part of the circulation of cold air through the combustion space of the appliance when not in operation. With this duct the air necessary for proper combustion is supplied directly to the burner, thereby making it possible to reduce the excess air passing through the combustion chamber.

Conversion units are made in many sizes both round and rectangular to fit different types and makes of boilers and furnaces. They may be secured with manual, push-button, or room temperature control.

Combustion Process and Adjustments

Because of the varying composition of gases used for domestic heating it is difficult to generalize on the subject of gas burner combustion.

Little difficulty should be experienced in maintaining efficient combustion conditions when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value and, therefore, the rate of heat supply is nominally constant. Since the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft diverter insures the maintenance of constant low draft condition in the combustion chamber with a resultant stability of air supply. A draft diverter is also helpful in controlling the amount of excess air and preventing back drafts which might extinguish the flame. (See Chapter 7).

Measurement of the Efficiency of Combustion

It is possible to determine the results of combustion by analyzing the gases of combustion with an Orsat apparatus. It is desirable to determine the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. While ultimate CO_2 values of 10 to 12 per cent may be obtained from the combustion of gases commonly used for domestic heating, a combustion adjustment which will show from 8 to 10 per cent CO_2 represents a practical value. Under normal conditions no CO will be produced by a gas-fired boiler or furnace. Limitations as to output rating by the *A.G.A.* are based upon operation with not more than 0.04 per cent CO in the products of combustion. This is too small an amount to be determined by the ordinary flue gas analyzer.

Controls

Temperature controls for gas burners are described in Chapter 33. Some central heating plants are equipped with push button or other manual control. The main gas valve may be of either the snap action or throttling type.

Sizing Gas-Fired Heating Plants

While gas-burning equipment usually is completely atuomatic, maintaining the temperature of rooms at a predetermined and set figure, there are in use installations which are manually controlled. Experience has shown that, in order to effectively overcome the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard cast-iron column radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up or starting loads. Consequently, it is possible to use a much lower selection, or safety factor. A gas-fired boiler under thermostatic control is sensitive to variations in room temperatures so that in most cases a factor of 20 per cent is sufficient for pick-up load.

The factor to be allowed for loss of heat from piping, however, must vary somewhat, the proportionate amount of piping installed being considerably greater for small installations than for large ones. Liberal selection factors to be added to the installed steam radiation under thermostatic control are given in Fig. 1 of Chapter 11.

Appliances used for heating with gas should bear the approval seal of the *American Gas Association* Testing Laboratory. Installations should be made in accordance with the recommendations shown in the publications of that association.

Ratings for Gas Appliances

Since a gas appliance has a heat-generating capacity that can be predicted accurately to within 1 or 2 per cent, and since this capacity is not affected by such things as condition of fuel bed and soot accumulation, makers of these appliances have an opportunity to rate their product in exact terms. Consequently all makers give their product an hourly Btu output rating. This is the amount of heat that is available at the outlet of a boiler in the form of steam or hot water, or at the bonnet of the furnace in the form of warm air. The output rating is in turn based upon the Btu input rating which has been approved by the *American Gas Association* Testing Laboratory and upon an average efficiency which has been assigned by that association.

In the case of boilers, the rating can be put in terms of square feet of equivalent direct radiation by dividing it by 240 for steam, and 150 for water. This gives what is called the *American Gas Association* rating, and is the manner in which all appliances approved by the *American Gas Association* Laboratory are rated. To use these ratings it is only necessary to increase the calculated heat loss or the equivalent direct radiation load by an appropriate amount for starting and piping, and to select the boiler or furnace with the proper rating.

The rating given by the *American Gas Association* Laboratory is not only a conservative rating when considered from the standpoint of capacity and efficiency, but is also a safe rating when considered from the standpoint of physical safety to the owner or caretaker. The rating that is placed upon an appliance is limited by the amount of gas that can be burned without the production of harmful amounts of carbon monoxide. Gas boilers are available with ratings up to 14,000 sq ft of steam radiation, while furnaces with ratings up to about 500,000 Btu per hour are available.

REFERENCES

Domestic Burners for Pennsylvania Anthracite, Commercial Standard CS48-40, U. S. Department of Commerce.

Uniform Stoker Rating Code, *Stoker Manufacturers Association*, adopted November 15, 1937.

Recommended Standards Governing Minimum Setting Heights, *Stoker Manufacturers Association*, adopted March 12, 1940.

Performance Expectancy of Domestic Underfeed Stokers for Anthracite, by Allen J. Johnson (*Transactions, A.I.M.E.*, Coal Division, Vol. 119, 1936).

The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers—The Relation of the Size in the Hopper to That Burned in the Retort, by R. A. Sherman and E. R. Kaiser, Technical Report No. 1, Bituminous Coal Research, Inc., (December, 1935) Pt. I.

The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers—Burning Tests on Four Typical Coals, by R. A. Sherman, E. R. Kaiser and H. R. Limbacher, Technical Report No. 1, Bituminous Coal Research, Inc. (July, 1937) Pt. II.

Stoker Coals, Anonymous, Bulletin of Chesapeake and Ohio Railway Co., 1935.

A.S.H.V.E. Research Report No. 907—Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. Transactions, Vol. 37, 1931, p. 517).

A.S.H.V.E. Research Report No. 925—A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 317).

Air Supply and Its Effect on Performance of Oil Burners and Heating Boilers, by L. E. Seeley, J. H. Powers and E. J. Tavanlar (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 75).

Study of Fuel Burning Rates and Power Requirements of Oil Burners in Relation to Excess Air, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 319).

Oil Burning in Residences, by D. W. Nelson (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 355).

Domestic Oil Burners, by A. H. Senner (*Mechanical Engineering*, November, 1936).

A Study of Oil-Fired Heating Boilers, by R. C. Cross and W. R. Lyman (*Heating and Ventilating*, October, 1931).

Value of Oil Burner Installation Surveys, by R. C. Cross and W. R. Lyman (*Heating and Ventilating*, January, 1931).

Comfort Heating, *American Gas Association*.

Approval Requirements of Central House Heating Gas Appliances, *American Gas Association*.

A Method for Determining Fuel Burning Rates in Heating Boilers Fired by Automatic Devices, by R. C. Cross (*Heating and Ventilating*, January, 1932).

Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 185).

Efficiencies and Costs of Various Fuels in Domestic Heating, by R. A. Sherman and R. C. Cross, Technical Report No. 3, Bituminous Coal Research, Inc., (December, 1936).

Automatic Heating Equipment, *American Architect Reference Data*, No. 7, September, 1933.

Chapter 10

HEAT AND FUEL UTILIZATION

Fuel Consumption Records, Calculated Heat Loss Estimation Method, Maximum Rate of Fuel Burning, Degree-Day Method, Unit Fuel Consumption per Degree-Day, Maximum Demands and Load Factors

MANY methods are in use for estimating in advance of actual operation the anticipated heat or fuel consumption of heating plants over long or short periods. With suitable modification in procedure these same general methods are frequently useful in checking the degree of effectiveness with which heat or fuel is utilized during plant operation.

In applying any of these estimating methods to the consumption of a particular building plant it should be noted that (a) reliable records of past heat or fuel consumptions of *this* building will usually produce more trustworthy estimates of future consumptions than will any data obtained by averages or from other similar buildings; (b) where no past records exist useful data can sometimes be obtained from records of similar buildings with similar plants *in the same locality;* (c) records of consumption, which are averages from many types of plants in many types of buildings in various localities, can produce no better than an average estimate which may be far from accurate; (d) estimates based on computed heat losses without the benefit of operating data are wholly dependent on how well the computation represents the actual facts.

Where records of past consumptions are available they should be examined for reliability to be sure that the records show fuel or heat for the heating plant only, or else make a suitable allowance for fuel used for other purposes, such as heating service water. Weights and measures shown on invoices may not always agree with fuel used, for residues left in bins or tanks may represent a considerable fraction of the fuel charged to a building. Generally, plant operating records of fuel used are to be preferred to those obtained from accounting or bookkeeping offices from fuel invoices.

Records from similar buildings even in the same locality should be examined with care before being used as the basis of estimates. The type of heating system, the quality of supervision in manual plants, the kind of control in automatic plants, and the attention given to the plant operation are all factors in fixing the consumption in any building. Many times these factors do not show up in superficial examination and are even difficult to evaluate when known to be present. Especially check the

records to be sure that they do not include energy or fuel used for other purposes than heating the building.

Estimates based on computed heat losses alone are frequently the only ones possible to obtain, especially where new equipment is put into unusual buildings and there is a scarcity of records and an absence of experience data. Such estimates also have to be made where direct information is not obtainable as, for example, if a survey is being made without the assistance or knowledge of the building operator and thus without information as to the actual consumption. Estimates of this kind are also useful in some cases where a *relative* standard of performance is desired to serve as a base of comparisons in a campaign of fuel utilization. In such situations it can be plausibly argued that an estimate based on computed heat quantities is to be preferred to one which is related to operating methods.

In interpreting and evaluating heat or fuel consumption estimates as well as in their preparation, it is well to realize that any estimating method used will produce a more reliable result over a long period operation than over a short period. Nearly all of the methods in common use will give trustworthy results over a full *annual* heating season, and in some cases such estimates will prove consistent within themselves for monthly periods. As the period of the estimate is shortened there is more chance that some factor not allowed for in the estimating method will become controlling and thus give discrepant and even ridiculous results.

Of the various estimating methods in use attention is directed in this discussion to but two as they are illustrative of all, viz: (1) calculated heat loss method, and (2) degree-day method.

CALCULATED HEAT LOSS METHOD

This method is theoretical and assumes constant temperatures for very definite hours each day throughout the entire heating season. It does not take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, sun effect, poor heating systems, and many others.

In order to apply this method the hourly heat loss from the building under maximum load, or design condition is computed following the principles discussed in Chapters 3 and 4 and the method described and illustrated in Chapter 5.

In some cases, however, depending on the presence of interior partitions, the computed heat loss is modified when used for estimating the heat or fuel consumption. If the building has no interior walls or partitions then, by the method of Chapters 4 and 5, the infiltration losses are calculated by using only half the total window crack. In such a building the calculated loss need not be modified in order to prepare heat or fuel estimates by this method. Where the building does contain interior walls or partitions instead of using as the calculated heat loss (H) which is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i), it is more desirable to let $H = H_t + \frac{H_i}{2}$.

In predicting fuel consumption for heating a building by the Calculated Heat Loss Method, the general formula is:

$$F = \frac{H\,(t - t_a)\,N}{E\,(t_d - t_o)\,C} \qquad (1)$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).

H = calculated heat loss, Btu per hour, during the design hour, based on t_o and t_d $\left(\text{generally } H = H_t + H_i \text{ but may on occasion equal } H_t + \frac{H_i}{2}\right)$.

t = average inside temperature maintained over estimate period, degrees Fahrenheit.

t_a = average outside temperature through estimate period, degrees Fahrenheit (Table 2, Chapter 5).

t_d = inside design temperature, degrees Fahrenheit (usually 70 F).

t_o = outside design temperature, degrees Fahrenheit (see Outside Temperatures, Chapter 5).

N = number of heating hours in estimate period (for an Oct. 1—May 1 heating season, 5088).

E = efficiency of utilization of the fuel over the period, expressed as a decimal; not the efficiency at peak or rated load condition.

C = heating value of one unit of fuel or energy.

Example 1. A residence in Philadelphia is to be heated to 70 F from 6 A.M. to 10 P.M. and 55 F from 10 P.M. to 6 A.M. The calculated hourly heat loss is 120,000 Btu per hour based on 70 F inside at −5 F outside. If the building is to be heated by metered steam, how many pounds would be required during an average heating season?

Solution. The heating value of steam may be taken as 1000 Btu per pound, and since it is purchased steam, the efficiency can be assumed as 100 per cent. From Table 2, Chapter 5, t_a = 42.7 F. The average inside temperature is:

$$\frac{(16 \times 70) + (8 \times 55)}{24} = 65 \text{ F.}$$

Substituting in Equation 1:

$$F = \frac{120{,}000\,(65 - 42.7)\,5088}{1.00\,[70 - (-5)]\,1000} = 181{,}239 \text{ lb.}$$

Example 2. How much would the fuel cost to heat the building in Example 1 during an average heating season with coal at $8 per ton and with a calorific value of 11,000 Btu per pound, assuming that the seasonal efficiency of the plant was 55 per cent?

Solution. Substituting in Equation 1: $F = \frac{120{,}000\,(65 - 42.7)\,5088}{0.55\,[70 - (-5)]\,11{,}000} = 30{,}013$ lb = 15 tons, which, at $8 per ton, costs $120.

Example 3. What will be the estimated fuel cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The seasonal efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 P.M. to 7 A.M. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

Solution. The average hourly temperature is:

$$t_a = \frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The maximum hourly heat loss will be:

$$H = 92{,}000 - \frac{26{,}000}{2} = 79{,}000 \text{ Btu.}$$

$$M = \frac{79{,}000\ (66.3 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

$2204.6 \times \$0.07 = \$154.34$ = estimated fuel cost per year of heating building.

Equation 1 can be expressed as:

$$F = \frac{H}{EC} \times \frac{(t - t_a)\ N}{(t_d - t_o)} \tag{2}$$

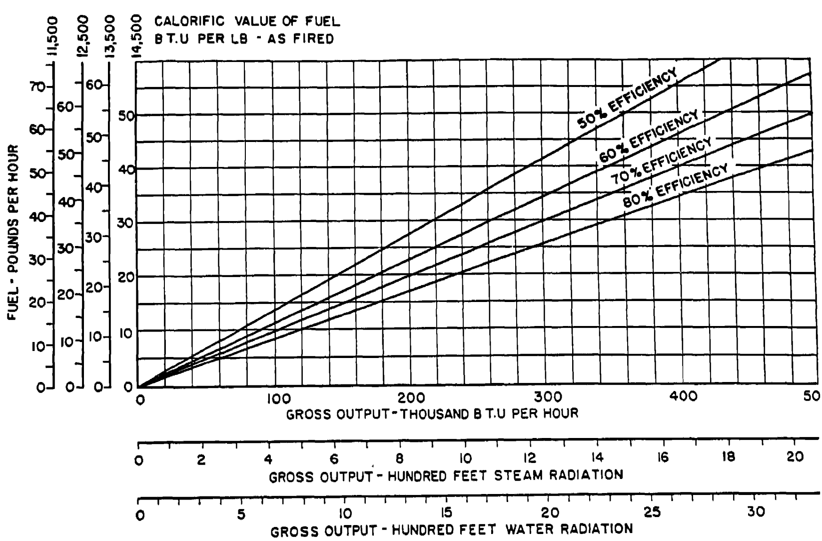


FIG. 1. COAL FUEL BURNING RATE CHART

where the expression $\frac{H}{EC}$ is the rate at which fuel is burned during the design hour. Values of this rate are plotted as ordinates in Figs. 1, 2 and 3 for coal, oil and gas. For a given efficiency, the rate of fuel burning is directly proportional to the load and therefore these charts can be extended by moving the decimal points the same number of digits in both vertical and horizontal scales. Use of these charts thus expedites the estimate.

The charts are plotted so that the load is expressed in three terms: (a) hourly heat loss at design conditions, (b) square feet of steam radiator surface (240 Btu per hour), and (c) square feet of hot water radiator surface (150 Btu per hour). By entering the chart at the correct point on the abcissa corresponding to the calculated heat loss (H), following vertically to the seasonal efficiency assumed and thence horizontally to the fuel rate, the rate of fuel burning during a maximum or design hour will be found along the left hand scale for various calorific values of the fuel.

In the case of gravity warm air heating installations, the load is usually expressed in square inches of leader pipe. This can be converted into hourly heat loss by multiplying by the factors in Table 1.

Example 4. A building located in Salt Lake City with an oil-burning heating plant has a calculated hourly heat loss of 260,000 Btu per hour. The plant is designed to maintain a temperature of 70 F inside during all 24 hours of the day, the outside design temperature is −5 F, the average outside temperature 40 F, the heating season 5088 hours long, the assumed efficiency 60 per cent, and the oil has a calorific value of 143,400 Btu per gallon. What will be the seasonal fuel consumption?

Solution. Enter Fig. 2 at 260,000 on the upper horizontal scale, move vertically to the 60 per cent efficiency curve, and horizontally to the vertical scale where the firing rate $\left(\frac{H}{EC}\right)$ is found as 3.0 gal per hour.

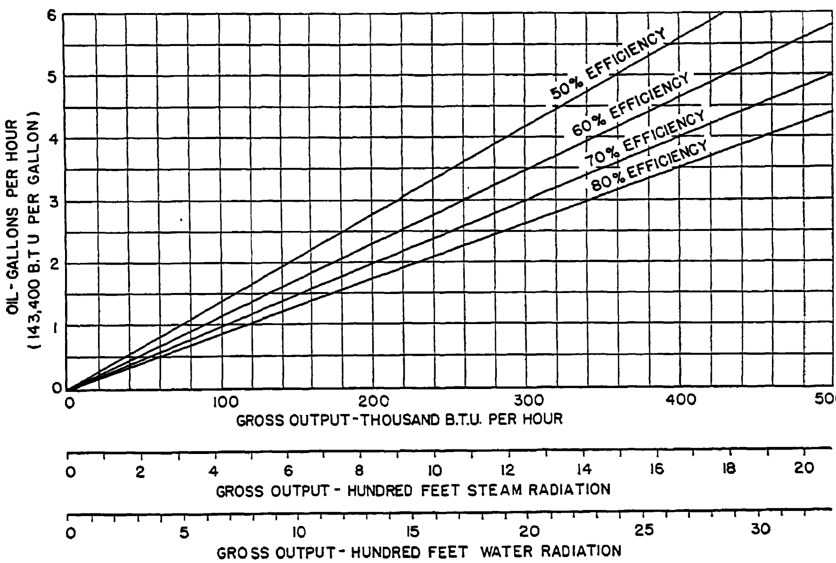


FIG. 2. OIL FUEL BURNING RATE CHART[a]

[a]This chart is based upon No. 3 oil having a heat content of 143,400 Btu per gallon. If other grades of oil are used multiply the value obtained from this chart by the following factors: No. 1 oil (139,000 Btu per gallon) 1.032; No. 2 oil (141,000 Btu per gallon) 1.017; No. 4 oil (144,500 Btu per gallon) 0.992; No. 5 oil (146,000 Btu per gallon) 0.982; and No. 6 oil (150,000 Btu per gallon) 0.956.

Substituting this in Equation 2 for $\left(\frac{H}{EC}\right)$:

$$F = 3.0 \times \frac{(70 - 40)\ 5088}{70 - (-5)} = 6106 \text{ gal.}$$

Example 5. What would be the total gas consumption over a full heating season of a gas-fired gravity warm air furnace designed according to the Code[1], and with four 12 in. and two 8 in. round leaders to the first floor and six 10 in. leaders to the second floor, if the gas has a heating value of 500 Btu per cubic foot, the plant operates at a 70 per cent

[1]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences (9th edition), and the Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems, may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 5 E. Long St., Columbus, Ohio.

seasonal efficiency and is designed to maintain an average inside temperature of 65 F when it is 10 F outside in a city where the average outside temperature is 45 F and the heating season is 5088 hours long?

Solution. The area of the round leaders is: 12 in., 113 sq in.; 10 in., 79 sq in.; and 8 in., 50 sq in. From Table 1 the total Btu transmitted is:

First Floor: [(4 × 113) + (2 × 50)] × 111 = 61,272 Btu per hour.
Second Floor: (6 × 79) × 167 = 79,158 Btu per hour.

Total 140,430 Btu per hour.

Allowing 10 per cent for duct and furnace losses, the gross output would be 154,500 Btu per hour.

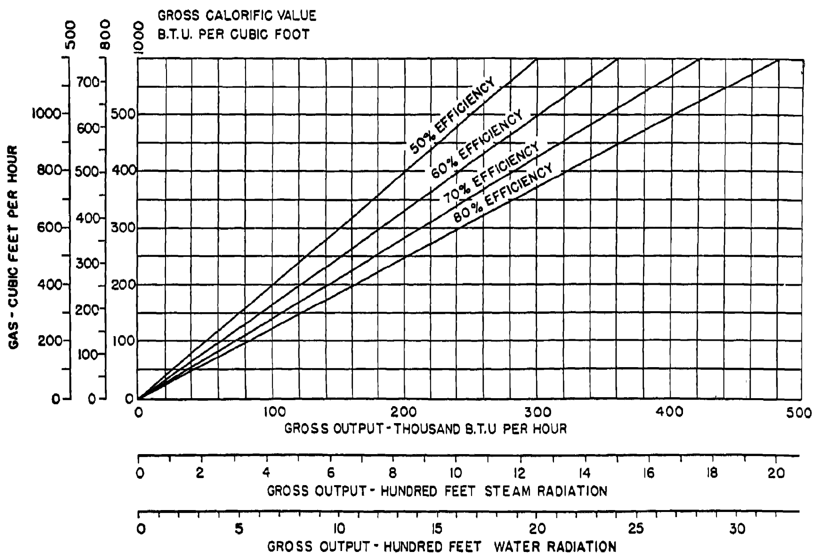


FIG. 3. GAS FUEL BURNING RATE CHART

Enter Fig. 3 at 154.5 on the upper horizontal scale, move to the 70 per cent efficiency curve and thence to the 500 Btu per cubic foot vertical scale and find $\left(\frac{H}{EC}\right)$ to be approximately 440 cu ft per hour.

Substituting in Equation 2:

$$F = 440 \times \frac{(65 - 45)\ 5088}{(70 - 10)} = 746{,}428 \text{ cu ft.}$$

Maximum Rate of Fuel Burning

The rate at which fuel is burned during the maximum, or design hour, is frequently useful in setting, or adjusting, the fuel feed devices attached to stokers, oil-burners, and gas burners. This rate is $\left(\frac{H}{EC}\right)$ and can be found from the charts of Figs. 1, 2 and 3 in the same way as outlined in the Examples 4 and 5. In using the charts for this purpose, however, it should be noted that the efficiency (E) is the overall efficiency of the boiler

or furnace at the time of peak load. This efficiency is generally considerably greater than the value selected for E when the seasonal efficiency of utilization is used in making seasonal fuel estimates. Failure to distinguish between the two essentially different meanings attached to E may result in grossly inaccurate estimates.

The correct fuel burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices usually introduce the fuel at a single fixed rate during the *on* periods and this rate should be sufficient to carry the gross or maximum design load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stoker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

Table 1. Heat Carrying Capacity of Gravity Warm Air Furnace Round Leader Pipes

180 F Register Temperature

Leader Pipe	Btu per Hr at Design Conditions per Sq In. of Leader Pipe
First floor	111
Second floor	167
Third floor	200

Example 6. The estimated net load (including domestic hot water supply) as calculated for a residence is 1500 sq ft of hot water radiation. Determine the firing rates for various mechanically fired fuels assuming an overall boiler efficiency of 70 per cent; using coal with a calorific value of 12,500 Btu per pound; No. 3 fuel oil and natural gas having a gross heating value of 1000 Btu per cubic foot.

Solution. Referring to Fig. 1, Chapter 11, a piping and pick-up factor for a net load of 1500 sq ft is found to be 43 per cent or the gross output is equivalent to $1500 \times 1.43 = 2145$ sq ft of hot water radiation.

Using the charts in Figs. 1, 2 and 3 project vertically from the gross output value on the proper horizontal scale to the intersection of the 70 per cent efficiency line. From the intersection of this line proceed horizontally to the proper vertical scale where a direct value of the required fuel burning rate is given. These values are rates of burning while firing device is in operation and are not indicative of hourly fuel consumption.

By use of the respective charts the firing rates for the various fuels will be found to be: coal 36.8 lb per hour, oil 3.2 gal per hour, and gas 460 cu ft per hour.

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the Calculated Heat Loss Method, it is of more value for practical use.

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. Some years ago the *American Gas Association*[2] determined from experi-

[2]See Industrial Gas Series, House Heating, (*third edition*) published by the *American Gas Association*.

ment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the outside temperature. In other words, on a day when the temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. The degree-day is defined in Chapter 46.

Some years ago the *National District Heating Association* studied the metered steam consumption of 163 buildings[3] in 22 different cities and published data substantiating the fact that the 65 F base originally chosen by the gas industry is approximately correct. (See Table 2.)

If the degree-days occurring each day are totaled for a reasonably long period, the fuel consumption during that period as compared with another period will be in direct proportion to the number of degree-days in the two periods. Consequently, for a given installation, the fuel consumption can be calculated in terms of fuel used per degree-day for any sufficiently long period and compared with similar ratios for other periods to determine the relative operating efficiencies with the outside temperature variable eliminated.

Predictions of fuel consumption are generally based on the average number of degree-days which have occurred over a long period of years, and such averages, by months, on a 65 F base, are given for various United States and Canadian cities in Table 3. In general, attempts to apply the degree-day method to fuel consumptions over a period of less than a month are of questionable value.

Formula for Degree-Day Method

The general formula for predicting fuel consumption by the Degree-Day Method is:

$$F = U \times N \times D \tag{3}$$

where

F = fuel consumption for the estimate period.

U = unit fuel consumption, or quantity of fuel used per degree-day per building load unit.

N = number of building load units.

D = number of degree-days for the estimate period.

Values of D for use in Equation 3 are given in Table 3. Values of N depend on the particular building for which the estimate is being prepared and must be found by surveying plans, by observation, or by measurement of the building. Values of U for use in this equation are the Unit Fuel Consumptions per Degree-Day and are obtained as a result of the collection of operating information. Certain of this information is presented later but before referring to these data attention is directed to the nature of the unit.

Unit Fuel Consumptions per Degree-Day

The quantity of fuel used per degree-day in a given heating plant can be reduced to a unit basis in terms of quantity of fuel (or steam) per degree-day per square foot of radiator, per cubic foot of heated building space, or per thousand Btu hourly heat loss at design conditions. A less frequently used basis is quantity of fuel per degree-day per square foot of

[3]These buildings are all served with steam from a district heating company.

floor area. In fact any convenient unit can be used to relate the consumption to the degree-day and to the building.

The choice of these units requires explanation and some discrimination and judgment. The use of heated space in preference to the gross building cubage used by architects is obviously more accurate for this purpose. The architects' cubage includes the outer walls and certain percentages of attic and basement space which are usually unheated. The net heated space is usually about 80 per cent of the gross cubage and can be calculated from the latter if it cannot be measured. The cubical content is somewhat inaccurate as a basis of comparison due to differences in types of construction, exposure, and ratio of exposed area to cubical contents. Use of equivalent radiator surface figures is fundamentally the same as

TABLE 2. BASE TEMPERATURE FOR THE DEGREE-DAY[a]

Type of Building	No. of Buildings Analyzed	Temperature F Corresponds to Zero Steam Consumption
Office	60	66.2
Office and Bank	4	65.8
Bank	3	66.2
Office and Telephone Exchange	2	65.5
Office and Stores	6	67.4
Stores	11	64.0
Department Stores	12	64.3
Hotels	7	66.5
Apartments	14	68.8
Residences	8	66.9
Clubs	4	65.5
Lodges	5	64.9
Theaters	3	67.6
Churches	2	65.8
Garages	2	64.8
Auto Sales and Service	4	61.2
Newspaper and Printing	3	67.7
Warehouse and Loft	3	67.7
Office and Loft	2	65.2
Manufacturing	8	65.4
Average for 163 Buildings		66.0 F

[a]Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932.

using a calculated heat loss and therefore units in terms of fuel per degree-day per equivalent square foot of calculated radiator surface, per 1000 Btu of calculated heat loss, or per Btu of heat loss at design conditions are all of equal accuracy and desirability. It is doubtful if installed radiator surface as determined by count should be used at all. Radiator units are also of questionable value where there is fan coil surface or warm air systems. In view of all these considerations it is believed that the unit based on *thousands of Btu of hourly calculated heat loss for the design hour* is probably the most desirable although the one most widely used seems to be *units of fuel (or heat) per degree-day per square foot of equivalent direct radiator surface.*

Since this unit is the one most widely used at present the unit fuel consumptions given in succeeding paragraphs of this chapter make use of this unit to a considerable extent, although it should be understood that most of these units of consumption can be transposed as desired.

TABLE 3. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a]

STATE	CITY	JAN.	FEB.	MAR.	APR.	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.	Total
Ala.	Birmingham	617	476	298	58						51	333	577	2410
	Mobile	418	288	164	11						3	192	397	1473
Ariz.	Phoenix	428	277	133	4							160	403	1405
Ark.	Fort Smith	790	622	384	97						89	420	710	3112
	Little Rock	732	563	372	92						72	387	645	2863
Calif.	Los Angeles	322	266	232	168	87	3				11	123	260	1472
	San Francisco	468	358	335	300	254	195	202	183	123	140	261	425	3244
Colo.	Denver	1091	904	797	537	273	18			76	428	756	1014	5894
	Grand Junction	1271	899	663	378	123				31	378	771	1162	5676
Conn.	New Haven	1141	1008	905	534	220	10			55	347	690	1008	5918
D. C.	Washington	980	832	694	351	61				3	236	594	880	4631
Fla.	Jacksonville	298	196	76								88	270	928
Ga.	Atlanta	694	552	403	120						80	387	629	2865
	Savannah	422	308	186	15						7	195	391	1524
Idaho	Boise	1091	846	691	438	245	30			102	431	720	1020	5614
Ill.	Chicago	1237	1053	890	519	202	3			17	307	714	1085	6027
	Springfield	1194	994	769	369	71				7	285	684	1032	5405
Ind.	Evansville	976	804	592	249	16					174	552	865	4228
	Indianapolis	1135	949	775	387	77				12	288	681	1017	5321
Iowa	Des Moines	1392	1156	902	447	117				28	360	798	1209	6409
	Sioux City	1463	1232	1001	516	142				68	437	894	1299	7052
Kan.	Dodge City	1116	890	688	342	65				3	276	672	1004	5056
	Topeka	1159	952	694	309	49				3	248	666	1023	5103
Ky.	Lexington	995	829	660	321	46				2	236	606	905	4600
	Louisville	949	778	608	258	16					178	549	849	4185
La.	New Orleans	335	216	71								104	291	1017
	Shreveport	558	395	208	16						24	270	493	1964
Me.	Eastport	1383	1218	1119	780	536	297	143	133	276	542	849	1200	8476
	Portland	1321	1154	1029	660	363	79		5	162	468	810	1159	7210
Md.	Baltimore	967	829	704	342	47				2	211	561	862	4525
Mass.	Boston	1150	1014	911	558	245	13			61	353	690	1008	6003
Mich.	Detroit	1259	1112	980	564	217	4			58	388	771	1107	6460
	Marquette	1510	1364	1246	816	496	183	12	37	225	567	951	1314	8721
Minn.	Duluth	1770	1501	1280	840	549	234	32	74	297	648	1050	1522	9797
	Minneapolis	1621	1375	1097	558	226	9			113	499	978	1407	7883
Miss.	Vicksburg	521	370	202	19						22	252	465	1851
Mo.	Kansas City	1141	946	691	306	43				2	226	639	1008	5002
	St. Louis	1051	846	648	267	15					191	588	933	4539
	Springfield	976	834	614	270	41				2	211	579	893	4420
Mont.	Havre	1615	1439	1175	639	360	94		22	258	636	1014	1383	8635
Neb.	Lincoln	1308	1089	852	405	107				21	335	777	1159	6053
	Omaha	1336	1106	868	414	91				15	332	795	1197	6154
Nev.	Winnemucca	1128	882	775	549	344	76			175	518	798	1085	6330
N. H.	Concord	1345	1182	1060	648	332	68		4	171	474	819	1184	7287
N. J.	Altantic City	1008	879	818	516	214	8			10	251	582	887	5173
N. M.	Santa Fe	1122	893	784	549	288	30			122	453	783	1063	6087
N. Y.	Albany	1299	1145	1001	546	177	1			69	400	771	1132	6541
	Buffalo	1252	1140	1051	666	322	41			81	406	768	1091	6818
	New York	1057	944	846	468	139				12	270	624	930	5290
N. C.	Raleigh	741	610	459	168	1					98	420	682	3179
	Wilmington	574	479	363	94						31	270	493	2304
N. D.	Bismark	1773	1532	1265	687	326	55		5	207	623	1095	1559	9127
Ohio	Cincinnati	1076	902	747	378	71				10	288	675	980	5127
	Cleveland	1194	1053	942	564	220	6			47	353	723	1048	6150
	Columbus	1128	960	803	414	93				15	304	693	1011	5421
Okla.	Oklahoma City	887	711	465	156	3					120	486	797	3625
Ore.	Baker	1243	1008	849	594	412	192	12	27	270	570	870	1169	7216
	Portland	794	641	561	396	251	80			100	335	546	738	4442
Pa.	Philadelphia	1004	871	750	387	77				3	223	579	890	4784

TABLE 3. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a] (CONCLUDED)

STATE	CITY	JAN.	FEB.	MAR.	APR.	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.	Total
Pa.	Pittsburgh	1063	916	787	414	91	...	...	...	15	288	654	955	5183
S. C.	Charleston	468	353	236	37	...	...	...	...	...	8	207	412	1721
	Columbia	589	470	304	63	...	...	...	...	...	54	330	552	2362
S. D.	Huron	1665	1420	1119	597	267	18	...	...	112	536	1005	1435	8174
	Rapid City	1333	1165	1004	621	341	49	...	...	140	512	873	1181	7219
Tenn.	Knoxville	812	647	505	210	8	...	...	...	...	160	513	766	3621
	Memphis	747	580	394	98	...	...	...	...	...	76	399	663	2957
	Nashville	818	655	490	180	3	...	...	...	...	130	480	744	3500
Texas	El Paso	620	448	285	59	...	...	...	...	...	72	369	623	2476
	Fort Worth	608	468	226	25	...	...	...	...	...	24	285	542	2178
	Houston	381	255	56	...	...	...	...	...	...	...	122	329	1143
	San Antonio	394	269	70	...	...	...	...	...	...	...	138	350	1221
Utah	Modena	1187	952	831	570	356	62	...	...	150	527	858	1114	6637
	Salt Lake City	1110	874	722	462	236	12	...	...	54	388	717	1026	5601
Vt.	Burlington	1432	1277	1113	651	264	21	...	...	140	490	861	1259	7508
Va.	Lynchburg	852	692	549	231	9	...	...	...	1	202	534	790	3860
	Norfolk	756	624	521	246	19	...	...	...	...	89	408	679	3342
	Richmond	840	711	552	252	15	...	...	...	...	167	501	781	3819
Wash.	Seattle	790	669	623	468	326	180	59	59	207	422	582	722	5107
	Spokane	1162	944	784	498	294	72	...	...	174	518	795	1071	6312
W. Va.	Elkins	1073	935	775	486	180	3	...	...	71	394	741	1001	5659
	Parkersburg	1008	862	688	348	55	...	...	...	10	276	636	924	4807
Wis.	Green Bay	1528	1333	1128	654	313	35	...	...	139	512	930	1324	7896
	LaCrosse	1516	1282	1038	534	177	1	...	...	87	456	894	1324	7309
	Milwaukee	1376	1182	1020	636	338	52	...	...	80	431	831	1206	7152
Wyo.	Cheyenne	1224	1056	989	723	456	138	...	13	240	626	906	1132	7503
	Lander	1448	1190	1011	678	428	135	...	18	279	666	1041	1383	8277

PROVINCE	CITY	JAN.	FEB.	MAR.	APR.	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.	TOTAL
Alta.	Calgary	1674	1428	1240	750	496	270	124	186	450	744	1170	1395	9,927
	Edmonton	1829	1512	1302	720	434	270	124	186	450	713	1230	1519	10,289
B. C.	Vancouver	899	756	713	510	341	180	62	31	270	496	660	837	5,555
Man.	Winnipeg	2139	1820	1581	810	465	90	...	62	270	744	1320	1829	11,130
N. B.	Moncton	1519	1428	1178	810	465	210	...	93	300	620	930	1333	8,886
N. S.	Halifax	1302	1176	1085	780	496	210	...	...	210	496	780	1147	7,682
Ont.	Ottawa	1674	1484	1271	690	279	30	...	...	210	589	990	1457	8,676
	Port Arthur	1829	1624	1426	900	558	240	62	86	360	713	1140	1550	10,588
	Toronto	1333	1204	1209	720	372	60	...	...	180	558	870	1209	7,715
P.E.I.	Charlottetown	1178	1120	1209	870	529	210	...	...	600	558	870	1240	8,382
Que.	Montreal	1581	1428	1209	720	310	...	...	...	180	558	960	1395	8,341
	Quebec	1705	1484	1333	870	434	120	...	31	270	651	1050	1519	9,467
Sask.	Saskatoon	2108	1820	1581	810	465	210	62	155	450	806	1290	1736	11,493

[a]Figures for United States cities taken from Degree-Day Normals over the United States, by A. G. Topil (*Monthly Weather Review*, U. S. Weather Bureau, July, 1937, p. 266). Figures for Canadian Cities abstracted from *Heating & Ventilating*, October, 1939.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant (U) for gas are given in Table 4 for various gas heating values, and different types and sizes of heating plants. *They are based on an inside design temperature of 70 F and an outside design temperature of 0 F and apply only to these conditions.* For other design conditions corrections must be made as given in Table 5. Estimates for industrial buildings where low inside temperatures are maintained cannot be made from this table.

The factors in Table 4, as corrected if necessary, are satisfactory for regions having 3500 to 6500 degree-days per heating season. In regions with less than 3500 degree-days the unit gas consumption is higher than given; where over 6500, the unit is less than given. Ten per cent addition or deduction in these cases is recommended by *AGA* publications.

For gas heating values other than those given in Table 4, simply interpolate or extrapolate. It will also be noted that Table 4 applies only to small installations. In general the larger the installation the smaller the unit gas consumption becomes and the values in the table should be used with care, if at all, in large gas-burning installations.

Example 7. Make an estimate of the gas required to heat a building located in Chicago, Ill., assuming that the calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperature of 0 F and 70 F. Chicago has 800 Btu mixed gas, and 6027 degree-days.

Solution. Using Equation 3 and Table 4, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0.085 cu ft of gas per degree-day per square foot of hot water radiation.

$$0.085 \times 1000 \times 6027 = 512{,}295 \text{ cu ft.}$$

Estimating Oil Consumption

Unit fuel consumption factors for oil, similar to those for gas in Table 4 are given in Table 6. *The factors in Table 6 apply only to an inside design*

TABLE 4. FACTORS FOR ESTIMATING GAS CONSUMPTION[a]

Btu Value of Gas per Cu Ft	Hot Water			Steam			Warm Air	
	Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per 1000 Btu Hourly Design Heat Loss	
	Up to 500 Sq Ft	500 to 1200 Sq Ft	Over 1200 Sq Ft	Up to 500 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.142	0.135	0.128	0.242	0.231	0.220	0.855	0.820
535	0.132	0.126	0.120	0.226	0.215	0.206	0.800	0.766
800	0.089	0.085	0.081	0.151	0.144	0.137	0.534	0.513
1000	0.071	0.068	0.065	0.121	0.115	0.110	0.428	0.410
1 Therm = 100,000 Btu	Gas Consumption in Therms per Degree-Day							
	0.000708	0.000675	0.000642	0.00121	0.00115	0.00110	0.00428	0.00409

[a]Abstracted from Comfort Heating, *American Gas Association*, 1938.

TABLE 5. CORRECTION FACTORS FOR OUTSIDE DESIGN TEMPERATURES

Outside Design Temp. Deg F	Inside Design Temp. Deg F	Multiply Values in Tables 4, 6 and 7 by
−20	70	0.778
−10	70	0.875
0	70	0.000
+10	70	1.167
+20	70	1.400

temperature of 70 F and an outside design temperature of 0 F. For other outside design temperatures, the constants in Table 6 must be multiplied by the values in Table 5 as explained under Estimating Gas Consumption.

Table 6 assumes the use of oil with a heating value of 140,000 Btu per gallon. For other heating values, multiply the values in Table 6 by the ratio of 140,000 divided by the heating value per gallon of fuel being used.

Example 8. What would be the estimated seasonal oil consumption of a boiler-burner unit in Minneapolis of a building having a calculated heat loss of 192,000 Btu per hour, burning 144,000 Btu per gallon oil and operating at a seasonal efficiency of 60 per cent, if the outside design temperature for Minneapolis is −20 F, and the inside design temperature is 70 F?

Solution. From Table 6, under 60 per cent efficiency and opposite the bottom column, find the uncorrected U to be 0.00476 gal per 1000 Btu hourly heat loss.

From Table 5, the correction multiplier for −20 F outside design temperature is 0.778. Solving, $0.778 \times 0.00476 = 0.00370$. Making a further correction for the heating value:

$0.0037 \times \frac{140{,}000}{144{,}000} = 0.0036$ gal per 1000 Btu per hour calculated heat loss per degree-day.

From Table 3, the average degree-days for Minneapolis number 7883, and from the problem $N = 192$. Substituting in Equation 3:

$$F = 0.0036 \times 7883 \times 192 = 5449 \text{ gal.}$$

TABLE 6. UNIT FUEL CONSUMPTION[a] CONSTANTS FOR OIL[b]

Unit	Efficiency in Per Cent				
	40	50	60	70	80
Gal Oil per Sq Ft Steam Radiator	0.00172	0.00137	0.00114	0.00098	0.00086
Gal Oil per Sq Ft Hot Water Radiator	0.00108	0.00086	0.00072	0.00062	0.00054
Gal Oil per 1000 Btu per Hour Heat Loss	0.00715	0.00571	0.00476	0.00409	0.00358

[a]Based on a heating value of 140,000 Btu per gallon.

[b]Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.

TABLE 7. UNIT FUEL CONSUMPTION[a] CONSTANTS FOR COAL[b]

Unit	Efficiency in Per Cent				
	40	50	60	70	80
Lb Coal per Sq Ft Steam Radiator	0.0200	0.0160	0.0133	0.0114	0.0100
Lb Coal per Sq Ft Hot Water Radiator	0.0125	0.0100	0.0084	0.0072	0.0063
Lb Coal per 1000 Btu per Hour Heat Loss	0.0825	0.0666	0.0550	0.0471	0.0412

[a]Based on a heating value of 12,000 Btu per pound.

[b]Abstracted by permission from *Degree-Day Handbook*, (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.

Estimating Coal or Coke Consumption

Coal or coke consumption estimates can be made in exactly the same way as for oil. The uncorrected values of U are given in Table 7. *These constants apply only to inside design temperatures of 70 F and an outside design temperature of 0 F*, and correction must be made for other conditions by use of the multiplying factors in Table 5. Table 7 is based on 12,000 Btu per pound coal and for other heating values of coal, values in Table 7 must be multiplied by the ratio of 12,000 divided by the heating value of fuel used.

Example 9. A building in Marquette, Mich., has an hourly heat loss at design conditions of 240,000 Btu per hour. If the inside design temperature is to be 70 F and the outside design temperature is -10 F, what will be the estimated normal seasonal coal consumption for heating if 12,000 Btu per pound fuel is burned at a 50 per cent seasonal efficiency, and what part of the total will be used during November, December, and January?

Solution. From Table 7, U uncorrected, is 0.0666 lb of coal per 1000 Btu per hour heat loss. Correcting for the outside design temperature, Table 5, the corrected value of U is $0.875 \times 0.0666 = 0.0583$. From Table 3, D is 8721 and from the problem N is 240.

Substituting in Equation 3:

$$F = 0.0583 \times 240 \times 8721 = 122{,}024 \text{ lb.}$$

Fuel used over any period is, according to the theory of the degree-day, proportional to the number of degree-days during the period. From Table 3, the average number of degree-days for November, December, and January in Marquette are 951, 1314, and 1510, a total of 3775. The yearly total is 8721, so that during these three months the estimated consumption is:

$$\frac{3775}{8721} \times 122{,}024 = 52{,}820 \text{ lb.}$$

Estimating Steam Consumption

In estimating steam consumption the efficiency is not ordinarily a factor and is assumed at 100 per cent. Ordinarily low pressure steam with a heating value of 1000 Btu per pound is used so that no correction is necessary for heating value in the usual case. In comparing values from different cities, correction should be made for design temperature (see Table 5) when the unit figures are in terms of square foot of radiator or 1000 Btu per hour calculated heat loss, but not when the values are in terms of building volume or floor space.

Consideration has been given to the difference in steam utilization of different types of buildings and Table 8[4] shows actual average units for these various types. These figures are obtained from operating results in 196 buildings located in 21 different cities in the United States. Being averages, and for small groups in each type, the figures may need considerable modification to allow for local variations. It should be especially noted that the steam used for heating hot water is included in the values given in Table 8, but in the case of office buildings, the steam for heating only is also shown. Presentation of the unit consumption in three ways permits making the estimate if either the calculated heat loss or the volume of net heated space in the building is known.

[4]The Heat Requirements of Buildings, by J. H. Walker and G. H. Tuttle (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 171).

TABLE 8. STEAM CONSUMPTION FOR VARIOUS CLASSES OF BUILDINGS[a]
(Heating Season Only)

Building Classification	No. of Buildings Listed	Steam Consumption Pounds per Degree-Day—65 F Basis[d]		
		Per M Cu Ft of Heated Space	Per M Sq Ft of Radiator[c] Surface	Per M Btu per Hr of Heat Loss[b]
Apartments	16	1.78	97.5	0.359
Hotels	10	1.46	80.6	0.371
Residences	12	1.32	64.2	....
Printing	7	1.25	105.5	....
Clubs and Lodges	10	0.96	77.0	....
Retail Stores	18	0.90	80.6	0.268
Theaters	6	0.90	75.0	0.498
Loft and Mfg.	16	0.89	72.3	0.283
Banks	7	0.88	45.2	....
Auto Sales and Service	8	0.83	62.2	....
Churches	6	0.58	49.4	....
Department Stores	14	0.57	60.7	0.238
Garages (Storage)[e]	6	0.42	72.3	....
Offices (Total)	35	1.09	70.0	0.283
Offices (Heating only)	35	0.975	65.4	0.256

[a]Includes steam for heating domestic water for heating season only.
[b]Heat loss calculated for maximum design condition (in most cases 70 F inside, zero outside).
[c]Equivalent steam radiator surface.
[d]The figures are a numerical—not a weighted—average for the several buildings in each class.
[e]Based on zero consumption at 55 F.

Example 10. A store in Detroit with a heating system designed to maintain 70 F inside in 0 F weather has 1700 sq ft of equivalent direct steam radiator surface, and uses only moderate quantities of hot water. What would be the estimated average yearly steam consumption of purchased steam for heating and for hot water during the heating season?

Solution. According to Table 8, a store (0 to 70 F conditions) would use 80.6 lb of steam per thousand square feet of radiation per degree-day, including winter hot water. From Table 3, Detroit has 6460 degree-days per normal year. Inserting in Equation 3:

$$F = 80.6 \times 1.7 \times 6460 = 885{,}149 \text{ lb of steam.}$$

TABLE 9. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS[a]

Building Classification	Load Factor	Lb of Demand per Hr per Sq Ft of Equivalent Installed Radiator Surface
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theaters	0.126	0.151

[a]Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 9.

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization habits. Thus, in Table 9, the theaters, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

SEASONAL EFFICIENCY

The task of predicting fuel consumption within reasonably accurate limits is a simple one where sufficient experience data are available for the fuel in question. Such data can be analyzed to the point where average unit factors can be determined and expressed in such terms as, for example, average gallons of oil actually burned per square foot of calculated steam radiator surface per degree-day. The unit U can be inserted directly in Equation 3 without reference to efficiency. Such experience factors are available for gas (see Table 4) and for district steam (Table 8), but not for coal or oil.

Since values of U are not available for oil or coal, an assumed seasonal efficiency E must be used. Selection of a value for this E must be made with caution, for its use implies a meaning not commonly associated with the word efficiency and consequently is frequently misleading.

The input of heat to a building consists not only of the energy in the fuel but that from occupants, the sun, appliances, processes, and all other sources. In many cases these make up, over a period, an important percentage of the total heat required, and if they are not taken into account a calculation of *efficiency* can show a figure over 100 per cent.

For this and other reasons the actual seasonal efficiency is a difficult thing to determine. Published data are widely scattered and insufficient. From the available published material it is found that the seasonal efficiency varies over a wide range, depending on the fuel used, and it varies widely even for a given fuel. For example, in a recent survey of 30 houses in one locality there was found a variation of from 45 to 75 per cent in the utilization efficiency depending on the fuel[5].

[5]Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross, (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 185).

Chapter 11

HEATING BOILERS

Cast-Iron Boilers, Steel Boilers, Special Heating Boilers, Gas-Fired Boilers, Hot Water Supply Boilers, Furnace Design, Heating Surface, Testing and Rating Codes, Output Efficiency, Selection of Boilers, Connections and Fittings, Erection, Operation and Maintenance, Boiler Insulation

STEAM and hot water boilers for low pressure heating work are built in a wide variety of types, many of which are illustrated in the *Catalog Data Section*, and are classified as (1) cast-iron sectional, (2) steel fire tube, (3) steel water tube, and (4) special.

CAST-IRON BOILERS

Cast-iron boilers usually fall into one of two general classifications (1) rectangular pattern with vertical sections and rectangular grate commonly known as sectional boilers, (2) round pattern with horizontal pancake sections and circular grates commonly known as round boilers. A few boilers of the sectional type use outside header construction where each section is independent of the other and the water and steam connections are made externally through these headers. The majority of boilers, however, both sectional and round, are assembled with push nipples and tie rods at the top and bottom of the sections in which case water and steam connections are internal. The present trend in design of sectional boilers is to use a large top push nipple so that in a steam boiler the water line may be carried through the top push nipple thereby permitting circulation of water between adjacent sections at both the top and bottom of the water content of the boiler. The primary purpose of this construction is to eliminate the necessity of connecting the sections below the water line with an external header to permit circulation of the water from one section to the other which is necessary in the case of a steam boiler equipped with an indirect water heater for summer-winter hot water supply.

Round and sectional boilers may be increased in size by the addition of sections which in the case of sectional boilers also increases the grate area. The grate area of round boilers remains the same as additional sections are added.

Cast-iron boilers are usually shipped knocked down. This facilitates handling at the place of installation where assembly is made in that separate sections can be taken into or out of basements and other places more or less inaccessible after the building is constructed. This feature

is of importance in the original installation of the boiler and also in making repairs to or replacing a damaged boiler at a later date.

Cast-iron boilers may be designed to burn efficiently one kind of fuel only or various kinds of fuel. Practical combustion rates for coal-fired boilers are given in Table 1. Many recent designs of oil burning boilers have been designed exclusively for oil fuel and in some cases for both oil and gas. The present trend in the design of boilers for hand firing is, however, to design them so they will be suitable for ready conversion to and efficient operation with oil and stoker firing even after the boiler has been installed and has been operating for a period of time on one type of fuel.

Table 1. Practical Combustion Rates for Coal-Fired Heating Boilers Operating at Maximum Load on Natural Draft of from 1/8 in. to 1/2 in. Water[a]

Kind of Coal	Sq Ft Grate	Lb of Coal per Sq Ft Grate per Hour
No. 1 Buckwheat Anthracite	Up to 4 5 to 9 10 to 14 15 to 19 20 to 25	3 3½ 4 4½ 5
Anthracite Pea	Up to 9 10 to 19 20 to 25	5 5½ 6
Anthracite Nut and Larger	Up to 4 5 to 9 10 to 14 15 to 19 20 to 25	8 9 10 11 13
Bituminous	Up to 4 5 to 14 15 and above	9.5 12 15.5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

Capacities of cast-iron boilers range from that required for small residences up to about 18,000 sq ft of radiation. For larger loads boilers must be installed in multiple. The maximum allowable working pressure for cast-iron boilers is limited by the *A.S.M.E.* Code to 15 lb per square inch for steam boilers. Hot water boilers are usually limited to 30 lb per square inch maximum working pressure but may be designed for higher pressures where required for heating purposes or for hot water supply where the boiler must withstand high local water pressures.

STEEL BOILERS

Steel heating boilers may be classified according to (a) position of combustion gas with respect to tube surface, (b) arrangement and construction of furnaces, and (c) type of fuel and method of firing.

Fire tube boilers are those in which the gases of combustion pass through the tubes and the boiler water circulates around them. In water

tube boilers, the gases circulate around the tubes and the water passes through them.

Steel heating boilers may be furnished with integral water jacketed furnaces or arranged for refractory lined brick or refractory lined jacketed furnaces. Those with integral water jacketed furnaces are called *portable firebox boilers* and are the most commonly used type. They may be either fire tube or water tube and are furnished for any fuel and method of firing used in heating boiler practice. They are usually shipped from the factory in one piece, ready for piping. Bridge-walls and smokeless furnace parts are shipped in place when furnished. Boilers with refractory lined furnaces may be either fire tube or water tube. They also may be arranged for any fuel or method of firing. Refractory furnaces are usually installed in such boilers after they are set.

SPECIAL HEATING BOILERS

A special type of boiler, known as the *magazine feed boiler*, has been developed for the burning of small sizes of anthracite and coke. These are built of both cast-iron and steel, and have a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types using buckwheat sizes of coal. Special attention must be given to insure adequate draft and proper chimney sizes and connections.

Oil-burner boiler units, in which a special boiler has been designed with a furnace shaped to meet the general requirements of oil burners, or are specially adapted to one particular burner, have been developed by a number of manufacturers. These usually are compact units with the burner and all controls enclosed within an insulated steel jacket. Ample furnace volume is provided for efficient combustion, and the heating surfaces are proportioned for effective heat transfer. Consequently, higher efficiencies are obtainable than with the ordinary coal-fired boiler designed primarily for hand firing and converted to oil firing.

GAS-FIRED BOILERS

Gas boilers have assumed a well-defined individuality. The usual boiler is sectional in construction with a number of independent burners placed beneath the sections. In most boilers each section has its own burner. In all cases the sections are placed quite closely together, much closer than would be possible when burning a soot-forming fuel. The effort of the designer is always to break the hot gas up into thin streams, so that all particles of the heat-carrying gases can come as close as possible to the heat-absorbing surfaces. Because there is no fuel bed resistance and because the gas company supplies the motive power to draw in the air necessary for combustion (in the form of the initial gas pressure), draft losses through gas boilers are low. (See Chapter 9.)

Most gas-fired boilers carry the approval of the *American Gas Association*. In order to obtain this approval the boilers must be submitted to the *American Gas Association Testing Laboratory* and meet the Approval Requirements for Central Heating Gas Appliances issued by the *American Gas Association*. The boiler ratings must be such that they meet the limitations as set forth in these Approval Requirements.

HOT WATER SUPPLY BOILERS

Boilers for hot water supply are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-making properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam heating system is installed, the domestic hot water usually is obtained from an indirect heater placed below the water line of the boiler. Indirect heaters may also be used with hot water heating systems to obtain domestic hot water and should be located as high as possible with respect to the boiler for most satisfactory performance.

FURNACE DESIGN

Good efficiency and proper boiler performance are dependent on correct furnace design embodying sufficient volume for burning the particular fuel at hand, which requires thorough mixing of air and gases at a high temperature with a velocity low enough to permit complete combustion of all the volatiles. On account of the small amount of volatiles contained in coke, anthracite, and semi-bituminous coal, these fuels can be burned efficiently with less furnace volume than is required for bituminous coal, the combustion space being proportioned according to the amount of volatiles present.

Combustion should take place before the gases are cooled by the boiler heating surface, and the volume of the furnace must be sufficient for this purpose. The furnace temperature must be maintained sufficiently high to produce complete combustion, thus resulting in a higher CO_2 content and the absence of CO. Hydrocarbon gases ignite at temperatures varying from 1000 to 1500 F.

The question of furnace proportions, particularly in regard to mechanical stoker installations, has been given some consideration by various manufacturers' associations. Arbitrary values have been recommended for minimum dimensions. A customary rule-of-thumb method of figuring furnace volumes is to allow 1 cu ft of space for a maximum heat release

of 50,000 Btu per hour. This value is equivalent to allowing approximately 1 cu ft for each developed horsepower, and it is approved by most smoke prevention organizations.

The setting height will vary with the type of stoker. In an overfeed stoker, for instance, all the volatiles must be burned in the combustion chamber and, therefore, a greater distance should be allowed than for an underfeed stoker where a considerable portion of the gas is burned while passing through the incandescent fuel bed. The design of the boiler also may affect the setting height, since in certain types the gas enters the tubes immediately after leaving the combustion chamber, while in others it passes over a bridge wall and toward the rear, thus giving a better opportunity for combustion by obtaining a longer travel before entering the tubes.

To secure suitable furnace volume, especially for mechanical stokers or oil burners, it often is necessary either to pit the stoker or oil burner, or, where water line conditions and headroom permit, to raise the boiler on a brick foundation setting.

Smokeless combustion of the more volatile bituminous coals is furthered by the use of mechanical stokers. (See Chapter 9.) Smokeless combustion in hand-fired boilers burning high volatile solid fuel is aided (1) by the use of double grates with down-draft through the upper grate, (2) by the use of a curtain section through which preheated auxiliary air is introduced over the fire toward the rear of the boiler, and (3) by the introduction of preheated air through passages at the front of the boiler. All three methods depend largely on mixing secondary air with the partially burned volatiles and causing this mixture to pass over an incandescent fuel bed, thus tending to secure more complete combustion than is possible in boilers without such provision.

HEATING SURFACE

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed. The heat transfer capacity of a radiant heating surface may be as high as 6 to 8 times that of an indirect surface. This is one of the reasons why the water legs of some boilers have been extended, especially in the case of stoker firing where the extra amount of combustion chamber secured by an extension of the water legs is important. For the same reason, care should be exercised in building a refractory combustion chamber in an oil-burning boiler so as not to screen any more of this valuable surface with refractories than is necessary for good combustion.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. The area of the

gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed. Inserting baffles so that the heating surface is arranged in series with respect to the gas flow increases boiler efficiency and reduces stack temperature and increases the draft loss through the boiler.

Heat Transfer Rates

Practical rates of heat transfer in heating boilers will average about 3300 Btu per square foot per hour for hand-fired boilers and 4000 Btu per square foot per hour for mechanically fired boilers when operating at *design load.* When operating at *maximum load*[1] these values will run between 5000 and 6000 Btu per square foot per hour. Boilers operating under favorable conditions at the above heat transfer rates will give exit gas temperatures that are considered consistent with good practice.

TESTING AND RATING CODES

The Society has adopted four solid fuel testing codes, a solid fuel rating code and an oil fuel testing code. A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June 1929)[2], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics. A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[2] is intended for use with A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[3]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler. The A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[4] is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers. In 1938 the Society adopted a Standard Code for Testing Stoker-Fired Steam Heating Boilers[5] which is intended to provide a test method for determining the efficiency and performance characteristics of any stoker or boiler combination burning any type of solid fuel such as anthracite or bituminous coal.

The *Steel Heating Boiler Institute* has adopted a method for the rating of low pressure boilers based on their physical characteristics and expressed in square feet of steam or water radiation or in Btu per hour as given in Table 2. The detailed requirements of this code were outlined in Chapter 13 of THE GUIDE 1939. The *Institute of Boiler and Radiator Manufacturers* has also adopted a method of rating cast-iron heating boilers based upon performance obtained under tests. This code became effective August 1, 1939 for sectional boilers of 20 in. width grate or less, but the *Institute* intends eventually to expand the code to apply to all

[1]For definitions of design load and maximum load see page 228.
[2]See A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, pp. 322 and 332.
[3]See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 42.
[4]See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 23.
[5]See A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 366.

TABLE 2. STEEL HEATING BOILER STANDARD RATINGS[a]

Hand-Fired Rating						Mechanically-Fired Rating				
Catalog			Net Load	Heating Surface Sq Ft	Grate Area Sq Ft	Catalog			Net Load	Furnace Volume, Oil, Gas or Bituminous Coal Cu Ft
Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hour in Thousands	Steam Radiation Sq Ft			Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hour in Thousands	Steam Radiation Sq Ft	
1,800	2,880	432	1,389	129	7.9	2,190	3,500	525	1,695	15.7
2,200	3,520	528	1,702	158	8.9	2,680	4,280	643	2,089	19.2
2,600	4,160	624	2,020	186	9.7	3,160	5,050	758	2,461	22.6
3,000	4,800	720	2,335	215	10.5	3,650	5,840	876	2,853	26.1
3,500	5,600	840	2,732	250	11.4	4,250	6,800	1,020	3,335	30.4
4,000	6,400	960	3,135	286	12.2	4,860	7,770	1,166	3,830	34.8
4,500	7,200	1,080	3,540	322	13.4	5,470	8,750	1,312	4,330	39.1
5,000	8,000	1,200	3,945	358	14.5	6,080	9,720	1,459	4,834	43.5
6,000	9,600	1,440	4,770	429	16.4	7,290	11,660	1,749	5,850	52.1
7,000	11,200	1,680	5,608	500	18.1	8,500	13,600	2,040	6,885	60.8
8,500	13,600	2,040	6,885	608	20.5	10,330	16,520	2,479	8,490	73.8
10,000	16,000	2,400	8,197	715	22.5	12,150	19,440	2,916	10,125	86.8
12,500	20,000	3,000	10,417	893	25.6	15,180	24,280	3,643	12,650	108.5
15,000	24,000	3,600	12,500	1,072	28.4	18,220	29,150	4,372	15,183	130.2
17,500	28,000	4,200	14,584	1,250	30.9	21,250	34,000	5,100	17,708	151.8
20,000	32,000	4,800	16,667	1,429	33.2	24,290	38,860	5,829	20,242	173.5
25,000	40,000	6,000	20,834	1,786	37.4	30,360	48,570	7,286	25,300	216.9
30,000	48,000	7,200	25,000	2,143	41.2	36,430	58,280	8,743	30,359	260.3
35,000	56,000	8,400	29,167	2,500	44.7	42,500	68,000	10,200	35,417	303.6

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards, United States Department of Commerce Simplified Practice Recommendations R 157-35.*

sizes of boilers. Methods of testing hand-fired and oil-fired boilers are specified and are referred to as *IBR* testing codes.

BOILER OUTPUT

Boiler output as defined in A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) is the quantity of heat available at the boiler nozzle with the boiler normally insulated. It should be based on actual tests conducted in accordance with this code. This output is usually stated in Btu and in square feet of equivalent heating surface (radiation). According to the A.S.H.V.E. Standard Code for Rating Steam Heating, Solid Fuel Hand-Fired Boilers, the performance data should be given in tabular or curve form on the following items for at least five outputs ranging from maximum down to 35 per cent of maximum: (1) fuel available, (2) combustion rate, (3) efficiency, (4) draft tension, (5) flue gas temperature. The only definite restriction placed on setting the maximum output is that priming shall not exceed 2 per cent. These curves provide complete data regarding the performance of the boiler under test conditions. Certain other pertinent information, such as grate area, heating surface and chimney dimensions is desirable also in forming an opinion of how the boiler will perform in actual service.

The output of large heating boilers is frequently stated in terms of *boiler horsepower* instead of in Btu per hour or square feet of equivalent radiation.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid and Gaseous Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound or cubic foot of fuel to the calorific value of 1 lb or cubic foot of fuel respectively.

Solid fuel boilers usually show an efficiency of 50 to 75 per cent when operated under favorable conditions at their rated capacities. Information on the combined efficiencies of boiler, furnace and burner has resulted from research conducted at Yale University in cooperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*[6].

SELECTION OF BOILERS

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined, is equivalent to the sum of the heat emission of the radiation to be actually installed, the allowance for the heat loss of the connecting piping, and the heat requirement for any apparatus requiring heat connected to the system.

The estimated design load is the sum of the following three items[7]:

1. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect or central fan) to be installed.

2. The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler.

3. The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or the equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry.

The estimated maximum load is given by[8]:

4. The estimated increase in the normal load in Btu per hour due to starting up cold radiation. This percentage of increase is to be based on the sum of Items 1, 2 and 3 and the heating-up factors given in Table 3.

Other things to be considered are:

5. Efficiency with hard or soft coal, gas, or oil firing, as the case may be.

---

[6]A.S.H.V.E. RESEARCH REPORT No. 907—Study of the Characteristics of Oil Burners and Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 517). A.S.H.V.E. RESEARCH REPORT No. 925—A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 317).

[7]A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

[8]Loc. Cit. Note 7.

6. Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas.
7. Combustion space in the furnace.
8. Type of heat liberation, whether continuous or intermittent, or a combination of both.
9. Miscellaneous items consisting of draft available, character of attendance, possibility of future extension, possibility of breakdown and headroom in the boiler room.

Radiation Load

The connected radiation (Item 1) is determined by calculating the heat losses in accordance with data given in Chapters 3, 4 and 5, and dividing by 240 to change to square feet of equivalent radiation as explained in Chapter 12. For hot water, the emission commonly used is 150 Btu per square foot, but the actual emission depends on the temperature of the medium in the heating units and of the surrounding air. (See Chapter 12.)

Although it is customary to use the actual connected load in equivalent square feet of radiation for selecting the size of boiler, this connected load usually represents a reserve in heating capacity to provide for infiltration

TABLE 3. WARMING-UP ALLOWANCES FOR LOW PRESSURE STEAM AND HOT WATER HEATING BOILERS[a, b, c]

Design Load (Representing Summation of Items 1, 2, and 3,[d]		Percentage Capacity to Add for Warming Up
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggert (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 292); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 35); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to hand-fired, solid fuel boilers. A factor of 20 per cent over design load is adequate when automatically-fired fuels are used (see Fig. 1).

[d]240 Btu per square foot.

in the various spaces of the building to be heated, which reserve, however, is not in use at all places at the same time, or in any one place at all times. For a further discussion of this subject see Chapter 4.

Hot Water Supply Load

When the hot water supply (Item 2) is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. The allowance to be made will depend on the amount of water heated and its temperature rise. A good approximation is to add 4 sq ft of equivalent radiation for each gallon of water heated per hour through a temperature range of 100 F. For more specific information, see Chapter 45.

Piping Tax (Item 3)

It is common practice to add a flat percentage allowance to the equivalent connected radiation to provide for the heat loss from bare and covered pipe in the supply and return lines. The use of a flat allowance of

25 per cent for steam systems and 35 per cent for hot water systems is preferable to ignoring entirely the load due to heat loss from the supply and return lines, but better practice, especially when there is much bare pipe, is to compute the emission from both bare and covered pipe surface in accordance with data in Chapter 42. A chart is shown in Fig. 1 indicating percentage allowances for piping and warming-up which are applicable to automatically-fired heating plants using steam radiation. With direct radiation served by bare supply and return piping the percentages may be higher than those stated, while in the case of unit heaters where the output is concentrated in a few locations, the piping tax may be 10 per cent or less.

Warming-Up Allowance

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature and heating up cold radiation and piping. (See Item 4.) The factors to be used for determining the

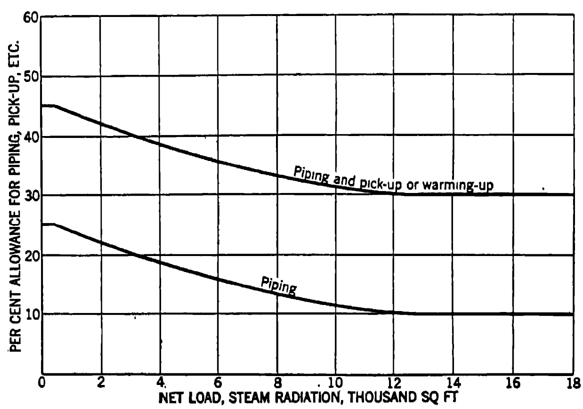


FIG. 1. PERCENTAGE ALLOWANCE FOR PIPING AND WARMING-UP

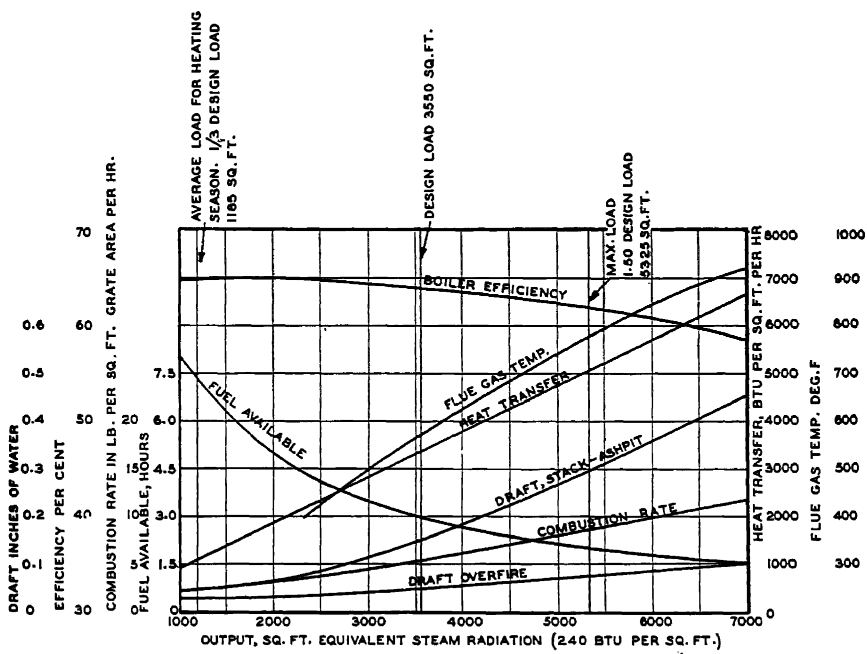
allowance to be made should be selected from Table 3 and should be applied to the estimated design load as determined by Items 1, 2 and 3. While in every case the estimated maximum load will exceed the design load if adequate heating response is to be achieved, there is, however, no object in over-estimating the allowances, as the only effect would be to reduce the time of warming-up by a few minutes. Otherwise, it might result in firing the boiler unduly and increasing the cost of operation.

Performance Curves for Boiler Selection

In the selection of a boiler to meet the estimated load, the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers recommends the use of performance curves based on actual tests conducted in accordance with the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3), similar to the typical curves shown in Fig. 2. It should be understood that performance data apply to test conditions and that a reasonable allowance should be made for decreased output resulting from soot deposit, poor fuel or inefficient attention.

Selection Based on Heating Surface and Grate Area

Where performance curves are not available, a good general rule for conventionally-designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load consisting of connected radiation, piping tax and domestic water heating load. As stated in the section on Boiler Output, this is equivalent to allowing 10 sq ft of boiler heating surface per



BOILER DATA—

GRATE AREA, SQ. FT.	18.0
HEATING SURFACE, SQ. FT.	254
WEIGHT, LB.	9160
FUEL CAPACITY, LB.	651
FUEL AVAILABLE, LB.	414
FUEL DEPTH, IN	10

FUEL—BITUMINOUS 3/4" LUMP

ANALYSIS—

VOLATILE MATTER	34.68%
FIXED CARBON	55.44
ASH	9.87
SULPHUR	2.86
MOISTURE	3.00
BTU PER LB.	13,655

FIG. 2. TYPICAL PERFORMANCE CURVES FOR A 36-IN. CAST-IRON SECTIONAL STEAM HEATING BOILER, BASED ON THE A.S.H.V.E. CODE FOR RATING STEAM HEATING SOLID FUEL HAND-FIRED BOILERS

boiler horsepower. In this case it is assumed that the maximum load including the warming-up allowance will be provided for by operating the boiler in excess of the design load, that is, in excess of the 100 per cent rating on a boiler-horsepower basis.

Due to the wide variation encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

G = grate area, square feet.

H = required total heat output of the boiler, Btu per hour (see Selection of Boilers, p. 228).

C = combustion rate in pounds of dry coal per square foot of grate area per hour, depending on the kind of fuel and size of boiler as given in Table 1.

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required heat output of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Formula 4. With small boilers where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of Steel Heating Boilers

Ratings obtained from the previously mentioned *Steel Heating Boiler Institute* code are intended to correspond with the estimated design load based on the sum of items 1, 2 and 3 outlined on page 228. Boilers with less than 128 sq ft of heating surface are classified as residence size. An insulated residence boiler for oil or gas, not convertible, may carry a net load expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler manufacturer guarantees it to be capable of operating at a maximum output of not less than 150 per cent of net load rating with overall efficiency of not less than 75 per cent with at least two different makes of each type of standard commercial burner recommended by the boiler manufacturer. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the net load.

When the estimated heat emission of the piping (connecting radiation, and other apparatus to the boiler) is not known the net load to be considered for the boiler may be determined from Table 2.

Selection of Gas-Fired Boilers

Gas-heating appliances should be selected in accordance with the percentage allowances given in Fig. 1. These factors are for thermostatically-controlled systems; in case manual operation is desired, a warming-up allowance of 100 per cent is recommended by the *A.G.A.* A gas boiler selected by the use of the *A.G.A.* factors will be the minimum size boiler which can carry the load. From a fuel economy standpoint, it may be advisable to select a somewhat larger boiler and then throttle the gas and air adjustments as required. This will tend to give a low stack temperature with high efficiency and at the same time provide reserve capacity in case the load is under-estimated or more is added in the future.

Conversions

The conversion of a coal or oil boiler to gas burning is simpler than the reverse since little furnace volume need be provided for the proper combustion of gas. When a solid fuel boiler of 500 sq ft (or less) capacity is converted to gas burning, the necessary gas heat units should be approximately double the connected load. The presumption for a conversion job is that the boiler is installed and probably will not be made larger; therefore, it is a matter of setting a gas-burning rate to obtain best results with the available surface. Assuming a combustion efficiency of 75 per cent for a conversion installation the boiler output would be $2 \times 0.75 = 1.5$ times the connected load, which allows 50 per cent for piping tax and pickup. In converting large boilers, the determination of the required Btu input should not be done by an arbitrary figure or factor but should be based on a detailed consideration of the requirements and characteristics of the connected load.

An efficient conversion installation depends upon the proper size of flue connection. Often the original smoke breeching between the boiler and chimney is too large for gas firing, and in this case, flue orifices can be used. They are discs provided with an opening of the size for the gas input used in this boiler. The size should be based on 1 sq in. of flue area for each 7500 hourly Btu input.

If dampers are found in the breeching they should be locked in position so that they will not interfere with the normal operation of the gas burners at maximum flow. In the case of large boiler conversions, automatic damper regulators proportion the position of the flue dampers to the amount of gas flowing and may be substituted for existing dampers. Generally in residence conversions automatic dampers are not of the proportioning type but close the flue during the off periods of the gas burners. Automatic shut-off dampers should be located between the back draft diverter and the chimney flue. Automatic dampers are usually designed to operate with electric contact mechanism, but frequently an arrangement is utilized which functions with mechanical fluid or gas pressure.

Physical Limitations

As it will usually be found that several boilers will meet the specifications, the final selection may be influenced by other considerations, as:

1. Dimensions of boiler.
2. Durability under service.
3. Convenience in firing and cleaning.
4. Adaptability to changes in fuel and kind of attention.
5. Height of water line.

In large installations, the use of several smaller boiler units instead of one larger one will obtain greater flexibility and economy by permitting the operation, at the best efficiency, of the required number of units according to the heat requirements.

Space Limitations

Boiler rooms should, if possible, be situated at a central point with respect to the building and should be designed for a maximum of natural

4. Condensation must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.

5. Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level are recommended for boilers mechanically fired.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; and (*h*) insufficient radiation installed.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; and (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

8. *Low carbon dioxide.* This may be due on oil burning boilers to: (*a*) improper adjustment of the burner; (*b*) leakage through the boiler setting; (*c*) improper fire caused by a fouled nozzle; or (*d*) to an insufficient quantity of oil being burned.

Cleaning Steam Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom

of the boiler and form sludge. These impurities have a tendency to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least 1¼ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When pressure recedes, close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

It is common practice when starting new installations to discharge heating returns to the sewer during the first week of operation. This prevents the passage of grease, dirt or other foreign matter into the boiler and consequently may avoid the necessity of cleaning the boiler. During the time the returns are being passed to the sewer, the feed valve should be cracked sufficiently to maintain the proper water level in the boiler.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.

2. All machined surfaces should be coated with oil or grease.

3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.

4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard of some one inadvertently building a fire in a dry boiler, however, it is safer to keep the boiler filled with water. A hot water system usually is left filled to the expansion tank.

5. The grates and ashpit should be cleaned.

6. Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings and regulator parts.

BOILER INSULATION

Insulation for cast-iron boilers is of two general types: (1) plastic material or blocks wired on, cemented and covered with canvas or duck; and (2) blocks, sheets or plastic material covered with a metal jacket furnished by the boiler manufacturer. Self-contained steel firebox boilers usually are insulated with blocks, cement and canvas, or rock wool blankets; HRT boilers are brick set and do not require insulation beyond that provided in the setting. It is essential that the insulation on a boiler and adjacent piping be of non-combustible material as even slow-burning insulation constitutes a dangerous fire hazard in case of low water in the boiler.

Chapter 12

RADIATORS AND GRAVITY CONVECTORS

Heat Emission of Radiators and Convectors, Types of Radiators, Output of Radiators, Heating Effect, Heating Up the Radiator and Convector, Enclosed Radiators, Convectors, Selection, Code Tests, Gravity-Indirect Heating Systems

THE accepted terms for heating units are: (1) *radiators*, for direct surface heating units, either exposed, enclosed, or shielded, which emit a large percentage of their heat by radiation; and (2) *convectors*, for heating units having a large percentage of extended fin surface and which emit heat principally by convection. Convectors are dependent upon enclosures to provide the circulation by gravity of large volumes of air.

HEAT EMISSION OF RADIATORS AND CONVECTORS

All heating units emit heat by *radiation* and *convection*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits less than half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible.

TYPES OF RADIATORS

Present day radiators may be classified as tubular, wall, or window types, and are generally made of cast-iron. Catalogs showing the many designs and patterns available now include a junior size sometimes known as slim tube radiation. The tubes in these radiators are materially smaller, and they are compactly assembled in less space than those of the standard radiator.

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but now wall type radiators are most frequently used for this service. When coils are used, the miter type assembly is to be

preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

OUTPUT OF RADIATORS

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not a true index of output. (The engineering unit of outputs is the *Mb* or 1000 Btu.) However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet*. For steam service this is based on an emission of 240 Btu per hour per square foot and for hot water service 150 Btu per hour per square foot.

TABLE 1. VARIATION IN DIMENSIONS AND CATALOG RATINGS OF 10-SECTION TUBULAR RADIATORS (STEAM)

No. of Tubes	3	4	5	6	7
Width of Radiator, Inches	4.6–5.1	6.0–7.0	8.0–8.9	9.1–10.4	11.4–12.8
Length per Section, Inches	2.5	2.5	2.5	2.5	2.5–3.0
HEIGHT WITH LEGS—INCHES	HEAT EMISSION—EQUIVALENT SQUARE FEET				
13–14	....	....	....	20	25.0–32.5
16–18	....	....	28.5	....	30.0–38.3
20–21	15.0–17.5	20.0–22.5	25.0–31.2	30	36.7–45.0
22–23	20.0–21.3	25	30.0–33.9	35	40.0–45.2
25–26	20.0–26.7	25.0–27.5	32.5–39.8	37.5–40.0	50.0–53.5
30–32	25.0–30.9	33.3–35.0	40.0–48.6	50	63.3–62.5
36–38	30.0–36.7	40.0–42.5	50.0–56.5	60	70.0–75.4

Table 1 illustrates the difficulty in tabulating tubular radiator outputs since there is so much variation in design between the products of the different manufacturers. Only on the four-tube and six-tube sizes is there any practical agreement in output value. The heat emission values appear as square feet but are entirely empirical, being based on the heat emission of the radiator and not on the measured surface.

An average value of 300 Btu per actual square foot of surface area per hour has been found for wall radiators one section high placed with their bars vertical. Several recent tests[1] show that this value will be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually so noticeable a difference in temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of a room satisfactorily.

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2. This has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu,

[1]University of Illinois, *Engineering Experiment Station Bulletin* No. 223, p. 30.

TABLE 2. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

SIZE OF PIPE	1 IN.	1¼ IN.	1½ IN.
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

Effect of Paint

The prime coat of paint on a radiator has no material effect on the heat output, but the finishing coat may influence the radiation emission and thus affect the heat output. Within the range of temperatures at which radiators operate, color has no appreciable influence on the radiation emitted. Thus, finishing coats of oil paints of various colors will give the same results. However, a bronze paint, applied as the finish coat will change the character of the surface and reduce the amount of heat emitted by radiation. No paint has a noticeable effect on the portion of heat which is given off by convection. The larger the proportion of direct radiating surface, the greater will be the effect of any finish coat of paint which changes the character of the surface. Available tests are on old-style column type radiators which give results as shown in Table 3.

Effect of Superheated Steam

Available research data indicate that there is probably a decrease in heat transfer rate for a radiator or gravity convector with superheated steam in comparison with saturated steam at the same temperature. The decrease is probably small for low temperatures of superheats and additional tests are necessary with varying degrees of superheat to establish accurate comparisons for all types of radiators and convectors[2].

TABLE 3. EFFECT OF PAINTING 32-IN. THREE COLUMN, SIX-SECTION CAST-IRON RADIATOR[a]

RADIATOR No.	FINISH	AREA SQ FT	COEFFICIENT OF HEAT TRANS. BTU	RELATIVE HEATING VALUE PER CENT
1	Bare iron, foundry finish	27	1.77	100.5
2	One coat of aluminum bronze	27	1.60	90.8
3	Gray paint dipped	27	1.78	101.1
4	One coat dull black heat resistant paint	27	1.76	100.0

[a]Comparative Tests of Radiator Finishes, by W. H. Severns (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 41).

[2]Tests of Radiators with Superheated Steam, by R. C. Carpenter (A.S.H.V.E. TRANSACTIONS, Vol. 7, 1901, p. 206).

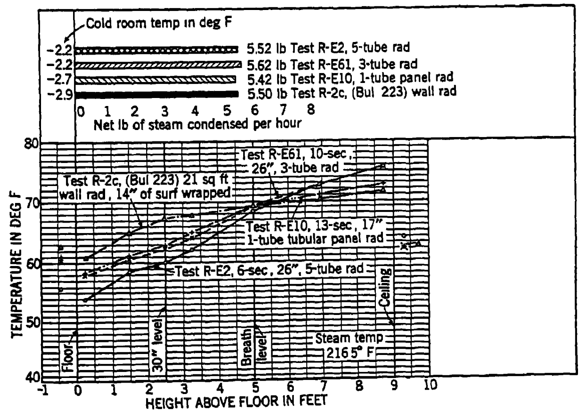


FIG. 1. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 60-IN. LEVEL

Note that the steam condensations are practically the same for all four radiators *when the same air temperature of 69 F is maintained at the 60-in. level.*

HEATING EFFECT

For several years the *heating effect* of radiators has been considered by engineers in order to use it for the rating of radiators and in the design of heating systems. Heating effect is the *useful output* of a radiator, in the comfort zone of a room, as related to the total input of the radiator[3].

The results of tests conducted at the University of Illinois are shown in Figs. 1 and 2[4]. For the four types of radiators shown, the following conclusions are given:

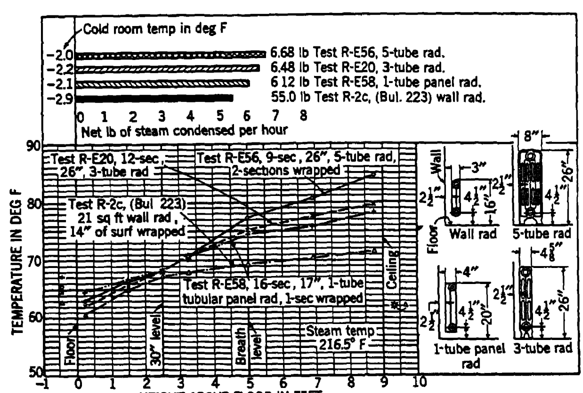


FIG. 2. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 30-IN. LEVEL

Note that the steam condensations are different for all four radiators *when the same air temperature of 68 F is maintained at the 30-in. level.*

[3]The Heating Effect of Radiators, by Dr. Charles Brabbée (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 33). A.S.H.V.E. RESEARCH REPORT No. 962—The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303).

[4]A.S.H.V.E. RESEARCH REPORT No. 905—Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 475).

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.
2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.
3. The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.
4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room. The comfort-indicating temperature should be taken below the breathing line level.
5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath window openings[5].

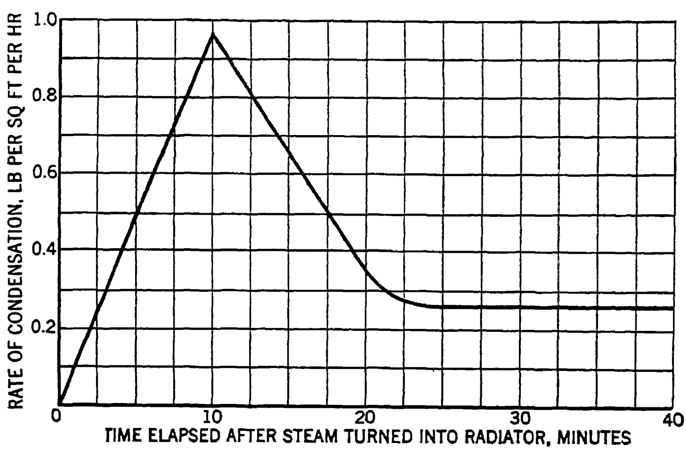


FIG. 3. CHART SHOWING THE STEAM DEMAND RATE FOR HEATING UP A CAST-IRON RADIATOR WITH FREE AIR VENTING AND AMPLE STEAM SUPPLY

HEATING UP THE RADIATOR AND CONVECTOR

The maximum condensation occurs in a heating unit when the steam is first turned on[6]. Fig. 3 shows a typical curve for the condensation rate in pounds per hour for the time elapsing after steam is turned into a cast-iron radiator. The data are from tests on old-style column type radiators. In practice the rate of steam supply to the heating unit while heating up is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard the supply of steam. Vacuum types of air venting valves may be used to reduce the length of the venting periods.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed,

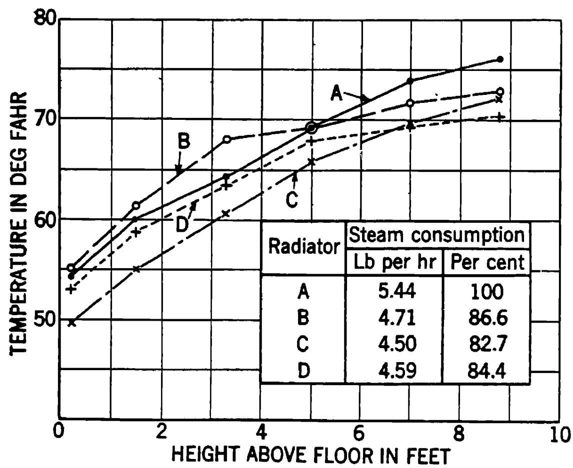
[5]Effect of Two Types of Cast-Iron Steam Radiators in Room Heating, by A. C. Willard and M. K. Fahnestock (*Heating, Piping and Air Conditioning*, March, 1930, p. 185).

[6]A.S.H.V.E. RESEARCH REPORT No. 1067—The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 389).

improve the heating effect. Recent investigations[7] indicate that in the design of the enclosure three things should be considered:

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.
2. The lessened steam consumption may not materially change the radiator heating performance.
3. The enclosed radiator may inadequately heat the space.

A comparison between a bare or exposed radiator (*A*) and the same radiator with a well-designed enclosure (*B*), with a poorly-designed enclosure (*C*), and with a cloth cover (*D*) will illustrate the relative heating effects. In Fig. 4 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about



Radiator	Steam consumption	
	Lb per hr	Per cent
A	5.44	100
B	4.71	86.6
C	4.50	82.7
D	4.59	84.4

Fig. 4. Steam Consumption of Exposed and Concealed Radiators

the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly designed enclosures. Curve (*D*) shows that the effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

Practically all commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[8] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when the radiator is steam hot and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

[7]University of Illinois, *Engineering Experiment Station Bulletins* Nos. 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A.S.H.V.E. Transactions, Vol. 35, 1929, p. 77).

[8]University of Illinois, *Engineering Experiment Station Bulletin* No. 230, p. 20.

CONVECTORS OR CONCEALED HEATERS

Although any standard radiator may be concealed in a cabinet or other enclosure so that the greater percentage of heat is conveyed to the room by convection thereby resulting in a form of gravity convector, generally better results are obtained with specially designed units which permit a free circulation of a larger volume of air at moderate temperatures. Since air stratifies according to temperature, moderate delivery temperatures at the outlet of the enclosure reduce the temperature differential between the floor and ceiling and accordingly accomplish the desired heating effect in the living zone.

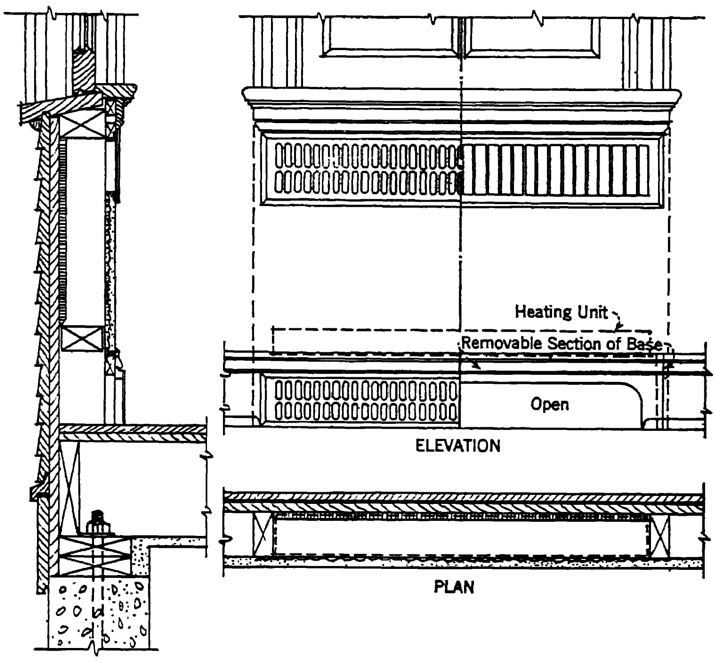


FIG. 5. TYPICAL CONCEALED CONVECTOR USING SPECIALLY DESIGNED HEATING UNIT

Fig. 5 shows a typical built-in convector. The heating element consisting of a large percentage of fin surface is usually shallow in depth and placed low in the enclosure in order to produce maximum chimney effect in the enclosure. The air enters the enclosure near the floor line just below the heating element, is moderately heated in passing through the core and delivered to the room through an opening near the top of enclosure. Since the air can only enter the enclosure at the floor line, the cooler air in the room which always lies at this level, is constantly being withdrawn and replaced by the warmer air. This air movement accomplishes the desired reduction in temperature differentials and assures maximum comfort in the living zone.

The *Convector Manufacturers Association* has adopted the A.S.H.V.E.

Standard[9] in the formulation of its ratings and has compiled a tentative standard of heating effect allowances for various enclosure heights to be included in the ratings by its members.

All published ratings bearing the title *C.M.C.* Ratings (*Convector Manufacturers Certified Ratings*) indicate that the convectors have been tested in accordance with the A.S.H.V.E. Code by an impartial and disinterested laboratory and that the ratings have been approved by the Standardization Committee of the *Convector Manufacturers Association.*

Concealed heaters or convectors are generally sold as completely built-in units. The enclosing cabinet should be designed with suitable air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are catalogued for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall-hung type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille in a plane parallel with the floor, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is important that the enclosure or housing for the convector fit as snugly as possible so that the air to be heated must pass through the convector and cannot be by-passed in the enclosure.

The output of a convector, for any given length and depth, is a variable of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation. When more than one heating unit is used, one mounted above the other in the same cabinet, the output of the upper unit or units will be materially less than that of the bottom unit.

RADIATOR AND CONVECTOR SELECTION

The capacity of a radiator varies as the 1.3 power, and that of a convector[10] as the 1.5 power of the temperature difference between the heating medium and the surrounding air in the case of the radiators, and the entering air in the case of the convector. It is obvious that for conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, the heat emission will be other than 240 Btu per square foot of rating.

Table 4 shows factors by which radiation requirements, as determined by dividing heat load by 240, shall be multiplied to obtain proper radiator

[9]A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam), (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367); (Hot Water), (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 237).

[10]A.S.H.V.E. RESEARCH REPORT No. 998—Factors Affecting the Heat Output of Convectors, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 443).

TABLE 4. CORRECTION FACTORS FOR DIRECT CAST-IRON RADIATORS AND CONVECTOR HEATERS[a]

Steam Press. Approx.		Heating Medium Temp. F Steam or Water	Factors for Direct Cast-Iron Radiators							Factors for Convectors						
			Room Temperature F							Inlet Air Temperature F						
Gage Vacuum In. Hg.	Abs. Lb per Sq In.		80	75	70	65	60	55	50	80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
LbperSqIn.																
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0.83	0.79	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

[a]To determine the size of a radiator or a convector for a given space, divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table.

To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, divide the heating capacities at the basic conditions by the proper factor from the above table.

or convector sizes from published rating tables for room temperatures ranging between 50 and 80 F as well as for steam or water temperatures from 150 to 300 F. For other room and heating medium temperatures the factor is determined by the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5}$$

where

C_s = correction factor.
t_s = steam temperature, degrees Fahrenheit.
t_r = room temperature, degrees Fahrenheit.
t_i = average inlet air temperature, degrees Fahrenheit.

As previously indicated, the output of radiators and convectors is still designated by the terms of older practice, but this is gradually giving place to an engineering method of designating heat emission. The A.S.H.V.E. has adopted the following standards: Code for Testing Radiators (1927); Codes for Testing and Rating Concealed Gravity Type Radiation (Steam), 1931, and (Hot Water), 1933, (see also A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 38).

For steam services the actual condensation weight is taken without any allowance for heating effect; for hot water services the weight of circulated water is used without allowance for heating effect. In all cases the total heat transmission varies as the 1.3 power for radiators[11] and the 1.5 power for convectors[12] of the temperature difference between that inside the radiator and the air in the room, and is expressed in Btu or *Mb* per hour.

[11]Loc. Cit. Note 9.
[12]Loc. Cit. Notes 9 and 10.

Standard test conditions specify either a steam pressure of 1 lb gage 15.6 lb per square inch absolute (215 F) or an average hot water temperature of 170 F and a room temperature of 70 F (5 ft above floor) for radiators, or an inlet air temperature of 65 F for convectors. The heating capacity of a *steam radiator* or *steam convector* is determined as follows:

$$H_t = W_s h_{fg} \tag{1}$$

where

H_t = Btu per hour under test conditions.
W_s = condensation in pounds per hour.
h_{fg} = latent heat in Btu per pound.

H_t may be converted to standard conditions of code ratings by using the proper correction factor from the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{T_s - T_r}\right)^{1.3} = \left(\frac{145}{T_s - T_r}\right)^{1.3} \tag{2}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{T_s - T_i}\right)^{1.5} = \left(\frac{150}{T_s - T_i}\right)^{1.5} \tag{3}$$

The output under standard conditions will be:

$$H_s = C_s H_t \tag{4}$$

where

C_s = correction factor.
T_s = steam temperature during test, degrees Fahrenheit.
T_r = room temperature during test, degrees Fahrenheit.
T_i = inlet air temperature during test, degrees Fahrenheit.
H_s = heat emission rating under standard conditions, Btu per hour.

Similarly, for *hot water convectors*, the output under test conditions may be determined as follows:

$$H = W(\theta_1 - \theta_2)\frac{3600}{t} \tag{5}$$

where

H = Btu per hour under test conditions.
W = pounds of water handled during test.
θ_1 = average temperature of inlet water, degrees Fahrenheit.
θ_2 = average temperature of outlet water, degrees Fahrenheit.
t = duration of test, seconds.

To convert test results to standard conditions, the following correction factor is used:

$$C = \left(\frac{170 - 65}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.5} = \left(\frac{105}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.5} \tag{6}$$

It has been shown that when the exponent 1.5 is used the range of error is less than 3 per cent[13] for convectors.

[13]Loc. Cit. Note 10.

Chapter 13

STEAM HEATING SYSTEMS

Gravity and Mechanical Return, Gravity One-Pipe Air-Vent, Gravity Two-Pipe Air-Vent, Air Line Heating, One-Pipe Vapor, Two-Pipe Vapor, Atmospheric, Condensation Return, Vacuum, Sub-Atmospheric, Orifice, Zone Control, Condensation Return Pumps, Vacuum Heating Pumps, Traps

STEAM heating systems may be classified according to the pipe arrangement, the accessories used, the method of returning the condensate to the boiler, the method of expelling air from the system, or the type of control employed. Information concerning the design and layout of steam heating systems will be found in Chapter 14.

GRAVITY AND MECHANICAL RETURN

Systems are classified as gravity or mechanical according to the method of returning the condensate from the system to the boiler. In *gravity systems* the condensate is returned by gravity due to the static head of water in the return pipes or mains. The elevation of the boiler water line must be sufficiently below the lowest heating unit, steam pipe or dry return pipe to permit the return by gravity. The *water line difference* forming the *static head* must be sufficient to overcome the maximum pressure drop in the system, including the pressure drop due to the condensing effect of the radiation. When radiator and drip traps are used, as in two-pipe vapor systems, the static pressure must also exceed the operating pressure of the boiler. The pressure drop caused by condensing rate of the radiation is especially important during those portions of the operating periods where changing pressure conditions prevail, as for example, when the system is being initially filled with steam. In systems where the condensate is wasted to the sewer, no water line difference is required as is the case with closed systems. However, the waste of condensate may introduce conditions which warrant the use of an appropriate mechanical system. Whenever the conditions of a heating system are such that the returns from the radiation cannot gravitate to the boiler, they must be returned by some mechanical means.

In *mechanical systems* the condensate flows to a receiver by gravity and is then forced into the boiler against its pressure. In all instances the preferable practice is to provide for gravity flow even where a vacuum pump is used. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system.

There are three general types of mechanical return devices in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return line pump.

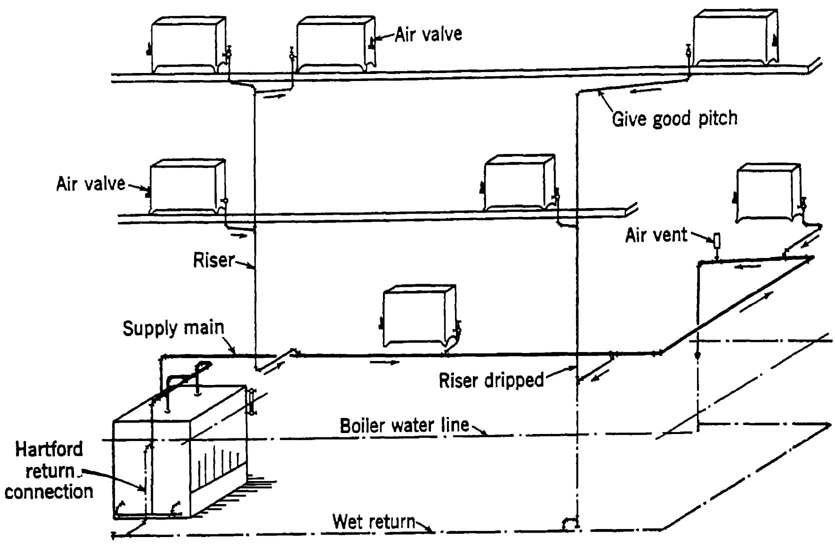


FIG. 1. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

GRAVITY ONE-PIPE AIR-VENT SYSTEM

This system is the most common of all methods of steam heating, especially for small size installations, due largely to its low cost and simplicity.

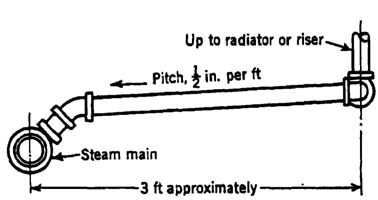


FIG. 2. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

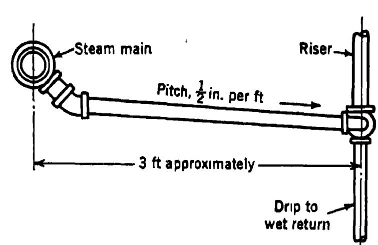


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

The downward pitch of a one-pipe air-vent system is indicated in Fig. 1. Low points and ends of steam mains pitched down from the boiler should be dripped. All drips should be sealed below water line before connecting together. In the risers and radiator connections, steam and

condensation flow in opposite directions. In long steam mains it flows in the same direction as the steam and is removed from the main through the drip. Short mains may be arranged for the condensate to flow in a direction opposite the steam by sizing them so the critical velocity is not exceeded. It is customary to drip the heel of each riser in buildings of several stories to avoid counter-flow of the steam and condensate in the riser branch. In buildings of one or two stories the condensate is returned to the steam main instead of being dripped. Both types of risers are shown in Fig. 1, and riser connections are shown in Figs. 2 and 3. A typical overhead down-feed system is illustrated in Fig. 4. While wet return mains need not be pitched toward the boiler to maintain steam circulation, they should be pitched for drainage.

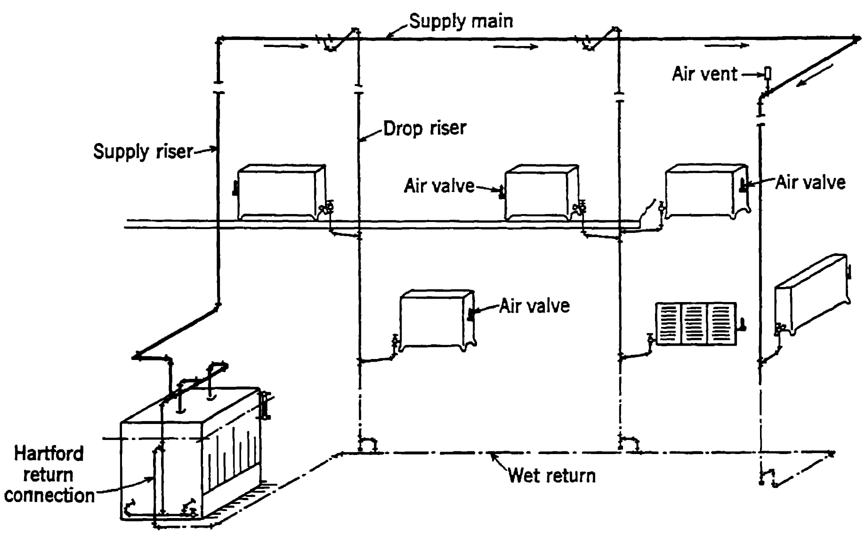


FIG. 4. TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

To improve steam circulation in one-pipe systems quick vent air valves should be provided at the ends and at intermediate points where the steam main is brought to a higher elevation. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 1, to prevent possible damage to their mechanisms by water.

The radiator valves may be the angle-globe, offset-corner pattern or gate type. Straight-globe and straight-corner type should not be used since the damming effect of the raised valve seat would interfere with the flow of condensation through the valve. Graduated valves cannot be used since the steam valves on this system must be fully open or fully closed to prevent the radiators filling with water and creating a dangerous water line condition. With a one-pipe system the heat cannot be modulated at the radiator, the steam being either *all on* or *all off*. Systems and devices are available which make it possible to obtain a partial modulating effect from one-pipe heating systems.

It is important to keep the lowest points of the steam mains and heating units sufficiently above the water line of the boiler to prevent flooding. The minimum water line difference depends on the initial steam pressure and piping pressure drop plus a safety factor for heating up.

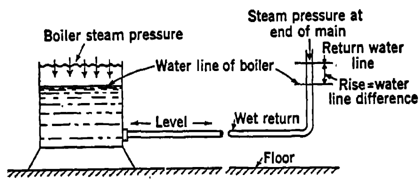


FIG. 5. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

Referring to Fig. 5 it will be noted that the water in the wet return is a U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction and resistance to the flow of steam in passing from the boiler to the far end of the main and the pressure reduction in consequence of the condensation occurring in the system. The water in the far end will

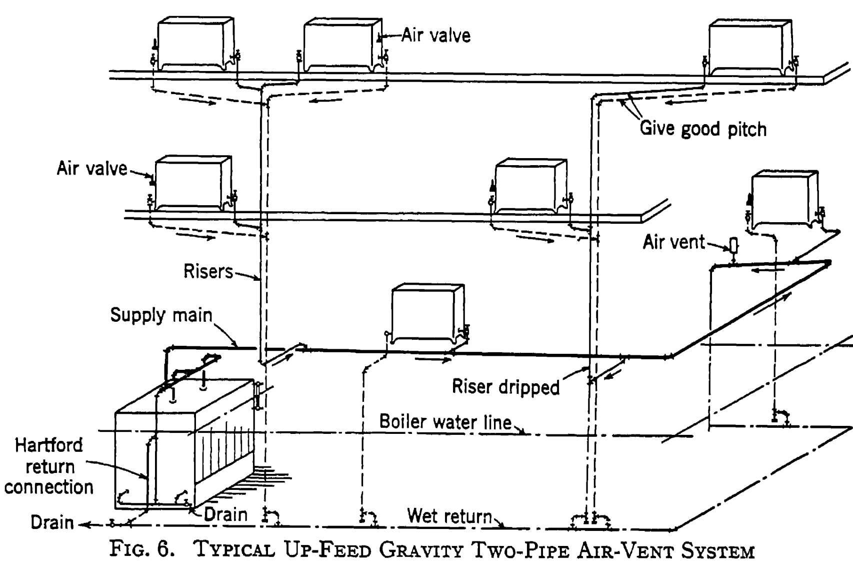


FIG. 6. TYPICAL UP-FEED GRAVITY TWO-PIPE AIR-VENT SYSTEM

rise sufficiently to overcome this difference in order to balance the pressures, and it will rise far enough to produce a flow through the return pipe and overcome the resistance of check valves if installed.

If a one-pipe steam system is designed, for example, for a total pressure drop of ⅛ lb, and utilizes an Underwriters' Loop instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be ⅛ of 28 in. (28 in. head being equal to one pound per square inch), or 3½ in. Adding 3 in. to overcome the resistance of the return main and 6 in. as a factor of safety for heating up gives 12½ in. as the distance; the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of ½ lb, and with a check in the return,

would require ½ of 28 in., or 14 in. for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

GRAVITY TWO-PIPE AIR-VENT SYSTEMS

The gravity two-pipe system indicated in Fig. 6 is now considered obsolete although many of these systems are still in use in older buildings. The same general principles governing its piping design are used when connecting radiators as in other types of gravity systems where they must discharge their condensation to the wet return pipe. Separate supply and return mains and connections are required for each heating unit. Radiator valves are required in both the supply and return connection to the radiator, and air valves are installed on the heating units and the mains. Where the return main has to be located high to function as a

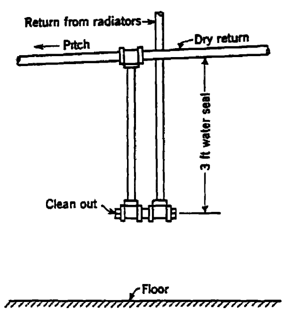


FIG. 7. METHOD OF CONNECTING TWO-PIPE GRAVITY RETURNS TO DRY RETURN MAIN

dry return, it is advisable to connect the return risers to the dry return main through water seals, as shown in Fig. 7, to prevent steam from one riser entering another.

The steam main in the down-feed system is carried to the top of the building, and the piping of the steam side is arranged as in the down-feed one-pipe gravity system. On the return side of the system, the piping is arranged in exactly the same manner as the up-feed gravity two-pipe system.

AIR LINE HEATING SYSTEMS

Both one- and two-pipe systems are at times provided with air valves which, instead of venting to the atmosphere direct, vent to a return pipe system of small size, which in turn is vented to atmosphere or connected to a vacuum pump. These are known as *one-pipe* and *two-pipe* air line systems. Where the air line is exhausted by a vacuum pump they are termed *one-pipe* or *two-pipe vacuum air line systems*.

ONE-PIPE VAPOR SYSTEM

The one-pipe vapor system operates under pressures at or near atmospheric and returns its condensation to the boiler by gravity. In this

system the automatic air valves are of special design to permit the ready release of air and prevent its ready return after it is expelled. The steam radiator valves are a type which, when opened, give a free and unobstructed passageway for water. The piping is the same as for the one-pipe gravity system but sized so as to permit operation at a few ounces pressure.

TWO-PIPE VAPOR SYSTEM

A two-pipe up-feed vapor system using separate supply and return pipes is shown in Fig. 8. The radiators discharge their condensation through thermostatic traps to the dry return pipe. These systems operate at a few ounces pressure and above, but those with mechanical condensate return devices may operate at pressures upward of 10 lb. The simplest

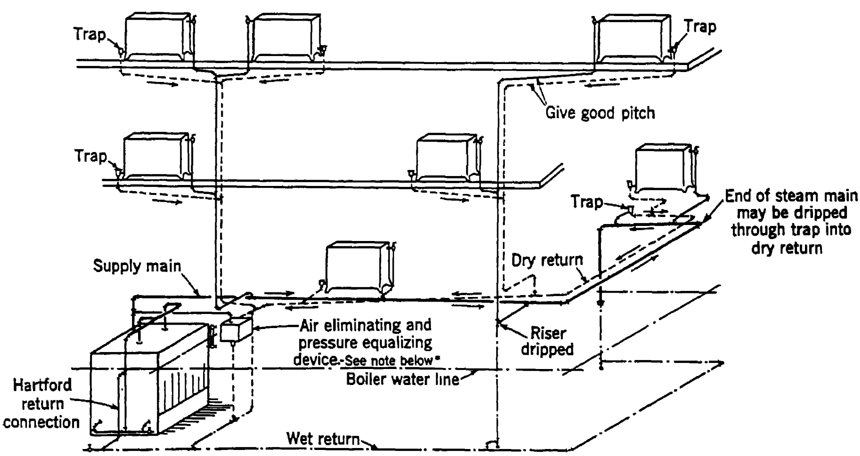


FIG. 8. TYPICAL UP-FEED SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

method of venting the system consists of a ¾-in. pipe with a check valve opening outward. Most systems employ various forms of vent valves, designed to allow the air to readily pass out of the system and to prevent its return. These systems permit control of the heat in the radiator by varying the opening of the graduated radiator valves. The boiler pressure is maintained at substantially constant pressure slightly above atmospheric pressure.

These systems may be classified as (1) *closed systems*, consisting of those which have a device to prevent the return of air after it has once been expelled from the system, and which can operate at both super and subatmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems*, comprising those which have the return line constantly open to the atmosphere without a check or other means to prevent the return of air. The open systems are not so popular because they have the disadvantage of not holding heat when the rate of steam generation is diminishing. Sys-

tems of this design should preferably be equipped with an automatic return trap to prevent water from backing out of the boiler. In installing the return trap a check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap, which is usually located with the bottom about 18 in. above the boiler water line. Some traps are constructed so that they will operate when they are installed with their bottom as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler. Fig. 9 shows a typical connection for an automatic return trap.

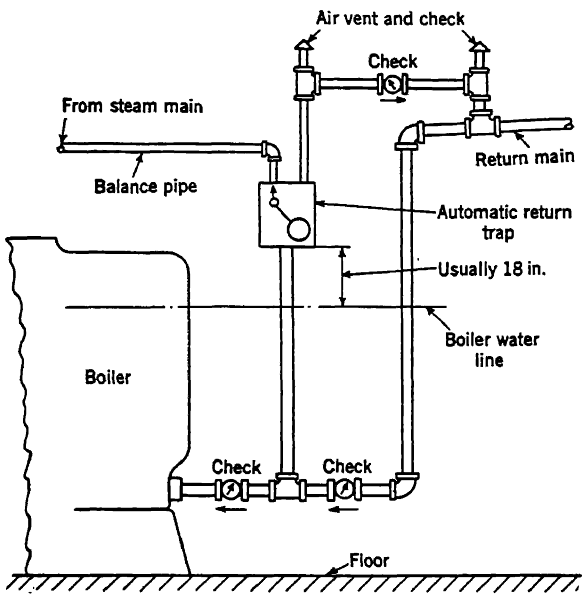


FIG. 9. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The branches are taken off the main from the bottom or at a 45-deg angle downward, with the runouts sloped toward the drops. Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop.

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small

and the normal size of drop required is 1 in. or less. The bottom of each steam drop should terminate with a dirt pocket and be dripped as shown in Fig. 10. The returns on a down-feed vapor system are the same as on an up-feed system. The runouts to the radiators and the radiator connections of the down-feed system are the same as those for the up-feed system already described.

CONDENSATION RETURN HEATING SYSTEMS

When automatic condensation return pumps are substituted for the gravity return of a two-pipe vapor system they are known as *return systems* or *return pump heating systems*. A typical installation of a motor driven automatic condensation unit is illustrated in Fig. 11. It will be noted that the returns are graded to cause flow by gravity to the vented receiver. As the receiver is filled, the float mechanism operates either a pilot or an across-the-line switch to start the pump and, upon emptying

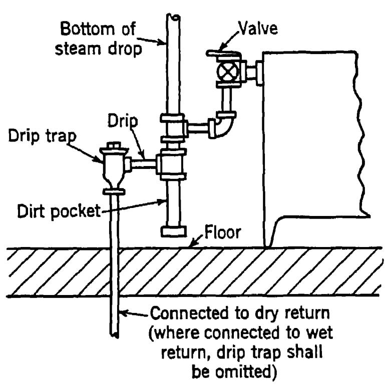


FIG. 10. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

the tank, to disconnect the power and stop it. The pump may be used to deliver the condensate direct to the boiler, to a feed water heater or to raise the water to any higher elevation or pressure than that of the return line. A useful application is a small condensation unit to handle a remote section of radiation that otherwise would be difficult to grade to the main return.

VACUUM SYSTEMS

In the vacuum system, a vacuum is maintained in the return line practically at all times. The pump is usually controlled by a vacuum regulator which operates the pump to maintain the vacuum within limits and operates in response to a pressure difference between the atmosphere and the return to control the vacuum in the return main. The source of steam supply may be a low pressure boiler as shown in Fig. 12, or a high pressure line through a pressure reducing valve. The piping and other details are the same as for the vapor systems.

The return risers are connected in the basement into a common return main which slopes downward toward the vacuum pump. The vacuum

pump withdraws the air and water from the system, separates the air from the water and expells it to atmosphere and pumps the water back to the boiler, or other receiver, which may be a feed-water heater or hot well. It is essential that no connection be made from the supply side to the return side at any point except through a trap. The desirable practice demands a return flowing to the vacuum pump by an uninterrupted downward slope. In some instances local conditions make it necessary to drop the return below the level of the vacuum pump inlet, before the pump can be reached. In such an event one of the advantages of the vacuum system is the ability to raise the condensate to a considerable height by the suction of the vacuum pump by means of a lift connection or fitting inserted in the return. The height the condensate can be raised depends on the steam pressure and the amount of vacuum maintained. It is

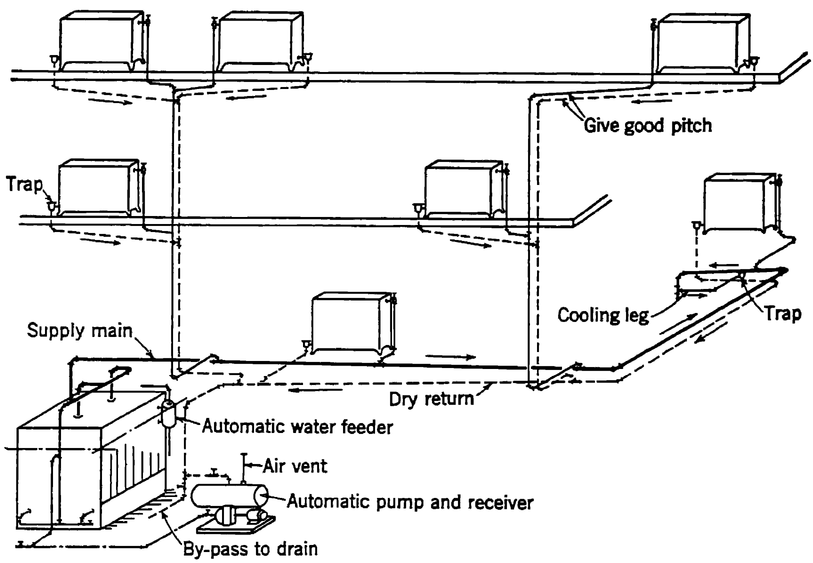


FIG. 11. TYPICAL INSTALLATION USING CONDENSATION PUMP

preferable to limit lift connections to a single lift at the vacuum pump. A still more preferable arrangement is the use of an accumulator tank, or receiver tank, with a float control for the pump at the low point of the return main located adjacent to the vacuum pump.

When the vertical lift is considerable, several lift fittings should be used in steps as shown in Fig. 13. This permits a given lift to be secured with a somewhat lower vacuum than where the vertical distance is served by a single lift. Where several lifts are present in a given system at different locations, the lifting cannot occur until the entire system is filled with steam. A lift connection for location close to the pump, where the size may be above the commercial stock sizes, is shown in Fig. 14. It is desirable that means be provided for manually draining the low point of the lift fittings to eliminate from the return piping all water in danger of freezing in case the system is shut down for a considerable length of time.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main so that they may serve as steam main drips. When this is done it is practical to run the steam main level if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 15). A slight pitch in the

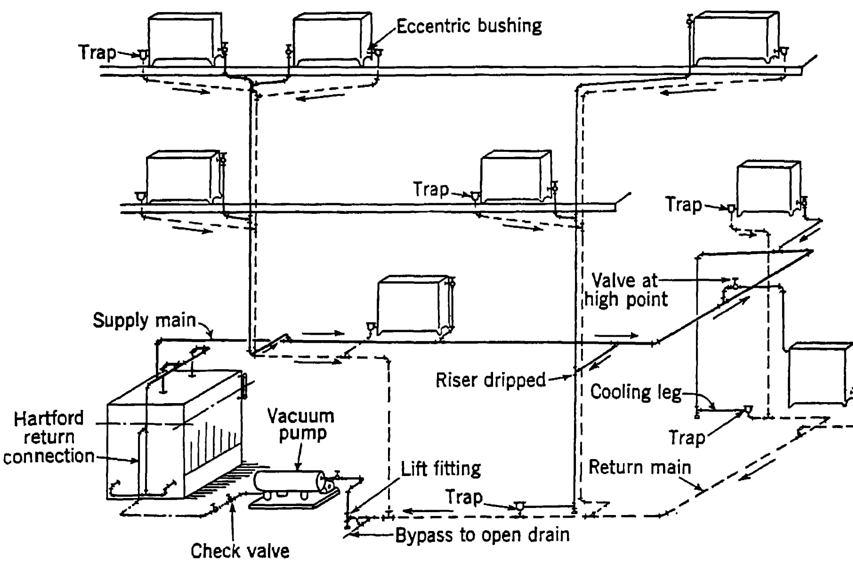


FIG. 12. TYPICAL UP-FEED VACUUM PUMP SYSTEM

steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 16.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature and volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on the return side. In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, atmospheric pressure or higher exists in the steam supply piping and

radiators only during severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25 in. Hg., after which further reduction in heat output is obtained by restricting the quantity of steam.

The rate of steam supply is controlled by a valve in the steam main or by thermostatically controlling the rate of steam production in the boiler. The control valve may be of the automatic *modulating* or *floating* type governed thermostatically from selected control points in the building, or it may be a special pressure reducing valve which will maintain the desired sub-atmospheric pressures by continuous flow into the heating main. All radiator supply valves have incorporated adjustable orifices or are equipped with regulating orifice plates. The sizes of orifices used are larger than for orifice systems because for equal radiator sizes the

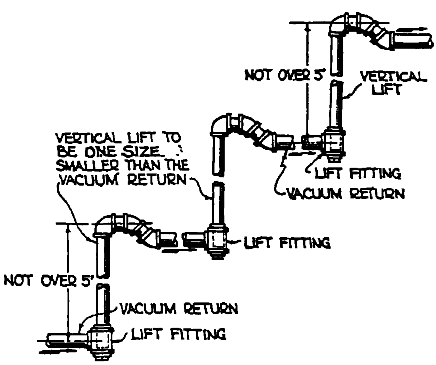


Fig. 13. Method of Making Lifts on Vacuum Systems when Distance is Over 5 ft

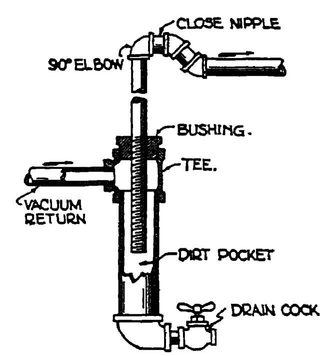


Fig. 14. Detail of Main Return Lift at Vacuum Pump

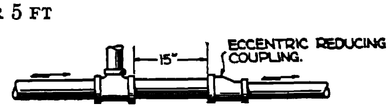


Fig. 15. Method of Changing Size of Steam Main when Runouts are Taken from Top

volume flowing is larger. These orifices are omitted on some systems, depending upon the type of control. Radiator traps and drips are designed to operate at any pressure from 15 lb gage to 26 in. of Hg. A vacuum pump capable of operating at high vacuum is preferable to promote accuracy in the distribution of steam throughout the system, particularly in mild weather. This vacuum is partially self induced by the condensation of the steam in the system under conditions of restricted supply for reduction of the radiator heat emission.

The returns must grade downward constantly and uninterruptedly from the radiator return outlets to the inlet of the receiver of the vacuum pump. One radical difference between this and the ordinary vacuum system is that no lifts should be made in the return line, except at the vacuum pump. The receivers are placed at a lower level than the pump and equipped with float control so the pump may operate as a return pump under night conditions. The system may be operated in the same manner as the ordinary vacuum system when desired.

Steam for heating domestic hot water should be taken from the boiler header back of the control valve so that pressures sufficiently high for heating the water may be maintained on the heater. The sub-atmospheric method of heating can be used for the heating coils of ventilating and air conditioning systems. The flexible control of heat output secured by this method materially reduces the required size of by-pass around the heaters. Sub-atmospheric systems are proprietary.

ORIFICE SYSTEMS

Orifice systems of steam heating may have piping arrangements identical with vacuum systems. Some of these omit the radiator thermostatic

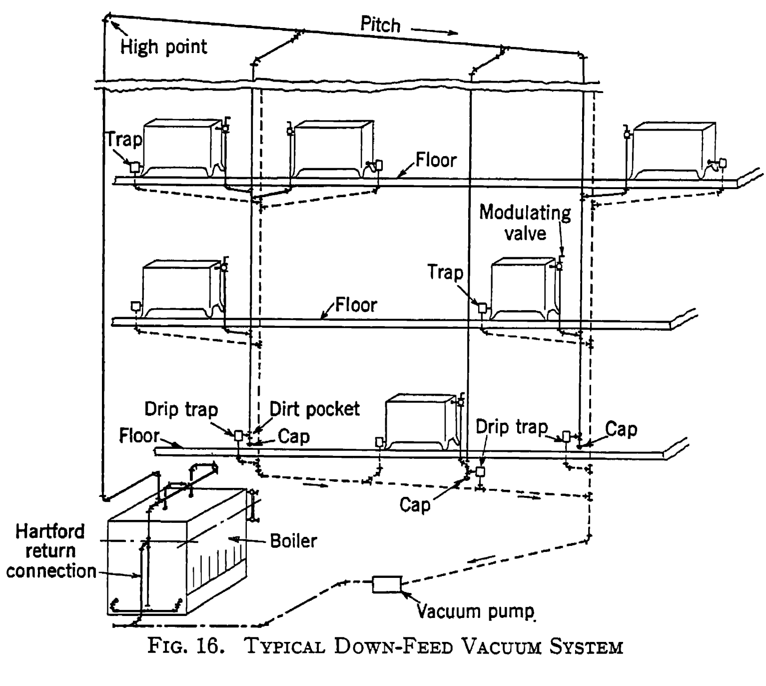


FIG. 16. TYPICAL DOWN-FEED VACUUM SYSTEM

traps but use thermostatic or combination float and thermostatic traps on all drip points. A return condensation pump with receiver vented to atmosphere, a return line vacuum pump, or a return trap, is generally used to return the condensation to the boiler or place of similar disposition, such as a feed-water heater or hot well. The heat emission from the radiators is controlled by varying the pressure maintained in the steam supply piping.

The principle on which these systems operate is based on the fact that the steam flow through an orifice will vary when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side, no further increase in flow will be obtained as a result of the increased pressure difference. If an orifice is so designed in

size as to exactly fill a radiator with 2 lb gage on one side and ¼ lb gage on the other, the absolute pressure relation is:

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ per cent.}$$

Should the steam pressure be dropped to ¼ lb on the supply pipe, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be apparent that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, reducing this steam main pressure will permit filling various desired portions of the radiator down to the point where the main pressure equals the back pressure in the radiator provided the supply pipe pressures may be controlled sufficiently close. If orifices are designed on a similar basis for a given system and proportioned to the heating capacity of the radiators they serve, all radiators will heat proportionately to the steam pressure. The range of pressure variation is limited by the permissible noise level of the steam flowing under the pressure difference required for maximum heat output. The control of the steam supply is obtained by a valve placed in the steam main, which maintains a determined pressure; or by a boiler pressure control. The valves are frequently manually set from a remote location, guided by temperature indicating stations in the building; or thermostatically controlled from a thermostat on the roof, which automatically measures the differential of outside and inside temperatures. Since the range through which the pressures may be varied is usually from 0 to 4.0 lb gage, the control should be capable of maintaining close regulation to maintain the desired space temperatures, particularly in mild weather.

Some systems use orifices not only in radiator inlets but also at different points in the steam supply piping for the purpose of balancing the system to a greater extent. In this manner the difference between the initial and terminal pressure in the steam main may be compensated to a great extent. For example, if the initial pressure was 3 lb gage and the pressure at the end of the main was 2 lb, an orifice could be used in each branch for the purpose of obtaining a more uniform pressure throughout the system. Such a provision may be particularly useful in this system for branches close to the boiler where the drop in the main has not yet been produced. Orifice systems are proprietary.

CONDENSATION RETURN PUMPS

Condensation return pumps are used for gravity systems when the local conditions do not permit the condensation to return to the boiler under the existing static head. The return of the condensate permits the water to repeatedly go through the cycle of vaporization, with subsequent condensation and return to the boiler. During such repeated cycles any incrustants or other substances in solution are precipitated and the water de-activated to a considerable extent so that corrosion of a serious nature is seldom ever encountered where the condensate is repeatedly used. Serious corrosion is more frequently found in systems where the condensation is not repeatedly used but is wasted and fresh make-up water is continually being introduced.

The most generally accepted condensation pump unit for low pressure

heating systems consists of a motor-driven centrifugal pump with receiver and automatic float control. Other types in use include rotary, screw and reciprocating pumps with steam turbine or motor drive, and direct-acting steam reciprocating pumps.

The receiver capacities of these automatic units should be sized so as not to cause too great a fluctuation of the boiler water line if fed directly to the boiler and at the same time not so small as to cause too frequent operation of the unit. The usual unit provides storage capacity between stops in the receiver of approximately 1.5 times the amount of condensate returned per minute and the pump generally has a delivery rate of 3 to 4 times the normal flow. This relation of receiver and pump size to heating system condensing capacity takes account of the peak condensation rate.

VACUUM HEATING PUMPS

On vacuum systems, where the returns are under a vacuum, and sub-atmospheric systems, where the supply piping, radiation and the returns are under a vacuum, it is necessary to use a vacuum pump to discharge the air and non-condensable gases to atmosphere and to dispose of the condensation. Direct-acting steam driven reciprocating vacuum pumps are sometimes used where high pressure steam is available or where the exhaust steam from the pump can be utilized. In general, however, these have been replaced by the automatic motor-driven return line heating pump especially developed for this service. Steam turbine drive is also frequently used where steam at suitable pressures is available, the steam being used afterward for building heating. The usual vacuum pump unit consists of a compact assembly of exhausting unit for withdrawing the air-vapor mixture and discharging the air to atmosphere and a water removal unit which discharges the condensate to the boiler. They are furnished complete with receiver, separating tank and automatic controls mounted as an integrated unit on one base. There are also special steam turbine driven units which are operated by passing the steam to be used in heating the building through the turbine with only a 2 to 3 lb drop across the turbine required for its operation. Under special conditions such as installations where it is necessary to return the condensate to a high pressure boiler, auxiliary water pumps may be supplied. In some instances separate air and water pumps may be used.

Practically all automatic motor-driven return line vacuum heating pumps make use of a portion of the condensate to operate either as a liquid piston pump or as a kinetic exhauster (which operate on a modified ejector principle) to withdraw the air and condensate from the system, discharge the air to atmosphere and return the condensate to the boiler. Some type of hydraulic action is utilized to produce the suction. Such hydraulic evacuating devices may be classified as:

a. Water ring centrifugal displacement pumps.
b. Water piston pumps.
c. Stationary kinetic exhauster pumps.
d. Rotary kinetic ejector pumps.

The evacuating element is generally combined with a centrifugal water impeller for the delivery of the condensate to the boiler or feed-water heater.

The assembled units may be further grouped under two general classifications:

a. Those which perform the function of air separation under atmospheric pressure.

b. Those which perform the function of air separation under a partial vacuum.

Pumps coming under the first classification remove both the air and condensate from the returns by means of the hydraulic evacuator and deliver both to a separating tank under atmospheric pressure. From this tank the air and non-condensable vapors are vented to atmosphere while the condensate is removed and delivered to the boiler by means of the built-in boiler feed pump impeller.

In the second classification, the air and condensate are first separated under vacuum by means of the receiver which is directly connected to the returns. The hydraulic evacuator withdraws only the air and non-condensable vapors from the top of the receiver and delivers them to atmosphere. The built-in condensate pump impeller removes the condensate from the bottom of the receiver and delivers it direct to the boiler or feed-water heater.

Under special conditions such as returning the condensate to a high pressure boiler or the furnishing of large air removal units for high vacuum systems, it is customary to supply separate motor-driven air and water pumps.

For rating purposes[1] vacuum pumps are classified as *low vacuum* and *high vacuum*. Low vacuum pumps are those rated for maintaining 5½ in. Hg. vacuum on the system, and high vacuum pumps are those rated to maintain vacuums above 5½ in.

The vacuum that can be maintained on a system depends upon the relationship of the air leakage rate into the system to the operating air capacity of the hydraulic evacuator when operating at any given return line temperature. The hotter the returns, the lower will be the possible vacuum for a given air leakage rate into the system. It is particularly essential on high vacuum installations to see that the entire system is tight in order to reduce the amount of inward air leakage and, furthermore, to see that relatively higher temperature steam is prevented from entering the vacuum return lines through leaky traps, high pressure drips, etc. It is for this reason that the condensate from equipment using steam at high pressures should not be connected directly to a vacuum return line but should drain to a receiver through a high pressure trap. The receiver should have an equalizing connection to a low pressure steam main and drain through a low pressure trap to the vacuum return main as indicated in Fig. 17.

Vacuum Pump Controls

In the ordinary vacuum system, the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired and cuts out when it has been increased to the highest point, these points being varied to suit the particular system or operating

[1]A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps, (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 33).

conditions. In addition to this vacuum control, a float control is included which will automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum on the system. A selector switch is usually provided to allow operation at night as a condensation pump only, also to give manual or continuous operation when desired.

There are several variations in the control of the vacuum maintained on the system by the pump. In some sub-atmospheric systems where orifices are used, the vacuum pump control maintains a pressure difference between the supply and the return piping, which is held within relatively close limits. There are other sub-atmospheric systems which utilize special temperature-pressure actuated controls for maintaining the desired conditions in the return lines. Where various zones are connected to the

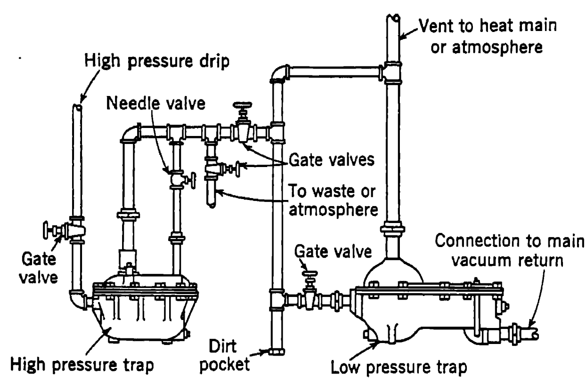


FIG. 17. METHOD OF DISCHARGING HIGH-PRESSURE APPARATUS INTO LOW-PRESSURE HEATING MAINS AND VACUUM RETURN MAINS THROUGH A LOW-PRESSURE TRAP

same return main, the return vacuum must be controlled to meet the requirements of the zone operating at the lowest steam supply pressure.

Piston Displacement Vacuum Pumps

Piston displacement return vacuum heating pumps may be either electric or steam driven. They should be provided with mechanical lubricators and their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. They are usually supplied with an air separating tank, open to atmosphere, placed on the discharge side of the pump and at an elevation sufficiently high to allow gravity flow of the condensate to the boiler. If the boiler pressure is too high for such gravity feed then an additional steam pump for feeding the boiler is desirable. The extra pump is sometimes avoided by using a closed separating tank with a float controlled vent. In both arrangements, the air taken from the system must be discharged against the full discharge pressure of the vacuum pump. In the case of high or medium pressure boilers, it is better to use the atmospheric separator and the second pump.

In figuring the required displacement for such pumps, a value of from 6 to 10 times the volumetric flow of condensation is used for average vacuums and systems.

TRAPS

Traps are generally classified as to function as (*a*) separating traps, (*b*) return, lifting or vacuum traps, and (*c*) air traps. Separating traps may be either float operated, thermostatically operated, or float and thermostatically operated. Return traps for low pressure service are referred to later as *alternating receivers* in this chapter. Return traps may also operate to receive condensate under a vacuum and return it to atmosphere or a higher pressure. Air traps are generally float operated.

Separating traps are used to release water of condensation but to retain steam. The thermostatic, and float and thermostatic types release both condensate and air but retain steam. Separating traps are used for draining condensate from radiators, indirect air heaters, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. Air traps release air but retain water. Devices known as air vents are, in principle, traps which allow the passage of air but prevent the passage of either water or steam.

Return traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap. If the static head is in favor of the trap discharge it is a minus quantity and may be deducted from the other factors of the discharge head.

Traps may also be classified according to the principle of operating device which supplies the power to cause them to function as (1) float, (2) bucket, (3) thermostatic, (4) float and thermostatic, (5) impulse, or (6) tilting traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the *upright bucket* trap, the water of condensation enters the trap and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the water rises above the edges of the bucket it flows into it and causes it to sink,

thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a discharge opening. When the bucket is emptied it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the valve. Water entering the trap fills the bucket, which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used, but this is unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators are called *blast* traps.

Float and thermostatic traps have both a thermostatic element to release air and a float element to release the water.

Impulse traps operate with a moving valve actuated by a control cylinder. When the trap is handling condensate, the pressure required to lift the valve is greater than the reduced pressure in the control cylinder and consequently the valve opens allowing a free discharge of condensate. As the remaining condensate approaches steam temperature, flashing results, flow through the valve orifice is choked and the pressure builds up in the control chamber closing the valve.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct-return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct-return trap assures safety for such systems and guarantees the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counter-balanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight overbalances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened thus admitting live steam pressure on the surface of the water and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying. The air relief of tilting traps is taken care of by the valves of the trap.

Chapter 14

PIPING FOR STEAM HEATING SYSTEMS

Operating Characteristics, Steam Flow, Pipe Sizes, Tables for Pipe Sizing, One-Pipe Gravity Air-Vent Systems, Two-Pipe Gravity Air-Vent Systems, Two-Pipe Vapor Systems, Vacuum, Orifice, Atmospheric and Sub-Atmospheric Systems, Boiler and Radiator Connections, Piping for Indirect Heating Units, Dripping

IT is important that steam piping systems distribute steam not only at full *design* load but during excess and partial loads. Usually the average winter steam demand is less than half of the demand at the *design* outside temperature. Moreover, in rapidly warming up a system even in moderate weather, the load on the steam main and returns may exceed the maximum operating load for severe weather due to the necessity of raising the temperature of the metal in the system to the steam temperature and the building to the *design* indoor temperature. Investigations of the return of condensation have revealed that as high as 143 per cent of the design condensation rate may exist under conditions of actual operation.

The functions of the piping system are the distribution of the steam, the return of the condensate and in systems where no local air vents are provided, the removal of the air. The distribution of the steam should be rapid, uniform and without noise, and the release of air should be facilitated as much as possible, as an air bound system will not heat readily nor properly. In designing the piping arrangement it is desirable to maintain equivalent resistances in the supply and return piping to and from a radiator. Arranging the piping so the total distance from the boiler to the radiation is the same as the return piping distance from the heating unit back to the boiler tends to obtain such a result. The condensation which occurs in steam piping as well as in radiators must be drained to prevent impeding the ready flow of the steam and air. The effect of back pressure in the returns and excessive revaporization, such as occurs where condensation is released from pressures considerably higher than the vacuum or pressure in the return, must be avoided.

The piping design of a heating system is greatly influenced by its operating chracteristics. Heating systems do not operate under constant conditions as they are continually changing due to variation in load. As the system is being filled with steam the pressure existing in various

Table 1. Flow of Steam in Pipes

P = loss in pressure in pounds.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

$$W = 5220\sqrt{\frac{PdD^5}{\left(1+\frac{3.6}{D}\right)L}}$$

$$P = 0.0000000367\left(1+\frac{3.6}{D}\right)\frac{W^2L}{dD^5}$$

Pressure Loss in Ounces	Col. 1	Pipe Size		Internal Area of Pipe Sq Inches	Col. 2	Steam Press. by Gage	Col. 3	Length of Pipe in Feet	Col. 4
	$5220\sqrt{\frac{P}{100}}$	Nominal	Actual Internal Diameter		$\sqrt{\frac{D^5}{1+\frac{3.6}{D}}}$		$\sqrt{d}$		$\sqrt{\frac{100}{L}}$
0.25	65.28	1	1.049	0.864	0.536	−1.0[a]	0.187	20	2.240
0.50	92.28	1¼	1.380	1.496	1.178	−0.5[a]	0.190	40	1.580
1.00	130.5	1½	1.610	2.036	1.828	0.0	0.193	60	1.290
2	184.6	2	2.067	3.356	3.710	0.3	0.195	80	1.120
3	226.0	2½	2.469	4.788	6.109	1.3	0.201	100	1.000
4	261.0	3	3.068	7.393	11.183	2.3	0.207	120	0.912
5	291.8	3½	3.548	9.887	16.705	5.3	0.223	140	0.841
6	319.7	4	4.026	12.730	23.631	10.3	0.248	160	0.793
7	345.3	4½	4.506	15.947	32.134	15.3	0.270	180	0.741
8	369.1	5	5.047	20.006	43.719	20.3	0.290	200	0.710
10	412.7	6	6.065	28.886	71.762	30.3	0.326	250	0.632
12	452.0	7	7.023	38.743	106.278	40.3	0.358	300	0.578
14	488.3	8	7.981	50.027	149.382	50.3	0.388	350	0.538
16	522.0	9	8.941	62.786	201.833	60.3	0.415	400	0.500
20	583.6	10	10.020	78.854	272.592	75.3	0.452	450	0.477
24	639.3	12	12.000	113.098	437.503	100.3	0.507	500	0.447
28	690.5	14	13.250	137.880	566.693	125.3	0.557	600	0.407
32	738.2	16	15.250	182.655	816.872	150.3	0.603	700	0.378
40	825.4					175.3	0.645	800	0.354
48	904.1					200.3	0.685	900	0.333
80	1167.2							1000	0.316
160	1650.7							1200	0.289
320	2334.5							1500	0.258
480	2859.1							2000	0.224

Column 1 × 2 × 3 × 4 = lb of steam per hour that will flow through a straight pipe for a given condition.

Example 1: 1 oz drop — 2 in. pipe — 1.3 lb press. — 100 ft equivalent length:

130.5 × 3.710 × 0.201 × 1 = 97.2 lb per hour.
97.2 × 4[b] = 388.8 sq ft equivalent radiation.

Table 1 does not allow for entrained water in low-pressure steam, condensation in covered pipe and roughness in commercial pipe as found in practice.

[a]Pounds per square inch gage = 2.04 in. Vacuum, Mercury Column.
[b]The factor 4 is the approximate equivalent in square feet of steam radiation of 1 lb of steam per hour.

locations may be different than those which exist for appreciable periods at other locations and which under constant pressure may have conditions that are approximately the same. In designing piping it is of especial importance to arrange the system to preclude trouble caused by such pressure differences. The systems which readily release the air permit uniform pressures to be attained in much shorter time intervals than those which are sluggish. Results are given in Fig. 1 from investigations[1] to determine the rate of condensate and air return from a two-pipe gravity heating system. Variations in the steam pressure during the warming up period when the rate of air elimination and condensation is high are clearly indicated in these curves.

It is evident that the condensation flow during the initial warming up

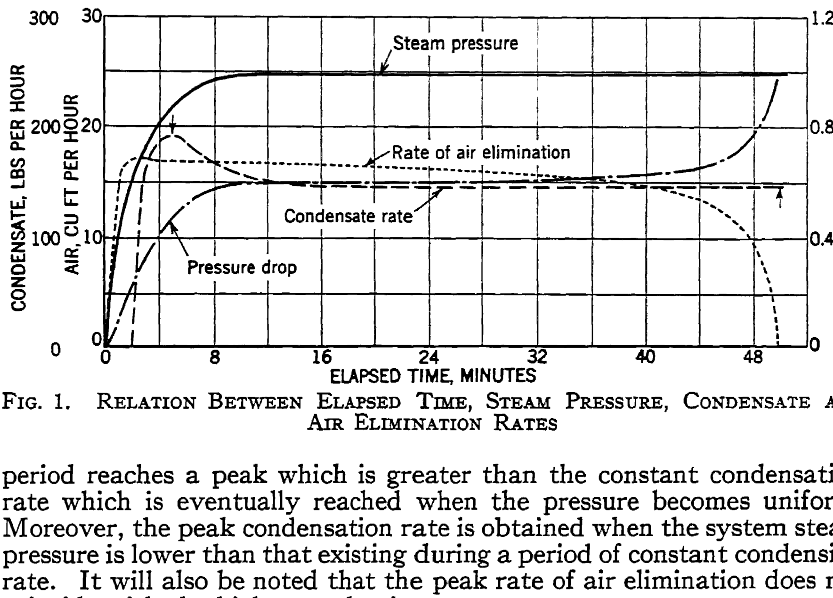


FIG. 1. RELATION BETWEEN ELAPSED TIME, STEAM PRESSURE, CONDENSATE AND AIR ELIMINATION RATES

period reaches a peak which is greater than the constant condensation rate which is eventually reached when the pressure becomes uniform. Moreover, the peak condensation rate is obtained when the system steam pressure is lower than that existing during a period of constant condensing rate. It will also be noted that the peak rate of air elimination does not coincide with the higher condensing rate.

STEAM FLOW

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship has been established by Babcock in the formula given at the top of Table 1. In Columns 1, 2, 3, and 4 of this table, the numerical values of the factors for different pressure losses, pipe diameters, steam densities and lengths of pipe have been worked out in convenient form so that the steam flowing in any pipe may be calculated by multiplying together the proper factors in each column as shown in the example at the bottom of the table.

[1]A.S.H.V.E. RESEARCH REPORT No. 954—Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 199).

PIPE SIZES

The determination of pipe sizes for a given load in steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.

2. The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow.

3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmospheric which normally operate under controlled partial vacua, the orifice, and the vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; and (4) the equivalent head due to pressure drop does not exceed the difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, or the dry return, and the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. A.S.H.V.E. Research Laboratory experiments limit this to the capacities given in Tables 2 and 3 for vertical risers and in Table 4 for horizontal pipes at varying grades.

Maximum Velocity

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced

TABLE 2. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR ONE-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
1	14.1	0.68	45	10.961	11.3
1¼	17.6	0.66	98	23.765	24.5
1½	20.0	0.66	152	36,860	38.0
2	23.0	0.57	288	69,840	72.0
2½	26.0	0.54	464	112,520	116.0
3	29.0	0.48	799	193,600	199.8
3½	31.0	0.44	1144	277,000	286.0
4	32.0	0.39	1520	368.000	380.0

INSTRUCTIONS FOR USING TABLE 2

1. Capacities given in Table 2 should never be exceeded on one-pipe risers.

2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.

3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 5 and 6).

TABLE 3. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR TWO-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
¾	20	......	40	9,550	10.0
1	23	1.78	74	17,900	18.45
1¼	27	1.57	151	36,500	37.65
1½	30	1.48	228	55,200	57.0
2	35	1.33	438	106,100	109.5
2½	38	1.16	678	164,100	169.4
3	41	0.95	1129	273,500	282.2
3½	42	0.81	1548	375,500	387.0
4	43	0.71	2042	495,000	510.5

INSTRUCTIONS FOR USING TABLE 3

1. The capacities given in this table should never be exceeded on two-pipe risers.

2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.

3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 5 and 6.)

sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, (3) the quantity of condensate flowing against the steam, and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided and which caused an actual difference of 20 per cent in the capacity of a 1-in. pipe in experiments carried on at the A.S.H.V.E. Research Laboratory (Table 5). The second is the reaming of the ends of the pipe after cutting, which, experiments indicate, might reduce the capacity of a 1-in. pipe as much as 28.7 per cent (Table 6). The third is the uniformity in grading the pipe line. All of the capacity tables given in this chapter include a factor of safety. However, the pipe

TABLE 4. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

Pitch of Pipe in Inches per 10 Ft

PITCH OF PIPE	¼ IN.		½ IN.		1 IN.		1½ IN.		2 IN.		3 IN.		4 IN.		5 IN.	
Pipe Size Inches	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

on which Table 5 is based showed no particular defects or constrictions on the inside, and the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 7 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it is necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLE 5. PER CENT DIFFERENCE IN CAPACITY FOR CARRYING STEAM AND CONDENSATE DUE TO VARIATION OF PIPE SIZE AND SMOOTHNESS[a]

	MAXIMUM CONDENSATION, LB PER HOUR			
Size of pipe	¾ In.	1 In.	1¼ In.	1½ In.
Minimum	14.00	24.89	45.42	70.50
Maximum	15.20	30.08	52.08	82.00
Per cent variation	8.6	20.8	14.7	16.3

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 6. EFFECT OF REAMING ENTRANCE TO ONE-INCH ONE-PIPE RISERS[a]

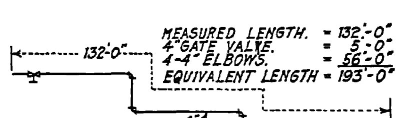
	MAXIMUM CAPACITY OF RISER	PER CENT DECREASE
Reamed entrances	24.7 lb per hour	0.0
Rounded entrances	23.9 lb per hour	3.2
Squared entrances	22.2 lb per hour	10.1
Three wheel cutter	19.2 lb per hour	22.2
Single wheel cutter	17.6 lb per hour	28.7

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 7. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

SIZE OF PIPE INCHES	LENGTH IN FEET TO BE ADDED TO RUN				
	Standard Elbow	Side Outlet Tee	Gate Valve	Globe Valve	Angle Valve
2	5	16	2	18	9
2½	7	20	3	25	12
3	10	26	3	33	16
3½	12	31	4	39	19
4	14	35	5	45	22
5	18	44	7	57	28
6	22	50	9	70	32
7	26	55	10	82	37
8	31	63	12	94	42
9	35	69	13	105	47
10	39	76	15	118	52
12	47	90	18	140	63
14	53	105	20	160	72

Example of length in feet of pipe to be added to actual length of run.



TABLES FOR PIPE SIZING[2]

Factors determining the size of a steam pipe and its allowable limit of capacity are the direction of the flow of condensate, whether against or with the steam.

Tables 8 and 9 are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation[3] (*abbreviated EDR*) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed without unusual or noticeable defects.

Table 8 may be used for sizing piping for steam heating systems by determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve, and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

Sizing of return piping may be done with the aid of Table 9 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 8. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-lb gage?

Solution. It will be assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 lb or less. With a pressure drop of 1 lb and a length of run of 1,000 ft, the drop per 100 ft would be 1/10 lb, while if the total drop were ½ lb, the drop per 100 ft would be 1/20 lb. In the first instance the pipe could be sized according to Column *D* for 1/16 lb per 100 ft, and in the second case, the pipe could be sized according to Column *C* for 1/24 lb. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

ONE-PIPE GRAVITY AIR-VENT SYSTEMS

One-pipe gravity air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized as follows:

[2]Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained. Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

[3]As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of THE GUIDE to gradually eliminate the empirical expression *square foot of equivalent direct radiation, EDR*, and to substitute a logical unit based on the Btu. The new terms to express the equivalent of 1000 Btu (Mb), and 1000 Btu per hour (Mbh), have been approved by the A.S.H.V.E.

Table 8. Steam Pipe Capacities

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

Pipe Size In.	Capacities of Steam Mains and Risers — Direction of Condensation Flow in Pipe Line — With the Steam in One-Pipe and Two-Pipe Systems: 1/32 lb or ½ Oz Drop	1/24 lb or ⅔ Oz Drop	1/16 lb or 1 Oz Drop	⅛ lb or 2 Oz Drop	¼ lb or 4 Oz Drop	½ lb or 8 Oz Drop	Against the Steam Two-Pipe Only: Vertical	Hori-zontal	Special Capacities for One-Pipe Systems Only: Supply Risers Up-Feed	Radiator Valves and Vertical Con-nections	Radiator and Riser Run-outs
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
¾	....	....	30	....	....	....	30	....	25	....	....
1	39	46	56	79	111	157	56	26	45	20	20
1¼	87	100	122	173	245	346	122	58	98	55	55
1½	134	155	190	269	380	538	190	95	152	81	81
2	273	315	386	546	771	1,091	386	195	288	165	165
2½	449	518	635	898	1,270	1,797	635	395	464	....	260
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	....	475
3½	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	....	745
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	....	1,110
5	3,214	3,712	4,546	6,429	9,092	12,858	....	3,150	....	....	2,180
6	5,276	6,094	7,462	10,553	14,924	21,105	....	....	....	....	....
8	10,983	12,682	15,533	21,967	31,066	43,934	....	....	....	....	....
10	20,043	23,144	28,345	40,085	56,689	80,171	....	....	....	....	....
12	32,168	37,145	45,492	64,336	90,985	128,672	....	....	....	....	....
16	60,506	69,671	84,849	121,012	169,698	242,024	....	....	....	....	....
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Un-dripped Run-outs	Up-Feed Risers	Radiator Con-nections	Run-outs Not Dripped

Note.—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed.

[a]Do not use Column *H* for drops of 1/24 or 1/32 lb; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 lb except on sizes 3 in. and over; below 3 in. substitute Column *B*.

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 8.

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use 1/16-lb drop (Column *D*).

2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.

3. *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.

4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.

5. *For the radiator valve size and the stub connection,* use Column *K*.

6. For the *dry return main,* use Column *U*.

7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over ¼ lb. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where 1/24-lb drop is being used, the steam main and dripped runouts would be sized from

TABLE 9. RETURN PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

CAPACITY OF RETURN MAINS AND RISERS

Pipe Size Inches	1/32 Lb or ½ Oz Drop per 100 Ft			1/24 Lb or ⅔ Oz Drop per 100 Ft			1/16 Lb or 1 Oz Drop per 100 Ft			⅛ Lb or 2 Oz Drop per 100 Ft			¼ Lb or 4 Oz Drop per 100 Ft			½ Lb or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
	Mains																	
¾	......	......	......	......	......	326	......	......	400	......	......	568	......	......	800	......	......	1,130
1	500	248	......	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	......	......	1,977
1¼	850	520	......	990	595	976	1,200	670	1,200	1,700	868	1,704	2,400	962	2,400	......	......	3,390
1½	1,350	822	......	1,570	943	1,547	1,900	1,058	1,900	2,700	1,362	2,696	3,800	1,512	3,800	......	......	5,370
2	2,800	1,880	......	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	......	......	11,300
2½	4,700	3,040	......	5,300	3,470	5,453	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	......	......	18,925
3	7,500	5,840	......	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,190	21,400	10,000	21,400	......	......	30,230
3½	11,000	7,880	......	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,710	32,000	14,300	32,000	......	......	45,200
4	15,500	11,700	......	18,300	13,400	17,910	22,000	15,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000	......	......	62,180
5	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
6	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100
	Risers																	
¾	......	190	......	......	190	570	......	190	700	......	190	994	......	190	1,400	......	......	1,977
1	......	450	......	......	450	976	......	450	1,200	......	450	1,704	......	450	2,400	......	......	3,390
1¼	......	990	......	......	990	1,547	......	990	1,900	......	990	2,696	......	990	3,800	......	......	5,370
1½	......	1,500	......	......	1,500	3,256	......	1,500	4,000	......	1,500	5,680	......	1,500	8,000	......	......	11,300
2	......	3,000	......	......	3,000	5,453	......	3,000	6,700	......	3,000	9,510	......	3,000	13,400	......	......	18,925
2½	......	......	......	......	......	8,710	......	......	10,700	......	......	15,910	......	......	21,400	......	......	30,230
3	......	......	......	......	......	13,020	......	......	16,000	......	......	22,710	......	......	32,000	......	......	45,200
3½	......	......	......	......	......	17,910	......	......	22,000	......	......	31,220	......	......	44,000	......	......	62,180
4	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
5	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100

Column *C;* radiator runouts and undripped riser runouts from Column *L;* up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of 1/24 lb); the dry return from Column *R;* and the wet return from Column *Q*.

With a 1/32-lb drop the sizing would be the same as for 1/24 lb except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

TABLE 10. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 2

PART OF SYSTEM	SECTION OF PIPE	RADIATION SUPPLIED (SQ FT)	THEORETICAL PIPE SIZE (INCHES)	PRACTICAL PIPE SIZE (INCHES)
Branches to radiators	....	100	2	2
Branches to radiators	....	50	1¼	1¼
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	2½	2½
Riser	*c* to *d*	400	2½	2½
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Branch to riser	*f* to *g*	600	3½	3½
Supply main	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	2½	3
Dry return main	*f* to *k*	600	1¼	2
Wet return main	*k* to *m*	600	1	2
Wet return main	*m* to *n*	600	1	2
Wet return main	*n* to *p*	600	1	2

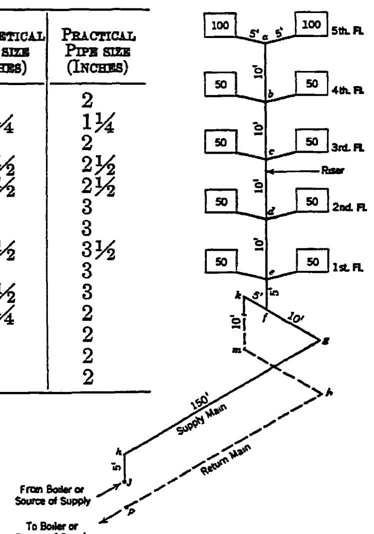


FIG. 2. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

Example 3. Size the one-pipe gravity steam system shown in Fig. 2 assuming that this is all there is to the system or that the riser and run shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of ¼ lb the drop per 100 ft will be slightly less than 1/16 lb. It would be well in this case to use 1/24 lb, and this would result in the theoretical sizes indicated in Table 10. These theoretical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should not be less than half its diameter at its largest part.

4. Supply mains, branches to risers, or risers, should be dripped where necessary.

5. Where supply mains are decreased in size they should be dripped, or be provided with eccentric couplings, flush on bottom.

TWO-PIPE GRAVITY AIR-VENT SYSTEMS

The method employed in determining pipe sizes for two-pipe gravity air-vent systems is similar to that described for one-pipe systems except that the steam mains never carry radiator condensation. The drop allowable per 100 ft of equivalent run is obtained by taking the equivalent length to the farthest radiator as double the actual distance, and then dividing the allowable or desired total drop by the number of hundreds of feet in the equivalent length. Thus in a system measuring 400 ft from the boiler to the farthest radiator, the approximate equivalent length of run would be 800 ft. With a total drop of ½ lb the drop per 100 ft would be $\frac{\frac{1}{2}}{8}$ or $\frac{1}{16}$ lb; therefore, Column *D* would be used for all steam mains where the condensation and steam flow in the same direction. If a total drop of ¼ lb is desired, the drop per 100 ft would be $\frac{1}{32}$ lb and Column *B* would be used. If the total drop were to be 1 lb, the drop per 100 ft would be ⅛ lb and Column *E* would be used.

For mains and riser runouts that are not dripped, and for radiator runouts where in all three cases the condensation and steam flow in opposite directions, Column *I* should be used, while for the steam risers Column *H* should be used unless the drop per 100 ft is $\frac{1}{24}$ lb or $\frac{1}{32}$ lb, when Columns *B* or *C* should be substituted so as not to exceed the drop permitted.

On an overhead down-feed system the main steam riser should be sized by reference to Column *H*, but the down-feed steam risers supplying the radiators should be sized by the appropriate Columns *B* through *G*, since the condensation flows downward with the steam through them. The riser runouts, if pitched down toward the riser as they should be, are sized the same as the steam mains, and the radiator runouts are made the same as in an up-feed system.

In either up-feed or down-feed systems the returns are sized in the same manner and on the same pressure drop basis as the steam main; the return mains are taken from Columns *O*, *R*, *U*, *X*, or *AA* according to the drop used for the steam main; and the risers are sized by reading the lower part of Table 9 under the column used for the mains. The horizontal runouts from the riser to the radiator are not usually increased on the return lines although there is nothing incorrect in this practice. The same notes apply that are given for one-pipe gravity systems.

TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low fire, and (3) because with large variation in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Column *U* (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed ⅛ lb to ¼ lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over ⅛ lb divided by 4, or 1/32 lb. In this case the steam mains would be sized from Column *B*; the radiator and undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of 1/32 lb. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over 1/32 lb. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.
3. In general it is not desirable to have a supply main smaller than 2 in.
4. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return, or may be connected into the dry return through a thermostatic drip trap.

VACUUM, ORIFICE, SUB-ATMOSPHERIC SYSTEMS

Vacuum, atmospheric, sub-atmospheric and orifice systems are usually employed in large installations and have total drops varying from ¼ to ½ lb. Systems where the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop while systems over 200 ft equivalent length of run more frequently go to the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of ½ lb divided by 12, or 1/24 lb. In this case the steam main would be sized from Column *C*, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of 1/24 lb). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed ⅛-lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.

2. Pitch of mains should not be less than ¼ in. in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

4. In general it is not considered desirable to have a supply main smaller than 2 in.

5. When necessary, the supply main, supply riser, or branch to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.

6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in Chapter 13.

7. No lifts can be used in orifice and atmospheric systems. In sub-atmospheric systems the lift must be at the vacuum pump.

BOILER CONNECTIONS

Steam

Cast-iron, sectional heating boilers usually have several outlets in the top. Two or more outlets are sometimes used to reduce the velocity of the steam in the vertical uptakes from the boiler and thus to prevent water being carried over into the steam main.

Return

Cast-iron boilers are generally provided with return tappings on both sides, while steel boilers are generally equipped with only one return tapping. Where two tappings are provided, both should be used to effect proper circulation through the boiler. The return connection should include either a Hartford loop or a check valve to prevent the accidental loss of boiler water to the returns with consequent danger of boiler damage. The Hartford loop connection is to be preferred over the check valve because the latter is apt to stick or not close tightly and, furthermore, because the check valve offers additional resistance to the condensate coming back to the boiler, which in gravity systems would raise the water line in the far end of the wet return several inches[4].

Hartford Return Connection

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford connection, or the Underwriters Loop, is recommended. This connection for a one- or two-boiler installation is shown in Fig. 3. The essential features of construction of a Hartford Loop connection are: (1) a direct connection (made without valves) between the steam side of the boiler and the return side of the boiler, and (2) a close nipple connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return pressure balance connection. Equalizing pipe connections between the steam and return are given in Fig. 3, based on grate areas, but in no case shall this pipe size be less than the main return piping from the system.

---

[4]See method of calculating height above water line for gravity one-pipe systems in Chapter 13.

Sizing Boiler Connections

Little information is available on the sizing of boiler runouts and steam headers. Although some engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any

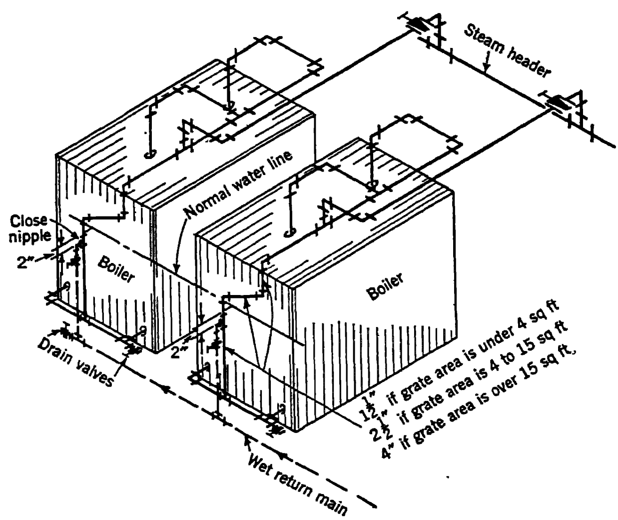


FIG. 3. THE HARTFORD RETURN CONNECTION

time, and sizing the runout on the same basis as the building mains. The difference in size between the vertical uptakes from the boiler and the horizontal main or runout is compensated for by the use of reducing ells.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop, the size being obtained from the Chart for Pressure Drop for Various Rates of Flow of Water, Chapter 45. The relative boiler loads should be considered, as in the case of gravity return connections. Boiler header and piping sizes should be based on the total load.

HIGH PRESSURE STEAM

When high pressure steam is being supplied and lower steam pressures are required for use in heating, domestic hot water, utility services, etc., one or more pressure reducing valves, or pressure regulators, as they are sometimes called, are required.

These are used in two classes of service, one where the steam must be shut off tight to prevent the low pressure building up at time of no load, and the other where the low pressure lines will condense enough steam to offset normal leaking through the valve. In the latter case, double seated valves may be used in a manner that reduces the work required of the diaphragm in closing the valve and consequently the size of the diaphragm. These valves also control the low pressures more closely under conditions of varying high pressures.

Valves that shut off all steam are called *dead end* type. They are single seated, and some of them have pilot operation that provides close control of the reduced pressure. If a thermostatically controlled valve is installed after, and near, a reducing valve in such a manner as to cut off the passage of steam, the dead end type should be used.

It is common practice when the initial steam pressure is 100 lb or higher to install two-stage reduction. This makes a quieter condition of steam flow, as it is apparent that with one reduction, as for example, from 150 to 2 lb, there is a smaller opening with greater velocity across the reducing valve, and consequently, more noise. A two-stage reduction also introduces a source of safety, since if one reducing valve were to build up its discharge pressure, this excess pressure would not be as great as the case might be in a one-stage reduction.

If an installation requires single seated valves, and the pilot type cannot be used, it is necessary to use two-stage reduction, as single seated valves require sufficient diaphragm area to overcome the unbalanced pressure underneath the single valve. In many cases the large diameter of diaphragm required would make it impractical in construction. With a two-stage reduction the diaphragm diameter required would be reduced. If a one-stage reduction is desired, it is necessary to use a pilot controlled pressure reducing valve, where low pressures are to be maintained closely.

In making two-stage reduction, allowance should be made, by increasing the pipe size, for expansion of steam on the low pressure side of the valve. This also allows steam flow to be at a more nearly uniform velocity. Separating the valves by a distance up to 20 ft is recommended to reduce excessive hunting action of the first valve.

When the reduced pressure is approximately 15 lb or lower, the weight and lever diaphragm valve gives the best results with minimum maintenance. Above 15 lb, spring loaded diaphragm valves should be used,

because of the extra weights required on weight and lever type. Equalizing line connections should be made not too close to the valve, and into the bottom of the reduced pressure steam main, to allow maximum condensation to exist in this equalizing line, or the connection is made into the top of the main and a water accumulator used to reduce the variation of the head of water on the diaphragm.

Care should be exercised in selecting the size of a reducing valve. The safest method is to consult the manufacturer. It is essential that sizes of piping to and from the reducing valve be such that they will pass the desired amount of steam with the maximum velocity desired. A common error is to make the size of the reducing valve the same size as that of the service, or outlet pipe size. Generally, this will make the reducing valve oversized, and bring about wire-drawing of valve and seat, due to small lift of the valve seat.

On installations where the steam requirements are relatively large and variable in mild weather or reduced demand periods, wire-drawing may occur. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30 per cent proportion of maximum flow. For example, if 50,000 lb of steam per hour are required, the size of one valve is on the basis of 0.7 + 50,000 lb, or 35,000 lb, and the other on the basis of 0.3 + 50,000 lb, or 15,000 lb. During the mild or reduced demand periods, steam will flow through the smaller valve only. During the remainder of the season, the larger valve is set to control at whatever low pressure is desired, and the smaller one at a somewhat lower pressure. Thus, when steam flow is not at its maximum, the smaller valve is shut, and automatically opens when the maximum steam demand occurs, since this maximum demand of steam creates a slight pressure drop in the service line.

The installation of reducing valves in pipe lines requires detailed planning. They should be installed to give ease of access for inspection and repair, and wherever possible with diaphragm downward, except in cases of pilot operated valves.

There should be a by-pass around each reducing valve of size equal to one half the size of reducing valve. The glove valve in by-pass line should be of a better type of construction, and must shut off absolutely tight. A steam pressure gage, graduated up to the initial pressure should be installed on the low pressure side. Safety valves located on the low pressure side should be set 5 lb higher than the final pressure but may be 10 lb higher than the reduced pressure if this reduced pressure is that of the first stage reduction of a double reduction. Strainers should always be installed on the inlet to the reducing valve but are not required before a second-stage reduction. If a two-stage reduction is made, it is well to install a pressure gage immediately before the reducing valve of the second-stage reduction also. In sizes 3 in. and above, it is advisable to tap the bodies of the reducing valve on inlet side for purposes of draining condensate accumulation through steam traps.

Control Valves

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never

be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

Connection to Heating Units

Riser, radiator and convector connections must not only be properly pitched at the time they are installed but must be arranged so that the pitch will be maintained under the strains of expansion and contraction. These connections may be made by swing joints which permit the expansion or contraction to occur under heating and cooling without bending of pipes. To take care of expansion in long risers, either expansion joints

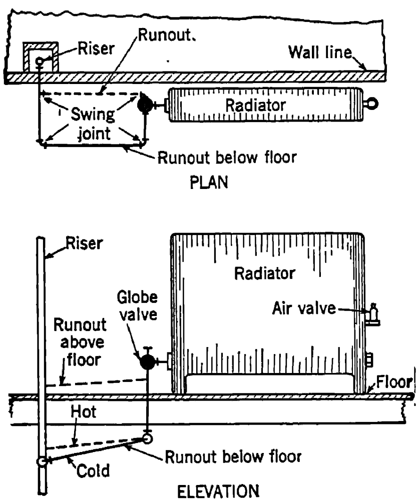


FIG. 4. ONE-PIPE RADIATOR CONNECTIONS

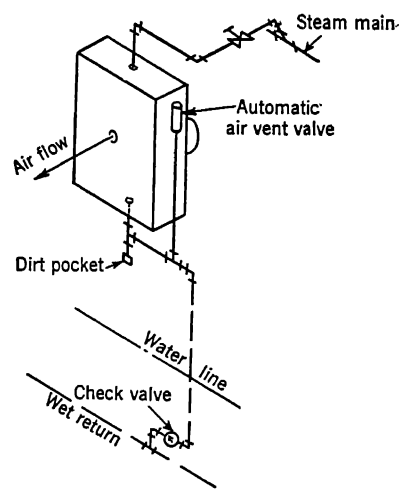


FIG. 5. UNIT HEATER CONNECTED TO ONE-PIPE AIR-VENT SYSTEM

of commercial construction or pipe swing joints are used. Anchoring of pipes between expansion joints is desirable.

Two satisfactory methods of making runouts for one-pipe systems for either the up-feed or the down-feed type are shown in Fig. 4. Where the vertical distance is limited and the runouts must run above the floor the radiator may be set on pedestals or of the high leg type. A method of connecting a unit heater to a one-pipe steam heating system is illustrated in Fig. 5.

Typical two-pipe radiator or convector connections are shown in Figs. 6, 7 and 8. While the top is the preferred location for the control valve, it may be located at the bottom. Short radiators may be top supply and bottom return on same end. With convectors the control valve is sometimes omitted and a damper in outlet grille used for heat control. The typical method of connecting pipe coils is shown in Fig. 9 and is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

Typical pipe connections for indirect radiators and tempering or heating stacks are shown in Figs. 10, 11, 12 and 13.

Where a building is served by a vacuum system or a sub-atmospheric system the stacks should be piped in the usual manner and traps of large capacity, preferably of the combination float and thermostatic type, should be used. In the orifice and *closed* two-pipe systems, traps should be used on the returns so that a pressure above that of the atmosphere may be secured on the heaters.

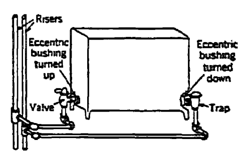


FIG. 6. TYPICAL CONNECTIONS FOR TWO-PIPE SYSTEM

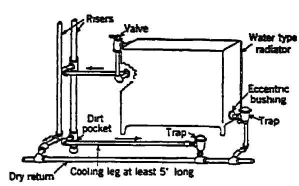


FIG. 7. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

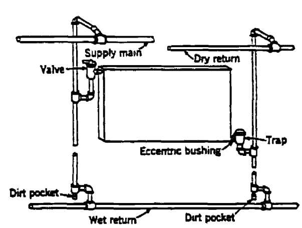


FIG. 8. CONNECTIONS TO RADIATOR HUNG ON WALL

Each stack should have a separate steam and return connection. Wide stacks are more evenly heated with two steam connections, one at each end, the stacks being divided and a return connection provided for each steam connection. For stacks of large capacity it is sometimes desirable to run a separate steam main direct from the boiler to the stacks.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_l - t_e)}{55.2 \times 240} = \frac{Q \times (t_l - t_e)}{220.8} \qquad (1)$$

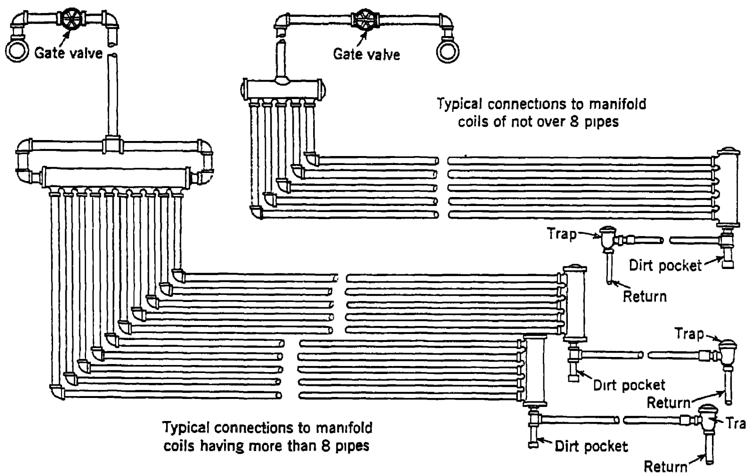


FIG. 9. TYPICAL PIPE COIL CONNECTIONS

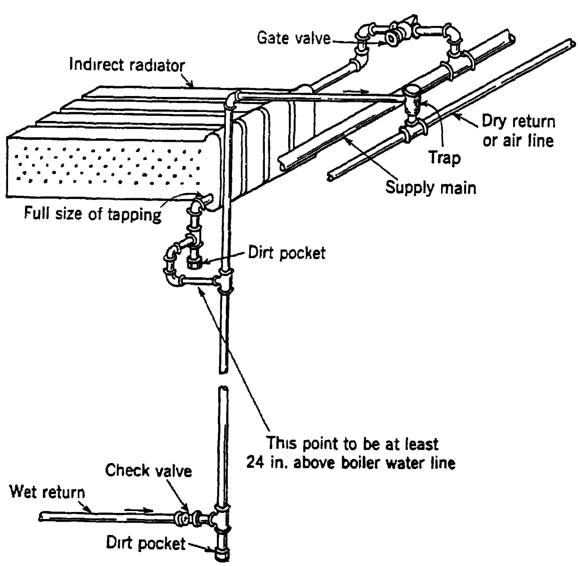


FIG. 10. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

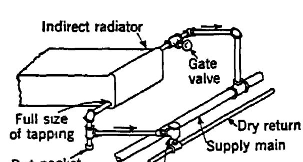


FIG. 11. PIPING CONNECTIONS TO INDIRECT RADIATORS

where

EDR = equivalent direct radiation, square feet.
Q = volume of air, cubic feet per minute.
t_e = the temperature of the air entering the row of heating units under consideration, degrees Fahrenheit.
t_l = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.
60 = the number of minutes in one hour.
55.2 = the number of cubic feet of air heated 1 F by 1 Btu.
240 = the number of Btu in 1 sq ft of EDR.

Example 4. Assume that the heating units shown in Fig. 14 are handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	Total Load (EDR)	Stack Load[a] (EDR)	Connection Load[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load.
[b]One half of stack load if two steam connections are made; otherwise, same as stack load.

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system.

DRIPS

A steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the direction of steam flow. Any steam main in any heating system can be elevated if dripped. Fig. 15 shows a connection where the steam main is raised and the drain is to a wet return. If the elevation of the low point is above a dry return it may be drained through a trap to the dry return in two-pipe vapor, vacuum and sub-atmospheric systems. Horizontal steam pipes may also be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for the condensation (Fig. 16).

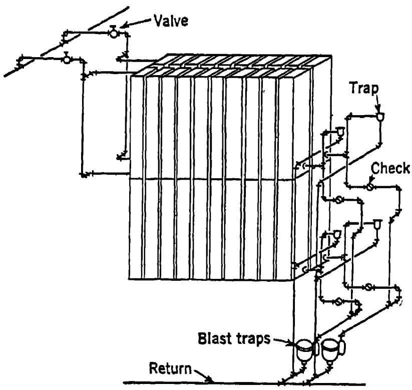


FIG. 12. SUPPLY AND RETURN CONNECTIONS FOR HEATING UNITS OF CENTRAL FAN SYSTEMS

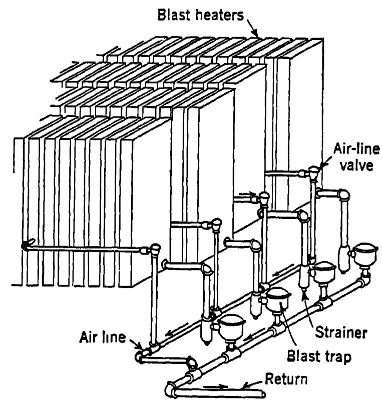


FIG. 13. TYPICAL CONNECTIONS TO CENTRAL FAN SYSTEM HEATING UNITS EXCEEDING 12 SECTIONS

Horizontal return pipes may be carried past doorways and other obstructions by using the scheme illustrated in Fig. 17. It will be noted that the large pipe, in this case, runs below the obstruction and the smaller one over it.

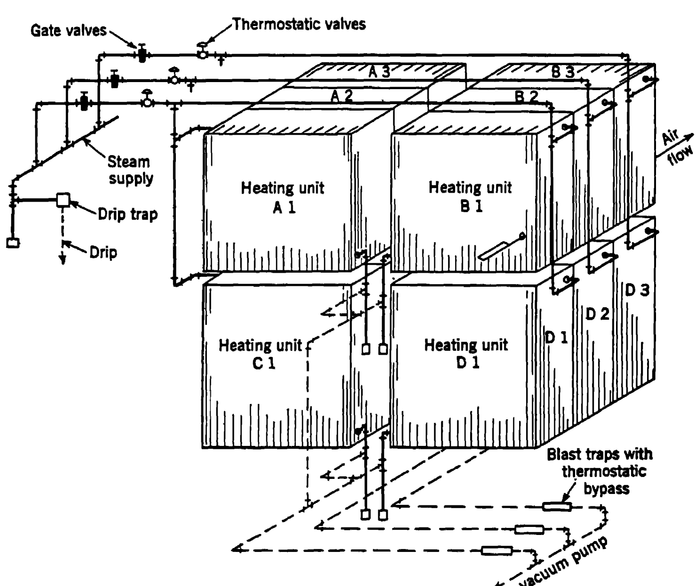


FIG. 14. TYPICAL PIPING FOR ATMOSPHERIC AND VACUUM SYSTEMS WITH THERMOSTATIC CONTROL (CENTRAL FAN SYSTEM)

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 18, but where radiator condensation does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing

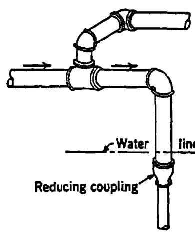


FIG. 15. DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

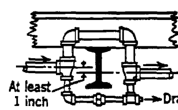


FIG. 16. LOOPING MAIN AROUND BEAM

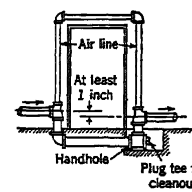


FIG. 17. LOOPING DRY RETURN MAIN AROUND OPENING

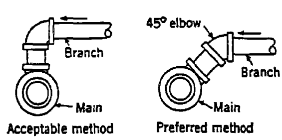
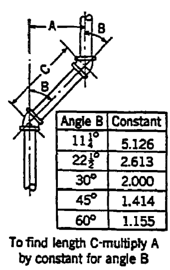


FIG. 18. METHODS OF TAKING BRANCH FROM MAIN



Angle B	Constant
$11\frac{1}{4}°$	5.126
$22\frac{1}{2}°$	2.613
30°	2.000
45°	1.414
60°	1.155

FIG. 19. CONSTANTS FOR DETERMINING LENGTH OFFSET PIPE

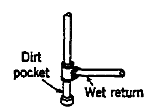


FIG. 20. DIRT POCKET CONNECTION

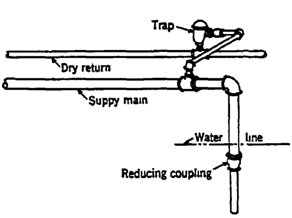


FIG. 21. DRIPPING END OF MAIN INTO WET RETURN

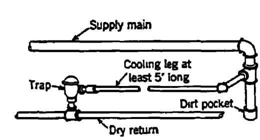


FIG. 22. DRIPPING END OF MAIN INTO DRY RETURN

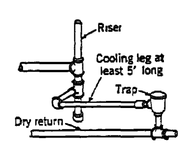


FIG. 23. DRIPPING HEEL OF RISER INTO DRY RETURN

without distorting the angle of the pipe. Runouts from the steam main are usually made about 5 ft long to provide flexibility for movement in the main.

Offsets in steam and return piping should preferably be made with 90-deg ells but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 19.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which

will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 20.

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry return line as shown in Fig. 21. On vacuum systems the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. The same method may be used in atmospheric systems. A float type trap is preferable to a thermostatic trap for dripping steam mains and large risers. If thermostatic traps are used, a cooling leg (Fig. 22) should always be provided. The cooling leg is for cooling the condensation sufficiently before it reaches the trap so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottom of the steam risers are dripped in the manner shown in Fig. 23. On large systems it is desirable to install a gate valve in the cooling leg ahead of the trap.

Chapter 15

HOT WATER HEATING SYSTEMS AND PIPING

One- and Two-Pipe Systems, Selecting Pipe Sizes, Forced Circulation, Gravity Circulation, Expansion Tanks, Installation Details

SYSTEMS for heating with hot water radiators may be divided into two general classes, the first known as gravity systems in which circulation is caused by the difference in density of the water in the flow and return lines, and the second known as forced circulation systems in which circulation is caused by a pump. Flow water temperatures vary from 150 to 220 F and the higher temperatures are generally used with the forced system.

For the sizing and selecting of boilers, radiators and piping, refer to Chapters 11, 12 and 17 respectively.

SYSTEMS OF PIPING

There are two general systems of piping used for either gravity or forced hot water systems:

(a) Two-pipe system.
(b) One-pipe system.

With either of these piping systems the distributing mains may be located in the cellar with up-feed to the radiators and risers or the supply main may be located in the attic with the return main located in the cellar. With the latter system of piping, the one-pipe system would be only one pipe for the risers. For basement radiators located on the floor the mains may be run at the ceiling, as one of the advantages of a forced hot water heating system is that the returns need not be below the radiators as required with a steam system. With some one-pipe systems there is one main in the cellar but separate flow and return risers and connections to the radiators.

In the two-pipe system there is separate supply and return pipes throughout so that the same water flows only through one radiator, resulting in the same water temperature in all radiators. With the one-pipe system part of the water flows through more than one radiator, so that the water temperature toward the end of the main is not as hot as near the boiler. However, with the one-pipe system, by maintaining a rapid circulation and small difference in temperature between the water leaving and returning to the boiler or other heat generator, the tendency to have variable temperatures in the radiators is much reduced.

The two-pipe system for larger buildings should, if possible, be arranged for reversed return. The direct and reversed return systems are shown

in Figs. 1 and 2. With the reversed return system, the water has to travel approximately the same distance from and back to the boiler for any one radiator as for any other radiator and, therefore, the friction and temperature losses to all radiators should be nearly the same.

In some cases the reversed return system involves no more piping than the direct return system. In the case of large buildings, it might be advisable to zone the piping.

Mechanical Circulators

Circulating pumps are usually of the centrifugal type. The capacity of the pump is figured from the Mbh (symbol representing 1000 Btu per hour) required for heating and the drop in temperature selected. For example, for 100 Mbh and 20 F drop a pump having a capacity of 5000 lb water per hour or 10 gpm. The resistance head is based on the system as designed. In large systems the economical size of pump may be determined by comparing the cost of power for operation, with the annual charges on the capital cost of the piping system, as larger pipe sizes mean less pump power. Velocities through piping in excess of 4 fps are likely to cause disturbing noises in buildings other than factories. In large

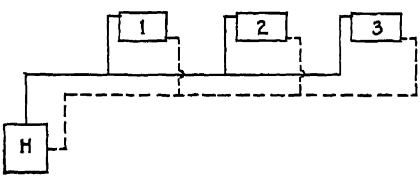


Fig. 1. A Direct Return System

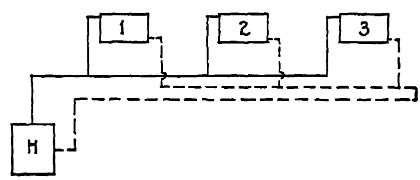


Fig. 2. A Reversed Return System

systems the pumps are run continuously while in small ones they are run either continuously or intermittently depending on the type of automatic temperature control selected. Small circulating pumps are usually driven by direct-connected electric motors. Under certain conditions a valved by-pass should be provided and the piping so designed that in case of breakdown of the pump or failure of electric current there will be sufficient gravity circulation to keep the building reasonably warm. In large buildings or groups of buildings, it is often advisable to have two pumps, each of about 70 per cent of the total capacity to take care of breakdown service. During mild weather, variations in water temperature may be utilized to balance the required heat loss. In the larger systems steam turbines are sometimes used to drive the pumps, the exhaust steam being used for heating the water, and in such buildings as hospitals this is usually the most economical method.

As the average pump used for water circulation is not over 60 per cent efficient, the cost of power on a large job should be figured and comparisons made between the savings made in capital cost of piping and the annual cost of power.

FORCED CIRCULATION PIPE SIZES

The pressure heads available in forced circulation systems are much greater than those in gravity circulation systems, consequently, higher

velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipe sizes of a heating system are reduced, the necessary increase in the velocity of the water increases the friction losses and thus the cost of operation and the initial cost of the circulating equipment. The increased velocity of a forced circulation system offers a number of advantages, such as a much shorter heating-up period and a more flexible control of hot water circulation. This improved performance merits the small increase in operating cost necessary to circulate the water mechanically. The velocities required should be determined by calculation for the particular system under consideration.

Since forced circulation velocities are higher than those in gravity systems, and since the friction heads in a heating system vary almost as the squares of the velocities, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in a gravity circulation system, and, consequently, it is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

In forced hot water systems, it is customary to use a temperature drop of 20 or 30 F between the water entering and leaving the boiler or other heater. The head against which the system is to operate must then be decided. This varies from 2 to 5 ft for small systems and may rise to 100 ft on large jobs with a group of buildings. For iron pipe, the sizes can be figured using Fig. 3 and Tables 1 and 3. For copper tubing Tables 2 and 3 are to be used. In systems designed with reversed returns, it will generally be found that very little adjustment is necessary to secure even distribution to all radiators. However, orifices may be used to control the flow and the capacities are given in Table 5. In large buildings provision should be made for quickly draining radiators in case of breakage, and it is often advisable to install a lock shield valve on one end of each radiator and a hand controlled valve on the other. In case of breakage the two valves can be closed and the radiator removed without affecting the rest of the system. The lock shield valve can also be used for balancing the water circulation.

The following examples will illustrate the procedure to be followed in designing forced circulation systems.

Example 1. From the plan of Fig. 4 note that the longest circuit consists of 151 ft of iron pipe; 1 boiler; 1 radiator; 1 radiator valve; 1 stop cock; 10 ells and 3 tees; and the shortest circuit consists of 127 ft of pipe; 4 tees; 1 boiler; 1 radiator; 1 radiator valve; 1 stop cock; and 6 ells. Design the piping for this system.

Solution. The friction in the various fittings can be expressed in terms of the friction in a 90-deg elbow from the values given in Table 3. The longest circuit consists of 151 ft of pipe and 44 elbow equivalents. The short circuit consists of 127 ft of pipe and 39 elbow equivalents.

The friction head in one elbow is approximately equal to the friction produced by the same sized pipe 25 diameters in length. Assume that the average pipe size for this system is 1 in. The equivalent length of the longest circuit will be 151 ft plus 100 ft or 251 ft of pipe. The equivalent length of the short circuit will be 217 ft.

Having determined the equivalent length of the circuits, the next step is to assume the rate at which the water is to be circulated in the system. The water may flow through the system so that it will cool any reasonable number of degrees. For the most economical average system a 20 F drop seems to be a satisfactory rate. This entails a slower water flow from the pumping equipment with a reasonable relationship between pipe

TABLE 1. EQUIVALENT LENGTH OF PIPE *vs.* PRESSURE HEAD AT VARIOUS FRICTION LOSSES STEEL PIPE

HEAD LOSS, FT	MILINCH FRICTION LOSS PER FOOT OF PIPE													
	720	480	360	300	240	180	160	144	120	96	90	80	70	60
	EQUIVALENT LENGTH OF PIPE IN FEET													
2	33	50	66	80	100	133	150	167	200	250	270	300	340	400
2½	42	62	84	100	125	167	188	208	250	312	333	375	428	500
3	50	75	100	120	150	200	225	250	300	375	400	450	510	600
3½	59	87	117	140	175	233	263	291	350	437	463	525	593	700
4	67	100	133	160	200	266	300	333	400	500	533	600	685	800
4½	75	112	149	180	225	300	338	374	450	562	593	675	758	900
5	83	125	167	200	250	333	375	416	500	625	666	750	860	1000
5½	92	137	183	220	275	366	413	457	550	687	713	825	923	1100
6	100	150	200	240	300	400	450	500	600	750	800	900	1030	1200
6½	108	162	217	260	325	433	488	540	650	812	843	975	1088	1300
7	116	175	233	280	350	465	525	580	700	875	933	1050	1200	1400
7½	124	187	249	300	375	500	563	623	750	937	973	1125	1252	1500
8	133	200	266	320	400	533	600	666	800	1000	1070	1200	1370	1600
8½	142	212	283	340	425	566	638	706	850	1062	1103	1275	1417	1700
9	150	225	300	360	450	600	675	750	900	1125	1200	1350	1540	1800
9½	159	237	317	380	475	633	713	789	950	1187	1233	1425	1577	1900
10	167	250	333	400	500	666	750	833	1000	1250	1333	1500	1715	2000
10½	175	262	349	420	525	700	788	872	1050	1312	1363	1575	1737	2100
11	183	275	366	440	550	733	825	916	1100	1375	1466	1650	1885	2200
11½	192	287	383	460	575	766	863	955	1150	1437	1533	1725	1897	2300
12	200	300	400	480	600	800	900	1000	1200	1500	1600	1800	2030	2400

NOMINAL PIPE SIZE, IN.		CAPACITY OF PIPES *Mbh* WITH A 20 F[a] DROP A = *Carrying Capacity*, B = *Velocity, inches per second* Friction Head of Pipe Milinches per foot													
		720	480	360	300	240	180	160	144	120	96	90	80	70	60
½	A	20	16	14	13	11	10	9	9	8	7	7	6	6	5
	B	*27*	*22*	*19*	*17*	*15*	*13*	*12*	*11*	*10*	*9*	*9*	*8*	*8*	*7*
¾	A	43	35	30	27	24	21	19	18	17	15	14	13	12	11
	B	*33*	*25*	*23*	*21*	*18*	*16*	*15*	*14*	*13*	*11*	*11*	*10*	*9*	*9*
1	A	85	70	60	54	48	41	39	36	33	30	28	27	25	23
	B	*39*	*32*	*27*	*25*	*22*	*19*	*18*	*17*	*15*	*13*	*13*	*12*	*11*	*10*
1¼	A	180	145	125	115	98	85	80	75	68	60	58	55	51	47
	B	*48*	*39*	*33*	*30*	*27*	*23*	*21*	*20*	*19*	*16*	*15*	*15*	*14*	*12*
1½	A	285	230	195	180	160	135	125	120	110	96	92	88	82	75
	B	*54*	*44*	*38*	*34*	*30*	*26*	*24*	*23*	*21*	*19*	*18*	*17*	*15*	*14*
2	A	540	435	370	340	300	255	240	230	205	180	175	165	150	140
	B	*64*	*52*	*45*	*40*	*36*	*30*	*29*	*27*	*24*	*22*	*21*	*20*	*19*	*17*
2½	A	890	720	610	550	480	420	390	370	330	300	280	270	250	230
	B	*74*	*60*	*50*	*46*	*41*	*35*	*33*	*31*	*28*	*24*	*24*	*22*	*21*	*19*
3	A	1650	1340	1130	1000	900	760	720	670	600	540	520	480	450	410
	B	*88*	*70*	*60*	*54*	*48*	*41*	*38*	*36*	*33*	*29*	*28*	*26*	*24*	*22*
3½	A	2500	2000	1700	1500	1350	1150	1080	1000	900	800	760	720	670	620
	B	*99*	*78*	*66*	*60*	*54*	*46*	*43*	*40*	*36*	*32*	*31*	*29*	*27*	*25*
4	A	3500	2800	2400	2200	1900	1600	1520	1440	1300	1150	1100	1050	960	880
	B	*110*	*87*	*74*	*66*	*58*	*50*	*47*	*45*	*40*	*35*	*34*	*32*	*30*	*27*
5	A	7000	5600	4700	4300	3700	3200	3000	2750	2500	2200	2100	2000	1800	1700
	B	*132*	*106*	*90*	*80*	*70*	*60*	*56*	*53*	*48*	*42*	*41*	*38*	*35*	*32*
6	A	12,000	9200	7800	7000	6200	5200	4800	4600	4100	3600	3500	3300	3000	2800
	B	*156*	*124*	*104*	*94*	*82*	*69*	*64*	*61*	*55*	*48*	*46*	*44*	*41*	*37*

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

TABLE 2. EQUIVALENT LENGTH OF TUBE *vs.* PRESSURE HEAD AT VARIOUS FRICTION LOSSES TYPE L COPPER TUBE

Head Loss, Ft	Milinch Friction Loss per Foot of Tube											
	720	600	480	360	300	240	180	150	120	90	75	60
	Equivalent Length of Tube in Feet (Longest Circuit)											
2	33	40	50	67	80	100	133	160	200	267	320	400
2½	42	50	63	83	100	125	167	200	250	333	400	500
3	50	60	75	100	120	150	200	240	300	400	480	600
3½	58	70	88	117	140	175	233	280	350	467	560	700
4	67	80	100	133	160	200	267	320	400	533	640	800
4½	75	90	113	150	180	225	300	360	450	600	720	900
5	83	100	125	167	200	250	333	400	500	667	800	1000
5½	92	110	138	183	220	275	367	440	550	733	880	1100
6	100	120	150	200	240	300	400	480	600	800	960	1200
6½	108	130	163	217	260	325	433	520	650	867	1040	1300
7	117	140	175	233	280	350	467	560	700	933	1120	1400
7½	125	150	188	250	300	375	500	600	750	1000	1200	1500
8	133	160	200	267	320	400	533	640	800	1067	1280	1600
8½	142	170	213	283	340	425	567	680	850	1133	1360	1700
9	150	180	225	300	360	450	600	720	900	1200	1440	1800
9½	159	190	238	317	380	475	633	760	950	1267	1520	1900
10	167	200	250	333	400	500	667	800	1000	1333	1600	2000
10½	175	210	263	350	420	525	700	840	1050	1400	1680	2100
11	183	220	275	367	440	550	733	880	1100	1467	1760	2200
11½	192	230	288	383	460	575	767	920	1150	1533	1840	2300
12	200	240	300	400	480	600	800	960	1200	1600	1920	2400

Nominal Tube Size, In.	Capacity of Tubes *Mbh* With a 20 F[a] Drop; A = *Carrying Capacity*, B = *Velocity, inches per second*; Friction Head of Pipe Milinches per Foot											
	720	600	480	360	300	240	180	150	120	90	75	60
⅜ A	10	9	8	6.8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
⅜ B	*27*	*24*	*21*	*18*	*16.5*	14	*13*	*11*	*10*	*8.5*	*8*	*7*
½ A	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
½ B	*33*	*30*	*25*	*21*	*19*	*17*	*15*	*13*	*12*	*10*	*9*	*8*
⅝ A	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	9	7.9
⅝ B	*37*	*34*	*30*	*24*	*21*	*19*	*17*	*15*	*13*	*11*	*10*	*9*
¾ A	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
¾ B	*42*	*38*	*33*	*27*	*24*	*21*	*19*	*17*	*14*	*12*	*11.5*	*10*
1 A	104	94	82	70	63	56	47	42	37	32	28	25
1 B	*48*	*45*	*39*	*34*	*30*	*25*	*22*	*19*	*17*	*14.5*	*13*	*12*
1¼ A	185	169	149	125	112	100	84	75	66	56	50	44
1¼ B	*55*	*51*	*45*	*39*	*35*	*30*	*25*	*22*	*19*	*17*	*15*	*13*
1½ A	300	270	235	200	180	160	134	120	105	90	81	71
1½ B	*62*	*57*	*51*	*43*	*39*	*35*	*30*	*25*	*22*	*19*	*17*	*15*
2 A	625	560	495	420	375	335	280	250	200	188	170	150
2 B	*76*	*68*	*59*	*51*	*47*	*42*	*36*	*32*	*27*	*22*	*20*	*18*
2½ A	1130	1010	890	750	680	600	500	450	395	335	305	270
2½ B	*90*	*80*	*69*	*58*	*49*	*47*	*42*	*37*	*33*	*26*	*23*	*21*
3 A	1840	1650	1450	1210	1100	980	820	740	650	550	490	420
3 B	*98*	*90*	*80*	*66*	*59*	*52*	*47*	*42*	*36*	*30*	*27*	*23*
3½ A	2750	2480	2170	1840	1650	1450	1210	1100	980	820	740	650
3½ B	*110*	*100*	*89*	*75*	*66*	*57*	*51*	*45*	*40*	*35*	*30*	*26*
4 A	3900	3505	3100	2600	2350	2090	1760	1580	1390	1180	1080	950
4 B	*120*	*108*	*96*	*83*	*73*	*63*	*55*	*49*	*44*	*37*	*34*	*29*

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

size and flow. Assume 20 F drop for this system. One gallon of water per minute with a density of 7.99 at 215 F will deliver approximately 9600 Btu per hour with a 20 F drop.

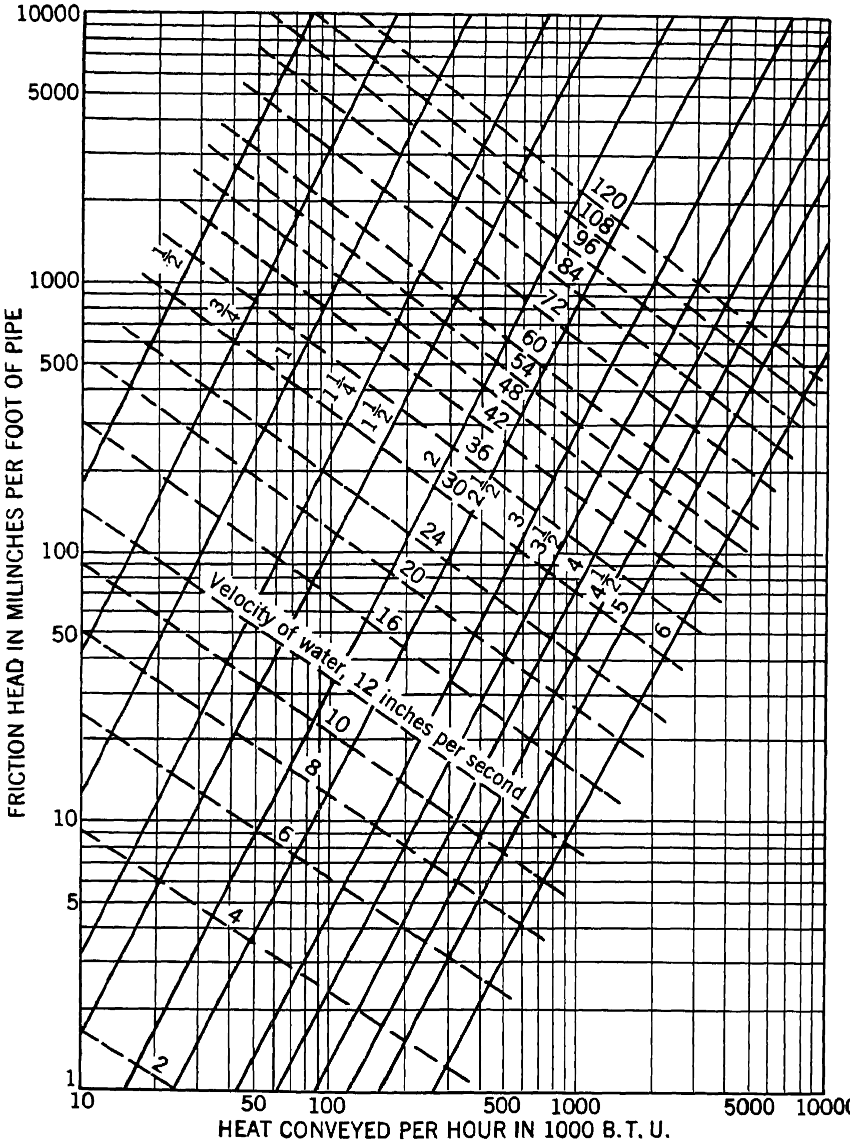


Fig. 3. Friction Heads in Black Iron Pipes for a 20 F Temperature Difference of the Water in the Flow and Return Lines

For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

The total radiation load is 98 Mbh, therefore the pump must deliver 10.2 gpm or 4900 lb of water per hour.

Knowing that the rate of flow is 10.2 gpm, the next step is to determine from the characteristics of available pumps, which one will produce a satisfactory velocity in the system. Assume that 4 pumps are available for this load which will produce 10.2 gpm at pressure heads of 2, 5, 10 and 18 ft. At these heads the pumps would produce a velocity high enough to make available a friction head per foot of pipe of 96, 240, 480 and 860 milinches per foot respectively. If 95 milinches per foot were used, the gravity head at 215 F average temperature in the mains would be 26 per cent of the total head and should be considered in sizing the system. At 240 milinches per foot the gravity effect is 10 per cent and as this is lower than the delivery variation from the pipe used, it can be neglected. At 480 and 860 milinches the gravity effect is still a smaller percentage of

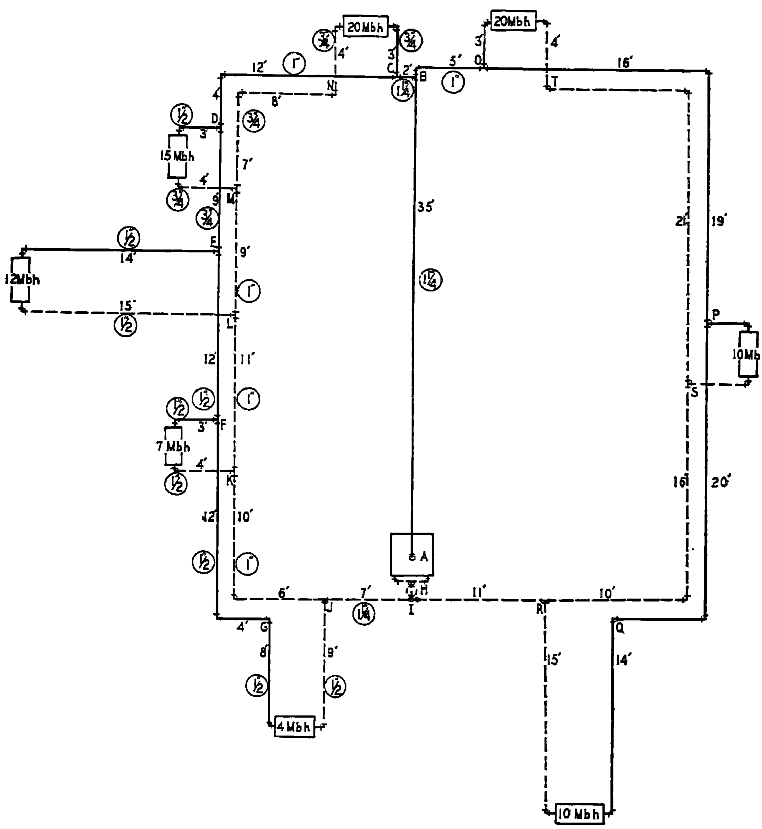


FIG. 4. A FORCED CIRCULATION REVERSED RETURN SYSTEM[a]

[a]Note that the numbers on the radiators indicate thousands of Btu per hour (Mbh) and not square feet.

the total, but at these losses in the average system the cost of pumping will more than offset the advantage gained in pipe sizes. Therefore, pipe size this system at 240 milinches per foot which is equivalent to a total loss of 60,000 milinches for the 250 ft equivalent length of pipe.

The pipe sizes may be selected from Fig. 3 or from Table 1 which has been derived from Fig. 3.

Size the supply main of the longest circuit first. Section AB carries 98 Mbh. From Fig. 3 it will be noted that at 240 milinches per foot, a 1¼-in. pipe carries 98 Mbh. Therefore, use 1¼-in. pipe in Section AB. Section BO carries 40 Mbh. A 1-in. pipe carries 48 Mbh at 240 milinches per foot. Use a 1-in. pipe. Section OP carries 20 Mbh

TABLE 3. IRON AND COPPER ELBOW EQUIVALENTS

Fitting	Iron Pipe	Copper Tubing
Elbow, 90-deg	1.0	1.0
Elbow, 45-deg	0.7	0.7
Elbow, 90-deg long turn	0.5	0.5
Open return band	1.0	1.0
Open gate valve	0.5	0.7
Open globe valve	12.0	17.0
Angle radiator valve	2.0	3.0
Radiator	3.0	4.0
Boiler or heater	3.0	4.0
Tee, per cent flowing through branch:		
100	1.8	1.2
50	4.0	4.0
25	16.0	20.0

and this will require ¾-in. pipe. Section PQ carries 10 Mbh and requires ½ in. pipe. To size the return start from the boiler and proceed backwards. Section IR carries 40 Mbh and from Fig. 3 a 1-in. pipe is required. Section RS carries 30 Mbh which is only slightly over the capacity of a ¾-in. pipe, so use ¾ in. Section ST carries 20 Mbh and requires a ¾-in. pipe. The radiator branches are determined in the same manner. It is evident from the chart that it is impossible to maintain a constant friction loss per foot and therefore as the delivery varies there will be a change in the desired friction loss per foot of pipe.

TABLE 4. PIPING CHECK CHART

Load, *Mbh*			Pipe Length Ft	Elbows	Pipe Size In.	Unit Head Milinches per Ft	Friction Milinches	Total Loss Milinches
Supply Main								
AB	98		37	1	1¼	240	9600	9,600
BC	58		2	4	1¼	90	1080	10,680
CD	38		16	1	1	155	2790	13,470
DE	23		9	0	¾	220	1980	15,450
EF	11		12	0	½	240	2880	18,330
FG	4		16	1	½	50	850	19,180
Return Main								
HI	98		5	5	1¼	240	4320	4,320
IJ	58		11	1	1¼	90	1260	5,580
JK	54		16	1	1	300	5400	10,880
KL	47		11	0	1	230	2530	13,410
LM	35		9	0	1	140	1260	14,670
MN	20		15	1	¾	170	2890	17,560
Radiator Circuits								
CN	20	*Supply*	3	13	¾	170	3910	
		Return	4	2	¾	170	1190	5,100
DM	15	*Supply*	3	19	½	420	9250	
		Return	4	17	¾	96	2880	12,130
EL	12	*Supply*	14	20	½	270	9180	
		Return	15	20	½	270	9450	18,630
FK	7	*Supply*	3	19	½	100	2200	
		Return	4	17	½	100	2100	4,300
GJ	4	*Supply*	8	5	½	50	650	
		Return	9	17	½	50	1300	1,950

It is desirable to check the various circuits so that if the variation from the calculated resistance is too great, it may be compensated by adding additional resistance at the proper point. This may be accomplished by sizing the short circuits by the procedure previously outlined. Prepare a chart such as Table 4 to be used in calculating the resistance of each circuit.

Section AB carries 98 Mbh with a unit head of 240 milinches per foot. In section AB there are 37 ft of pipe and 1¼ in. elbow. At 240 milinches per foot this is equivalent to 9600 milinches total loss in this section. Section BC carries 58 Mbh with a length of 2 ft and 4 elbows. The unit loss in this section is 90 milinches per foot. Loss in this section is then 1080 milinches. Section CD carries 38 Mbh and has 16 ft of pipe and 1 elbow. The unit loss in 1-in. pipe is 155 milinches. The loss in this section is 2790 milinches. The balance of the supply main and the return main are handled in a similar manner.

The radiator circuits are then checked. The 20 Mbh radiator on this circuit has 3 ft of supply pipe and 13 elbow equivalents while the return is composed of 4 ft and 2 elbows. The unit loss in ¾ in. pipe at this delivery is 170 milinches per foot. The total loss in the supply is 3910 milinches. The loss in the return is 1190. Total loss in the radiator circuit is 5100 milinches. Check each radiator circuit in a similar manner.

The total calculated loss for the longest circuit was determined as 60,000 milinches. The maximum loss in the short circuit is 18,630 plus 13,410 plus 15,450 or a total of

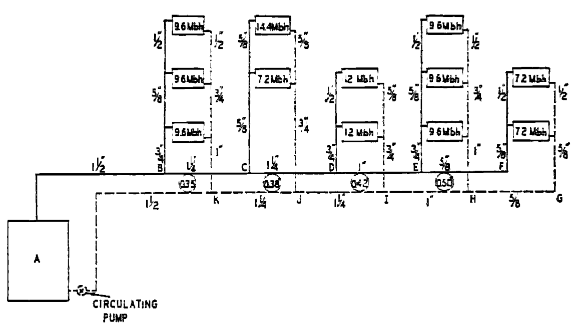


FIG. 5. A FORCED CIRCULATION DIRECT RETURN SYSTEM

47,490 milinches. This difference is caused by the variation in length of the two circuits and may be corrected by using a flow control in the return main to supply the additional resistance or by introducing resistance into each separate circuit to compensate for the difference. A 10 per cent variation will cause no complication as the flow from the various pipes will not exactly follow the curves of Fig. 3 any closer than this value.

Example 2. Design a two-pipe direct return forced circulation system with copper tubing and fittings for the piping layout as detailed in Fig. 5, based on a 20 F temperature drop through the radiation.

The piping circuit from the boiler to the highest radiator on the farthest riser and back to the boiler is 250 ft of pipe. There are about 16 elbow equivalents having an equivalent pipe length of about 50 ft, so that the total equivalent pipe length is 300 ft.

Assume that a circulator is available which will provide a pressure head of 6 ft.

Solution. Refer to Table 2, which indicates the total equivalent lengths for pressure heads from 2 to 12 ft. With a circulator having a 6 ft pressure head and a system with a total equivalent length of 300 ft, the piping system will be designed on a basis of 240 milinch.

Checking the piping diagram it will be noted that sections AB and KA, both supply 117.6 Mbh. Referring to the 240 milinch column of Table 2, 1½ in. is shown to be the necessary pipe size. Sections BC and JK carry 88.8 Mbh and require 1¼ in. tubing. Sections CD and IJ supply 67.2 Mbh and require 1¼ in. tubing. Sections DE and HI supply 43.2 Mbh, which requires 1 in. tubing. Sections EF and GH with a load of 14.4 Mbh require ⅝ in. tubing.

The risers are pipe sized in a similar manner. To secure proper distribution of hot

water in the direct return system among the several risers, it is necessary to introduce resistances to balance the circuit.

The first riser is 80 ft nearer the boiler than the fifth riser. In order that the two may be balanced, that is, operated under equal pressure heads, resistance must be added to

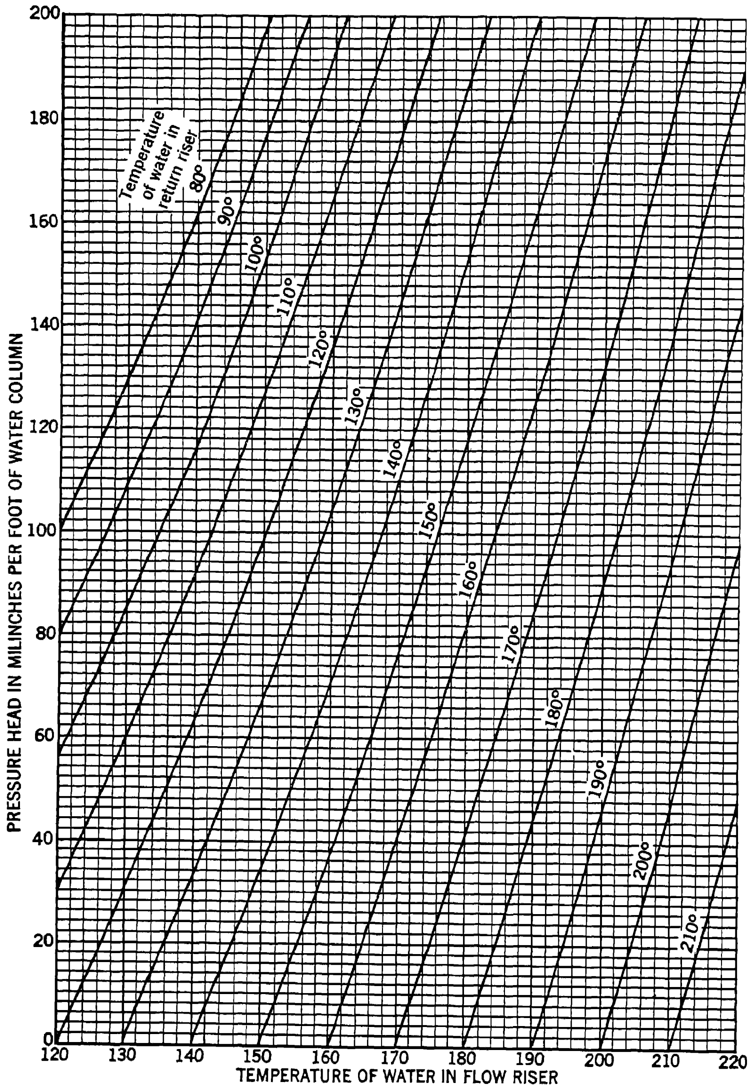


Fig. 6. Gravity Pressure Heads for Various Temperature Differences

the first riser equal to the friction head in the 80 ft of supply main B to F plus the 80 ft of return main G to K for a total of 160 ft of pipe.

Having designed the piping system on a 240 milinch basis, the total friction head in the supply and return mains between the first and fifth risers is therefore 160 × 240 = 38,400 milinches, or 3.2 ft which must be supplied by additional resistance in the first riser. This resistance can be supplied by an adjusting valve or by an orifice of size selected from Table 5.

GRAVITY CIRCULATION PIPE SIZES

In gravity hot water heating systems the difference in temperature (density) between the flow and return produces the circulation of the water. The temperature difference is usually made from 25 to 35 F. Having determined the temperature difference and the temperature of flow, Fig. 6 can be used to obtain the pressure head, and from this point the calculations are the same as for forced hot water. Heat emission rates from 150 to 170 Btu per square foot are commonly used so that flow temperatures range from 180 to 200 F or higher. Assuming a flow temperature of 200 F and 35 F drop and the mains 4 ft above the boiler, a circulating pressure head of 600 milinches results. Assuming first floor radiators 3 ft above the mains and second floor radiators 12 ft above mains, third floor 21 ft and fourth floor 30 ft, the circulating pressure heads are 450, 1800, 3150 and 4500 milinches respectively.

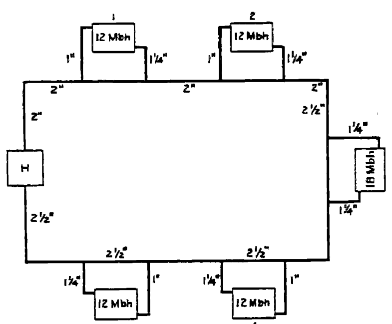


FIG. 7. A ONE-PIPE GRAVITY CIRCULATION SYSTEM

The following examples will illustrate the method to be followed in designing a gravity hot water system:

Example 3. Design a one-pipe gravity circulation system for the layout shown in Fig. 7. Assume that the main circuit consists of 150 ft of pipe, 7 elbows, and one boiler.

Solution. Replace the boiler by 3 elbow equivalents and assume that the size of the main will be about 2 in. According to Table 6, Column 2, a 2-in. elbow is equivalent to 4 ft of pipe, and the total equivalent length of the main will be about 150 plus 40, or 190 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main and that the temperature drop in the system is to be 35 F, Table 6 may be used to determine the size of the mains. Note from Column 8, for a 200 ft length, that a 2-in. main will supply 48 Mbh and a 2½-in. main, 75.4 Mbh. Since the system to be designed is to supply 66 Mbh, a 2-in. pipe is too small and a 2½-in. pipe too large. The solution is to use some 2 in. and some 2½ in. pipe. Since the 2½ in. is nearer the correct size than the 2 in., select 2-in. pipe for the first 50 or 60 ft from the boiler and 2½ in. for the remaining pipe back to the boiler.

Tables 7 and 8 may be used to design the radiator risers and connections. According to Table 7, for 12 Mbh the flow riser should be ¾ in. and the return riser 1 in., and the riser branches should be 1 in. and 1¼ in., respectively. Note that according to Table 8, both radiator tappings should be 1 in. To simplify the construction, select 1-in. flow risers with 1-in. riser branches and 1-in. radiator tappings. Also select 1¼-in. return risers with 1¼-in. riser branches, and 1¼-in. radiator tappings. Similarly, for 18 Mbh, select 1¼-in. flow and return risers and riser branches, and 1¼-in. radiator tappings.

To develop a rule for determining radiator sizes, assume a system similar to that of Fig. 7, in which the total temperature drop is to be 35 F

Table 5. Friction Heads (in Milinches) of Central Circular Diaphragm Orifices in Unions

Diameter of Orifices (Inches)	Velocity of Water in Pipe in Feet per Minute									
	10	15	20	30	40	50	60	90	120	180
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1½-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois, (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

and which is equipped with 7 radiators, all radiators dissipating equal quantities of heat. The mean temperature of the water in the radiators will be reduced 5 F for each successive radiator. If the mean temperature of the water in the first radiator is 200 F, the mean temperature of the water in the seventh radiator will be 170 F, and, according to Table 4, Chapter 12, the heat dissipation of these two radiators will be to each other as 1.62 is to 1.15, or as 140 is to 100, and therefore if the last radiator is to dissipate as much heat as the first, its size must be 40 per cent larger.

Example 4. Design a two-pipe, direct return, gravity circulation system for the layout shown in Fig. 8. Assume that the main circuit from the boiler to the farthest flow riser and from the farthest return riser back to the boiler consists of 160 ft of pipe, 6 elbows, and 1 boiler.

Solution. Replacing the boiler by 3 elbow equivalents and assuming that the largest size of the main will be about 3 in., the total equivalent length of the main will be 160 plus 45, or 205 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main, and that the temperature drop will be 35 F for the

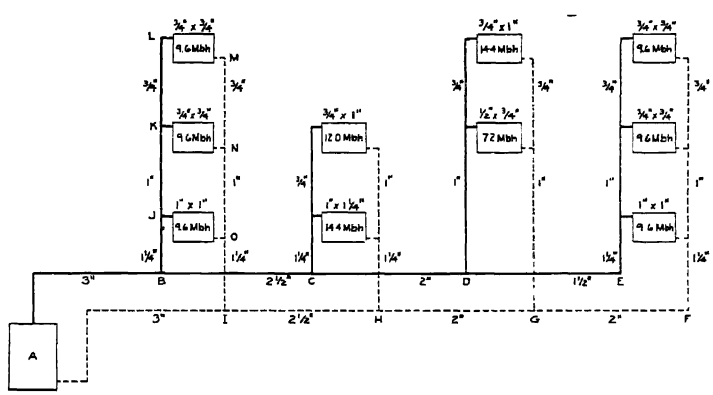


FIG. 8. A TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEM

system, the pressure head caused by the difference in weight between the water in the flow and return risers joining the mains to the boiler will be about 0.6 in. of water.

Table 6 may be used to determine the size of the main as follows: Refer to Column 8 and note that for Sections *AB* and *IA*, which supply 105.6 Mbh, a 3-in. pipe is too large and a 2½-in. pipe is too small; hence, select 2½ in. rather than 3 in. as noted in Fig. 8 for Section *AB* and 3 in. for Section *IA*. For Sections *BC* and *HI*, which supply 76.8 Mbh, a 2½-in. pipe is almost exactly the correct size and is selected for both sections.

Tables 7 and 8 are based on the assumption that the boiler pressure head must be equal to the friction head in the mains, and that the several radiator pressure heads must be equal to the respective radiator and riser friction heads.

To design the radiator risers, use Table 7 and begin with the set nearest the boiler. The first floor risers must supply 28.8 Mbh. According to the table, 1¼-in. flow and return risers will supply 26.0 Mbh; if the return riser is increased to 1½ in., the capacity will be increased to 34.0 Mbh. This is considerably larger than necessary, and 1¼-in. flow and return risers are selected. However, it must be remembered that the riser branches, which are the connections from the flow and return mains to the flow and return risers, are to be one size larger than the risers.

The second floor risers must supply 19.2 Mbh. According to the table, the capacity of 1 in. flow and return risers is 20.0 Mbh, and that size is selected.

The third floor risers must supply 9.6 Mbh. If a ½-in. flow and a ¾-in. return riser

TABLE 6. CAPACITIES OF MAINS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 0.6 IN., A TEMPERATURE DROP OF 35 F, WHEN THE MAINS ARE 4 FT ABOVE THE CENTER OF THE BOILER

1	2	3	4	5	6	7	8	9	10	11
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[a])	Equivalent Total Length of Pipe in Feet in Longest Circuit								
		75	100	125	150	175	200	250	300	350
		Unit Friction Head, in Milinches								
		8.0	6.0	4.8	4.0	3.4	3.0	2.4	2.0	1.7
1½	3.0	*43.0*	*37.5*	*33.0*	*30.0*	*27.0*	*25.0*	*22.2*	*20.2*	*18.7*
2	4.0	*83.0*	*72.0*	*63.0*	*57.0*	*51.0*	*48.0*	*42.0*	*38.0*	*35.0*
2½	4.5	*140.0*	*115.0*	*100.0*	*90.0*	*81.5*	*75.4*	*67.2*	*61.0*	*56.0*
3	5.0	*234.0*	*204.0*	*175.5*	*160.0*	*143.0*	*133.0*	*110.0*	*107.5*	*100.0*
3½	5.5	*347.0*	*300.0*	*260.0*	*236.0*	*214.0*	*200.0*	*177.0*	*160.0*	*146.0*
4	6.0	*490.0*	*422.0*	*370.0*	*334.0*	*297.0*	*278.0*	*248.0*	*223.0*	*205.0*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

are used, the capacity will be 8.0 Mbh; if both risers are ¾ in., the capacity will be 14.0 Mbh. The ¾-in. pipe is selected for both risers.

To design the radiator connections, use Table 8 and note that for the first floor radiator connections the capacity of a ¾-in. flow and 1-in. return is 9.1 Mbh, and that of a 1-in. flow and a 1-in. return is 12.5 Mbh. The former is more nearly the correct size, but since it is difficult to secure a good flow through first floor radiators, the 1-in. flow and return connection is selected. For the two upper floors, the capacity of a ¾-in. flow and return connection is 10.5 Mbh, and that size is used.

TABLE 7. MAXIMUM CAPACITIES OF RISERS[a] IN *Mbh*, AND **Velocities of Water in Pipes in Inches Per Second** FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A DROP OF 35 F THROUGH EACH RADIATOR

Pipe Size (Inches)		Equivalent Length of Pipe (Feet[c])	1st Floor[b]			2nd Floor	3rd and 4th Floors
				Vel (In. per Sec[d])			
Flow	Return		*Mbh*	Flow	Return	*Mbh*	*Mbh*
½	½	1.0				*5.0*	*6.2*
½	¾					*6.4*	*8.0*
¾	¾	1.5	*9*	**2.3**	**2.3**	*10.1*	*14.0*
¾	1		*12*	**3.2**	**2.0**	*12.8*	*17.1*
1	1	2.0	*18*	**2.5**	**2.5**	*20.0*	*26.0*
1	1¼		*21*	**3.0**	**2.0**	*25.2*	*34.0*
1¼	1¼	3.0	*26*	**3.0**	**3.0**	*43.0*	*55.0*
1¼	1½		*34*	**4.0**	**2.5**		
1½	1½	3.5	*48*	**3.0**	**3.0**		

[a]This table is based on pressure heads of 450, 1800, 3150, and 4500, respectively, for the first, second, third, and fourth floor radiators, and on friction heads of 200 milinches for the first floor radiators and connections, and 700 milinches for all other radiators and their connections.

[b]The riser branches, the piping which connects the risers to the mains, are to be one size larger than the risers.

[c]Approximate length of pipes in feet equivalent to one elbow in friction head. This value varies with the velocity.

[d]Velocities apply to the riser branches.

TABLE 8. MAXIMUM CAPACITIES OF RADIATOR CONNECTIONS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TEMPERATURE DROP OF 35 F THROUGH EACH RADIATOR

Pipe Size		Equivalent Length of Pipe (Feet[a])	1st Floor	2nd, 3rd, and 4th Floors
Flow	Return		*Mbh*	*Mbh*
½	½	1.0	*4.1*	*5.9*
½	¾		*5.2*	*7.5*
¾	¾	1.5	*7.0*	*10.5*
¾	1		*9.1*	*13.0*
1	1	2.0	*12.5*	*17.8*
1	1¼		*17.5*	*23.2*
1¼	1¼	3.0	*23.3*	*33.2*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

As explained in the design of the forced circulation system of Fig. 5, the two-pipe direct return system of Fig. 8 will not function correctly unless its four sets of risers are balanced among themselves. This necessary balancing is accomplished by adding resistances to all risers, except the one farthest from the boiler, equal to the excess boiler pressure heads available for those risers above the boiler pressure head available for the farthest riser. For example, the first set of risers is 60 ft nearer the boiler than the last set. Since the flow and return mains are designed for a friction head of 3 milinches per foot (see Table 6, Column 8), the boiler pressure head available for the first set of risers is 360 milinches in excess of that available for the fourth set. The velocity in the riser branch is 3 in. per second (see Table 7) and, therefore, according to Table 5, an 0.65-in. orifice in a 1¼-in. union should be used. This will provide a resistance of about 420 milinches. In the same manner it is found that for the second set of risers a resistance of 240 milinches is required and that an 0.70-in. orifice in a 1¼-in. union will provide a resistance of 285

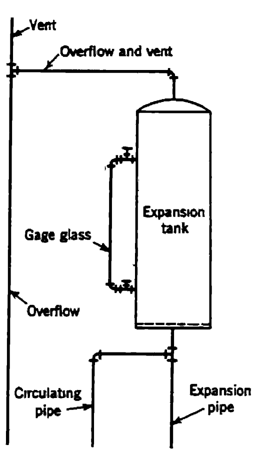


FIG. 9. AN OPEN EXPANSION TANK

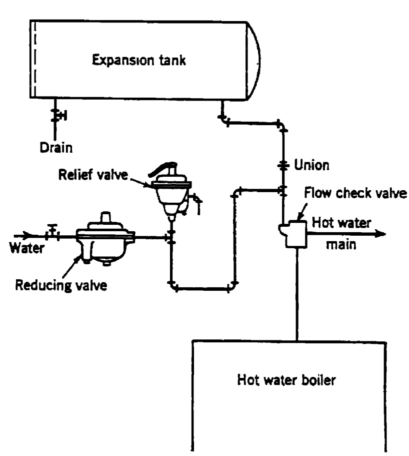


FIG. 10. A CLOSED EXPANSION TANK

milinches. For the third set of risers, a resistance of 120 milinches is required and an 0.60-in. orifice in a 1-in. union will provide sufficient resistance.

EXPANSION TANKS

Expansion tanks may be either of the open or of the closed type. In the open type, (see Fig. 9) the water is subject to atmospheric pressure only, but in the closed tank (see Fig. 10) the system is under pressure and, therefore, a relief valve should be placed on the tank. Water expands about 4 per cent when being heated from 40 F to 200 F, and the expansion tank should have a volume about twice the actual expansion or about 8 per cent of the total volume of water in the entire system including boiler, radiators, pipes, etc. Open expansion tanks should be at least 3 ft above the highest radiator and be protected against freezing. Closed tanks are generally placed in the cellar over the boiler.

A relief valve installed on a closed tank will not operate often provided the tank is of adequate size. It is essential that the relief valve be kept in good condition to eliminate any possible failure when operation is necessary.

INSTALLATION DETAILS

Attention should be paid to the following:

All piping must be so pitched that all air in the system can be vented either through an open expansion tank, radiators or automatic relief valves.

All piping must be arranged so that the entire system can be drained. Sections of piping individually valved shall have corresponding drain valves.

In large buildings, the piping may be zoned according to exposure of building, usage of building, or method of control.

All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

The pipe system should be designed so that each circuit has its correct friction head for balanced water distribution. This may be done by change of pipe size or change in piping detail.

The connections from the boiler to the mains should be short and direct, to reduce the friction head and allow for expansion. It is frequently possible to avoid an elbow and to reduce the length of the pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

Generally connections to risers or radiators are taken out of the top of mains either 45 or 90 deg. In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side.

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end. With gravity circulation the flow connection may be either at the top or at the bottom of the radiator. With short radiators both flow and return may be at same end, but top and bottom.

Unless used as heating surface, all piping, both flow and return, should be insulated.

Chapter 16

DISTRICT HEATING

Steam Distribution Piping, Selection of Pipe Sizes, Provision for Expansion, Capacity of Returns with Various Grades, Conduits for Piping, Pipe Tunnels, Inside Piping, Steam Requirements, Fluid Meters and Metering, Rates, Utilization, Automatic Temperature Control

THOSE phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories. Some data are included to cover the piping peculiar to heating systems which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

STEAM DISTRIBUTION PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per hour per square foot of equivalent heating surface is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Miscellaneous steam requirements such as laundry, cooking, or process should be individually calculated.

The steam requirements for water heating should be taken into account,

but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend upon (1) boiler pressure, (2) whether exhaust or live steam, (3) pressure requirements of apparatus to be served. If steam has been passed through electrical generating units, the pressure will be considerably lower than if live steam, direct from the boilers, is used.

The advantages of low pressure distribution (2 to 30 lb per square inch) are (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, (3) simpler problems in pressure reduction at the buildings, and (4) general reduction in maintenance costs. With distribution pressures not exceeding 40 lb per square inch there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than this a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators.

The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than building heating.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 lb per square inch, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 lb per square inch is usually demanded although 30 lb per square inch is sufficient if the mangle is equipped with a large number of rolls and if a slow rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 lb per square inch.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of the Unwin pressure drop formula. This is one of several formulae which may be used. Unwin's formula, which gives pressure drops slightly larger than actual test results, is as follows:

$$P = \frac{0.0001306\ W^2L\left(1 + \frac{3.6}{D}\right)}{dD^5} \tag{1}$$

where

P = pressure drop, pounds per square inch.
W = weight of steam flowing, pounds per minute.
L = length of pipe, feet.
D = inside diameter of pipe, inches.
d = average density of steam, pounds per cubic foot.

This formula is similar to the Babcock formula given in Chapter 14.

Information on provision for expansion will be found in Chapter 17. Where steam and return piping are installed in the same conduit, the return piping usually follows the same grade as the steam piping. In general, the condensation is pumped back under pressure. Where the condensation returns by gravity, Table 1 gives the sizes of the return piping. It is evident that at points where the grade is great, smaller pipes can be installed.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural or cast steel set in concrete. In general, cast steel is preferable to structural steel.

TABLE 1. CAPACITY OF RETURNS FOR UNDERGROUND DISTRIBUTION SYSTEMS IN POUNDS OF CONDENSATE PER HOUR

Size[a] of Pipe In.	Pitch of Pipe per 100 Ft						
	6″	1′	2′	3′	5′	10′	20′
1	448	998	1890	2240	3490	5490	7490
1¼	1740	2490	3990	4880	6480	9480	13500
1½	2700	4190	5740	7480	9480	14500	20900
2	4980	7380	10700	13900	16900	24900	36900
3	13900	22500	30900	37400	50400	74800	105000
4	30900	44800	64800	79700	105000	154000	229000
5	54800	79800	120000	144800	195000	294000	418000
6	90000	138000	187000	237000	312000	449000	--------
8	190000	277000	404000	508000	660000	938000	--------
10	344000	498000	724000	900000	1190000	--------	--------
12	555000	798000	1148000	1499000	1990000	--------	--------

[a]Size of pipe should be increased if it carries any steam.

In laying out underground conduits the following points should be borne in mind:

1. The depth of the buried conduit should be kept at a minimum. Excavation costs are a large factor in the total cost.

2. An expansion joint, offset, or bend should be placed between each two anchors.

3. If the distance between buildings is 150 ft or less and the steam line contains high-pressure steam, the line may be anchored in the basement of one building and allowed to expand into the basement of the second building. If the steam line contains low-pressure steam (up to 4-lb pressure), this method may be used if buildings are 250 ft or less apart.

4. If the distance between buildings is between 150 ft and 300 ft and the steam line contains high-pressure steam, the lines should be anchored midway between the buildings and allowed to expand into the basements of both buildings. If the steam line contains low-pressure steam this method may be used if buildings are between 250 ft and 600 ft apart. No manhole is required at the anchor, and a blind pit is all that is necessary.

5. For longer lines, manholes must be located according to experience, physical conditions and the expansion value of the type of expansion joint or bend that is used. The

minimum number of manholes will be required when an expansion bend, or an anchor with double expansion joint, is placed in each manhole and the pipes are anchored midway between manholes.

6. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic test pressure should be one and one-half times the maximum allowable pressure and it should be held for a period of at least two hours without evidence of leakage. In any case the pressure should be no less than 100 lb per square inch.

There are many types of conduits, some of which are manufactured products and some of which are built in the field. The styles and construction of conduits commonly used may be classified as follows. Some of the more common forms are illustrated in Fig. 1.

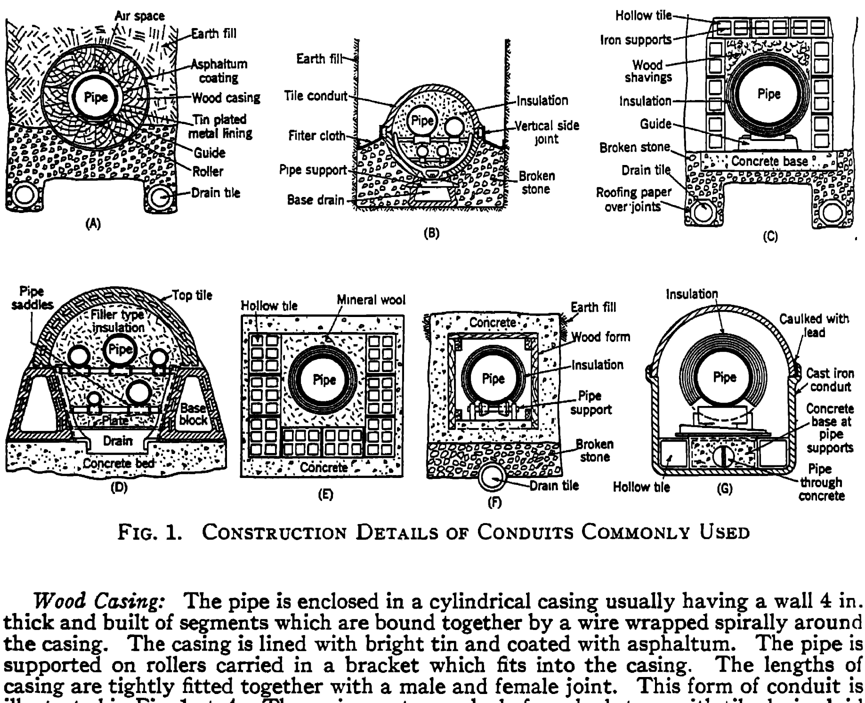


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

Wood Casing: The pipe is enclosed in a cylindrical casing usually having a wall 4 in. thick and built of segments which are bound together by a wire wrapped spirally around the casing. The casing is lined with bright tin and coated with asphaltum. The pipe is supported on rollers carried in a bracket which fits into the casing. The lengths of casing are tightly fitted together with a male and female joint. This form of conduit is illustrated in Fig. 1 at *A*. The casing rests on a bed of crushed stone with tile drains laid below. The tile drains are of 4-in. field tile or vitrified sewer tile, laid with open joints.

Filler Type: The pipes are supported on expansion rollers properly supported from the conduit or independent masonry base. The pipes are protected by a split-tile conduit, and the entire space between the pipes and the tile is filled with an insulating filler. Thus the pipes are nested and the insulation between them and the tile effectively prevents circulation of air. The conduit is placed on a bed of gravel or crushed rock from 4 to 6 in. thick, which is extended upward so as to come about 2 in. above the parting lines of the tile. A tile underdrain is placed beneath the conduit throughout the entire length and is connected to sewers or to some other point of free discharge. At *B* and *D* in Fig. 1 are shown two forms of tile conduit of the filler type.

Circular Tile or Cast-Iron Conduit: The pipes are carried on expansion rollers supported on a frame which rests entirely on the side shoulders of the base drain foundation. The pipes are protected by a sectional tile conduit, scored for splitting, or a cast-iron conduit, both being of the bell and spigot type. The conduit has a longitudinal side joint

for cementing, after the upper half of conduit is in place, so shaped that the cement is keyed in place while locking the top and bottom half of the conduit together with a water-tight vertical side joint. The cast-iron conduit has special side locking clamps in addition to the vertical side joint. The entire space between the conduit and the pipes is filled with a water-proofed asbestos insulation. The conduit is supported on the base drain foundation, each section resting on two sections of the base drain, thus interlocking. The base drain is so shaped that it provides a cradle for the conduit, resting solidly on the trench bottom and providing adequate drainage area immediately under the conduit. The underdrain is connected to sewers or some other point of free discharge. For tile conduit the base drain is vitrified salt glazed tile and for cast-iron conduit it is either extra heavy tile or cast-iron. A free internal drainage area is also provided to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint in the conduit. Broken stone is filled in around the base drain and up to the vertical side joint. The broken stone is covered with an asphalted filter cloth to prevent sand from sifting through the broken stone and clogging the drainage area of the base drain. The tile conduit is made in 2-ft lengths and the cast-iron conduit in 4-ft lengths, cast in separate top and bottom halves. Special reinforcing ribs give the cast-iron conduit ample strength with minimum weight.

Insulated Tile Type: The insulating material, diatomaceous earth, is molded to the inside of the sectional tile conduit. The space between the pipes and the insulating conduit lining may also be filled with insulation. The pipes are carried on expansion rollers supported on a frame which rests on the side shoulders of the base drain foundation. This type of conduit has the same mechanical features as those described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Cast-Iron): Each pipe is insulated in the usual way with any desired type of sectional pipe insulation over which is placed a standard waterproof jacket with cemented joints. The pipes are enclosed in a sectional tile or cast-iron conduit as described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Concrete Trench): A type of construction frequently used in city streets, where service connections are required at frequent intervals, the pipes are insulated as described in the preceding paragraph, and are enclosed in a box or trench made either entirely of concrete, or with concrete bottom and specially constructed tile sides and tops. The pipes are supported on roller frames secured in the concrete. At *C* and *E*, Fig. 1, are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The use of loose material in addition to the sectional insulation is, of course, optional and is only justifiable where high pressure steam is used. The conduit shown at *F* is of a similar type and has the advantage of being made entirely of concrete and other common materials.

Sectional Insulation Type (Bituminized Fibre Conduit): Each pipe is individually insulated and encased in a bituminized fibre conduit. The insulating material is 85 per cent carbonate of magnesia sectional pipe covering, applied in the usual manner as on overhead pipes, except that bands are omitted. After every fifth section of magnesia covering there is applied a short, hollow section of very hard asbestos material in the bottom portion of which rests a grooved-iron plate carrying ball-bearings upon which the pipe rides when expanding or contracting. This short expansion section is of the same outside diameter as the adjacent 85 per cent magnesia covering. Over the pipe covering and expansion device there are placed two layers of bituminized fibre conduit with all joints staggered, and the surface of each conduit is finished with liquid cement. Conduits are placed on a bed of crushed rock or gravel, approximately 6 in. deep, and this is extended upward to about the center line of the conduit when trench is backfilled. Underdrains leading to points of free discharge are placed in the gravel or crushed rock beds.

Special Water-Tight Designs: It is occasionally necessary to install pipes in a very wet ground, which calls for special construction. The ordinary tile or concrete conduit is not absolutely water tight even when laid with the utmost care. The conduit shown at *G*, Fig. 1, is of cast-iron with lead-calked joints and is water-tight if properly laid. It is obviously expensive and is justified only in exceptional cases. A reasonably satisfactory construction in wet ground is the concrete or tile conduit with a water-proof jacket enclosing the pipe and its insulation, and with the interior of the conduit carefully drained to a manhole or sump having an automatic pump. It is useless to install external drain tile when the conduit is actually submerged.

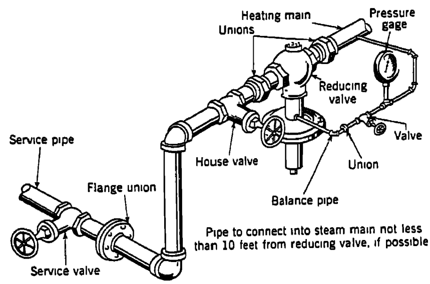


FIG. 2. CONNECTIONS FOR REDUCING VALVES OF SIZE LESS THAN 4 IN.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

On account of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes not installed where provision for the heating lines is the only consideration, but only where they are required to accommodate miscellaneous other services or provide underground passage between buildings.

OVERHEAD DISTRIBUTION

In some industrial and institutional applications, the distribution piping may be installed, entirely or in part, above ground. This method of construction has the advantage of requiring no excavation and being easily maintained.

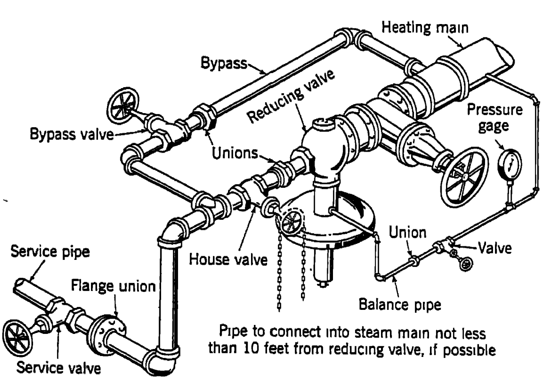


FIG. 3. CONNECTIONS FOR REDUCING VALVES OF SIZE 4 IN. AND LARGER, AND FOR EXPANDED VALVES

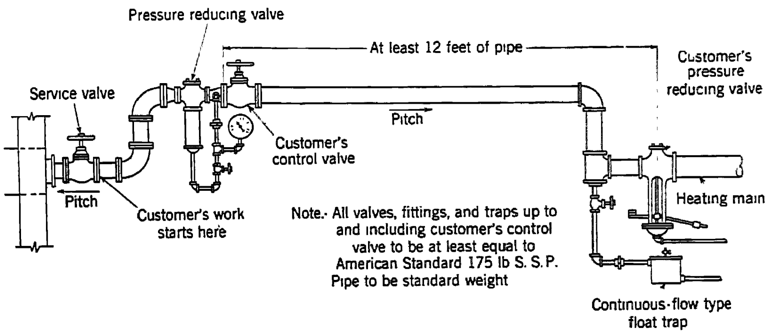


FIG. 4. STEAM SUPPLY CONNECTION WHEN USING TWO REDUCING VALVES

INSIDE PIPING

Figs. 2 and 3 show typical service connections used for low pressure steam service. As shown in Fig. 2, no by-pass is used around the reducing valve on sizes less than 4 in. Fig. 3 illustrates the use of a by-pass around reducing valves 4 in. and larger. This latter construction permits the operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam until repairs are made.

Fig. 4 shows a typical installation used for high pressure steam service. The first reducing valve effects the initial pressure reduction. The second reducing valve reduces the steam pressure to that required.

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer. There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

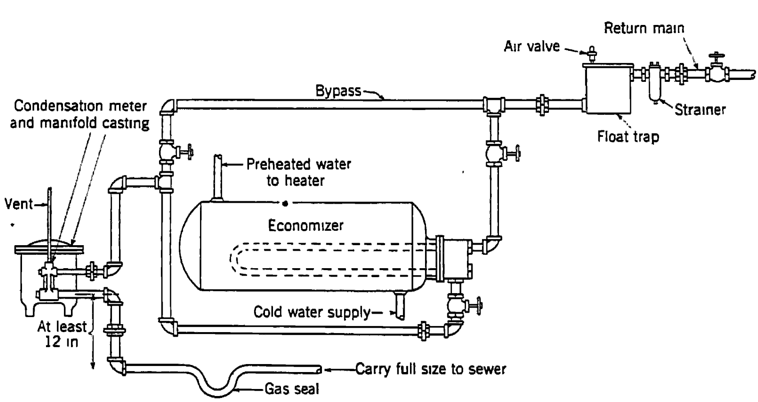


FIG. 5. RETURN PIPING FOR CONDENSATION METER

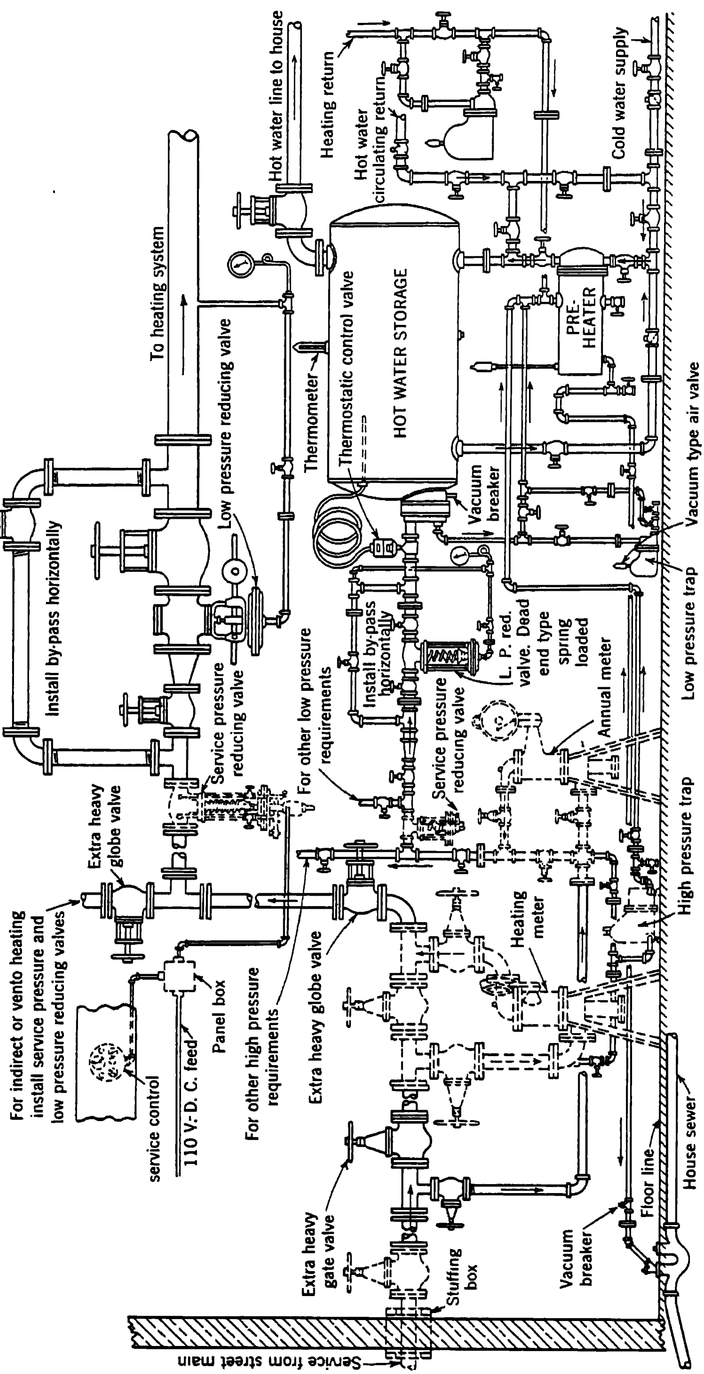


Fig. 6. Typical Service Installation also Showing Water Heater and Economizer Connections

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very cold weather without endangering plumbing. It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling coil, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building. Fig. 6 shows a typical steam service installation for high pressure steam, complete for steam flow metering, water heating, preheating, automatic heating control, and for using steam for other purposes.

The condensation from the heating system, after leaving the trap, passes through the economizer. The supply to the hot water heater passes through the economizer, absorbing heat from the condensate. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the economizer and the water heater proper, not at the economizer inlet, because the recirculated hot water is itself at a high temperature. The number of square feet of heating surface in the economizer should be approximately equal to one per cent of the equivalent square feet of heating surface in the building.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensate if storage capacity is provided for the preheated water. Frequently a type of economizer is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

This may be done in several ways, as by the use of temperature controls of various types or by orifice systems. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds pressure, thus producing some control over the heat output of the heating system by varying the temperature of the steam in the radiators. Several proprietary systems are on the market which accomplish this automatically, either with outdoor or indoor controls or a combination of both. One form of control which appears to be well suited for controlling district steam service to a building is the weather compensating control. It regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, and at the same time insures delivery of steam to all the radiators. This type of control can be equipped with time clocks and thermostats to provide a warming-up period in the morning.

Another form of regulation, known as the time-limit control, is sometimes employed for regulating the steam supply from the central station main to the building. Such a control provides an intermittent supply of steam to the radiators either throughout the 24 hours of the day or during the daytime hours only. The setting of a switch may provide no service, continuous service, or periodic service. For the latter, by means of several intermittent settings, steam will be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. These settings afford from 15 to 80 per cent of the maximum heating effect required on days of zero temperature. A night switch with a variety of settings may be adjusted so as to maintain throughout the night the intermittent supply called for by the day switch setting, or may be set to interrupt the opera-

tion of the day switch and entirely cut off the supply of steam to the radiation at night during certain hours which are selected by the operating engineer.

The maximum in economical operation and satisfactory heating can only be obtained by the use of some automatic temperature control system.

FLUID METERS

The perfection of fluid meters has contributed more to the advancement of district heating than any other one thing. These meters may be classified as follows:

1. *Positive Meters:* The fluid passes in successive isolated quantities—either weights or volumes. These quantities are separated from the steam and isolated by alternately filling and emptying containers of known capacity.

2. *Differential Meters:* In the differential meter, the quantity of flow is not determined by simple counting, as with the positive meter, but is determined from the action of the steam on the primary element.

Additional sub-divisions of these two general classifications can be made as follows:

- Fluid Meters
  - Positive - Quantity
    - Weighing
      - Weighers
      - Tilting trap
    - Volumetric
      - Rotary
      - Bellows
  - Differential
    - Quantity - Current - Turbine
    - Rate of flow
      - Head (Kinetic)
        - Venturi
        - Flow nozzle
        - Orifice
        - Pitot tube
      - Area (Geometric)
        - Orifice and plug
        - Cylinder and piston
      - Head area (Weir)
        - V-notch
        - Special notch

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations:

1. Its use in a new or an old installation.
2. Method to be used in charging for the service.
3. Location of the meter.
4. Large or small quantity to be measured.
5. Temporary or permanent installation.
6. Cleanliness of the fluid to be measured.
7. Temperature of the fluid to be measured.
8. Accuracy expected.
9. Nature of flow: turbulent, pulsating, or steady.
10. Cost.
    - (*a*) Purchase price.
    - (*b*) Installation cost.
    - (*c*) Calibration cost.
    - (*d*) Maintenance cost.
11. Servicing facilities of the manufacturer.
12. Pressure at which fluid is to be metered.
13. Type of record desired as to indicating, recording or totalizing.
14. Stocking of repair parts.

15. Use of open jets where steam is to be metered.
16. Metering to be done by one meter or by a combination of meters.
17. Use as a check meter.
18. Its facilities for determining or recording information other than flow.

Condensation Meters

The majority of the meters used by district heating companies in the sale of steam to their customers are condensation meters.

The condensation meter is a popular type for use on small and medium sized installations, where all of the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Two types of condensation meters are in general use: the *tilting bucket* meter and the *revolving drum* or *rotor* meter of which there are several makes on the market. Condensation meters should not be operated under

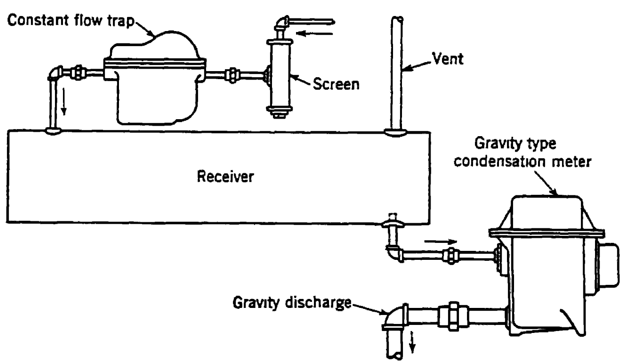


FIG. 7. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

pressure; they are made for either gravity or vacuum installation. Continuous flow traps are necessary ahead of the meter if a vented receiver is not used. Where bucket traps are used, a vented receiver before the meter is essential. If desirable a receiver may be used with a continuous flow trap, but this is not necessary.

Fig. 7 illustrates a gravity installation using a vented receiver ahead of the meter, while Fig. 8 shows a vacuum installation without a master trap.

Flow Meters

Steam flow meters are available in many types and combinations. The *orifice* and *plug* meter is one in which the steam flow varies directly as the area of the orifice. The vertical lift of the plug, which is proportional to the flow, is transmitted by means of a lever to an indicator and to a pencil arm which records the flow on a strip chart. The total flow over a given period is obtained by measuring the area by using a planimeter on the chart and applying the meter constant.

Fig. 9 shows a typical orifice-type meter connection and indicates typical requirements in the installation of this type of meter.

Flow meters using an orifice, Venturi tube, flow nozzle, or Pitot tube as the primary device are made by a number of manufacturers and can be obtained in either the mechanically or electrically operated type. The electric flow meter makes it possible to locate the instruments at some distance from the primary element.

Flow meters employing the orifice, Venturi tube, flow nozzle or Pitot tube should be so selected as to keep the lower operating range of the load above 20 per cent of the capacity of the meter. This is desirable for accuracy as the differential pressure at light loads is too small to properly actuate the meter. A few general points to be considered in installing a meter of this type are:

1. It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available.

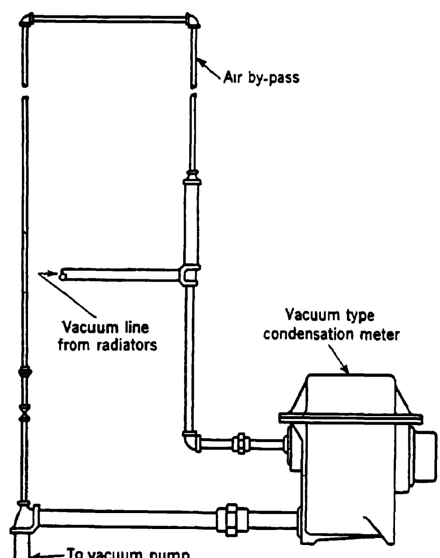


FIG. 8. VACUUM CONDENSATION METER INSTALLATION WITHOUT MASTER TRAP

2. Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company.
3. The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above.
4. Meter piping should be kept free from leaks.
5. Sludge should not be permitted to collect in the meter body.
6. The meter body and meter piping should be kept above freezing temperatures.
7. It is best not to connect a meter body to more than one service.
8. Special instructions are furnished for metering a turbulent or pulsating flow.

STEAM REQUIREMENTS

Steam requirements for heating various types of buildings are given in Chapter 10.

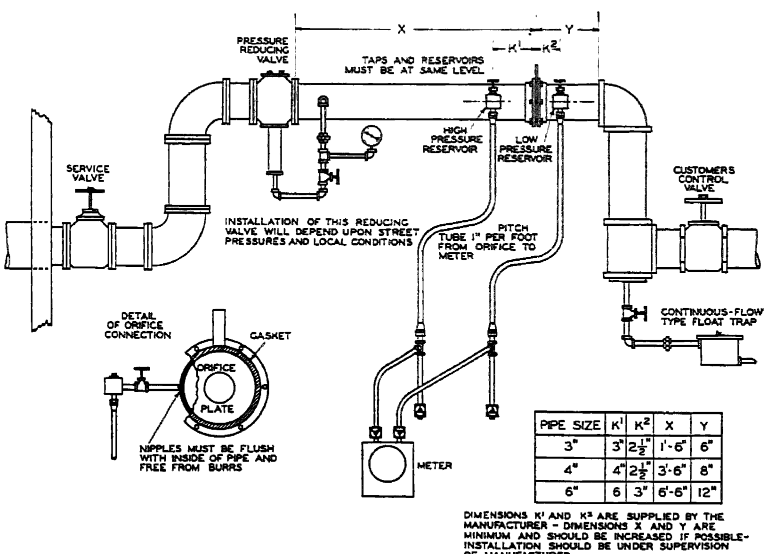
Steam requirements for water heating can be satisfactorily estimated

by using a consumption of 0.0025 lb per day per cubic foot of heated space for office buildings, and 0.0065 lb per day per cubic foot for apartment houses.

Additional data on steam requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association.*

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are



PIPE SIZE	K^1	K^2	X	Y
3"	3"	$2\frac{1}{2}$"	1'-6"	6"
4"	4"	$2\frac{1}{2}$"	3'-6"	8"
6"	6	3"	6'-6"	12"

FIG. 9. ORIFICE METER STEAM SUPPLY CONNECTION

other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not conform with this principle. Lastly, the rate schedule must be reasonably easy for the intelligent layman to comprehend.

Depreciation should be based on a careful estimate of the life of various elements of the property. Appropriations to reserves should be made, with generosity in good years and with discretion in less favorable years.

Glossary of Terms

Load Factor. The ratio, in per cent, of the average hourly load to the

maximum hourly load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface or required radiator surface and the demand of the particular installation. It varies from 0.25 to 0.3 lb per hour per square foot of surface.

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

A. Flat Rates.
- 1. Radiator surface charge. *Obsolescent.*

B. Meter Rates.
- 1. Straight-line.
- 2. Step. *Obsolescent.*
- 3. Block.
  - (*a*) Class rates.

C. Demand Rates.
- 1. Flat demand.
- 2. Wright.
- 3. Hopkinson.
- 4. Doherty (or Three charge)

Straight-Line Meter Rate. The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers.

Block Meter Rate. The pounds of steam consumed by a customer are divided into blocks of thousands of pounds each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefitted at the expense of the others.

Demand Rates. These refer to any method of charge based on a measured maximum load during a specified period of time.

The *flat demand rate* is usually expressed in dollars per *M* lb of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a flow meter is not practicable.

The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand. It is seldom used but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements:

(*a*) A charge based upon the demand, either estimated or measured.
(*b*) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates.

The *Doherty rate* is divided into three elements:

(*a*) A charge based upon demand.
(*b*) A charge based upon steam consumed.
(*c*) A customer charge.

In the *Hopkinson rate*, the last two elements are combined into one element.

Demand rates are comparatively new and are not yet widely used; though they are equitable and competitive they are difficult for the average layman to understand.

They are of benefit to utility companies and to consumers because the investment and operating costs can be divided to suit the particular circumstances into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate. Demand rates are an advantage to the customer in that the use of such a rate reduces the rate per thousand pounds to the long-hour user.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per *M* lb of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure.

UTILIZATION

Considerable savings can be made by the proper and intelligent operation of heating systems. It should be borne in mind that a heating system is designed to heat a building to 70 F inside when the outside temperature is at its lowest point for that particular locality. There is a tendency to overheat the building at any time the outside temperature is above the design temperature unless some method of regulation is used, either automatic or manual.

The general rules for economical operation are as follows:

1. Weatherstrip all windows, and calk all window frames.

2. Provide revolving or vestibule doors on all entrances. Separate shipping and receiving rooms by partitions so that the ever-open large doors will not ventilate the entire building.

3. Keep the radiation near the outside walls, under the windows, if possible.

4. Eliminate all unnecessary ventilation. Ventilating equipment is sized to meet extreme requirements. Do not supply ventilation to a theater or auditorium adequate for an audience of 2000 when there are only 200 present.

5. Determine the hours that heating is required during the day and see that the steam is shut off for the maximum time at night, on Sundays, and holidays.

6. Shut steam off entirely in unoccupied sections of the building, taking care to avoid freezing the water in the plumbing system.

7. Shut off steam during the day whenever possible. During the year steam can be shut off about 55 per cent of the total daytime, and the saving is proportional. An automatic control will do it, but it can be done by hand with amazingly good results.

8. Determine the temperature required for the occupancy of the building. Do not heat a storage garage or a furniture warehouse to the temperature required in a hospital ward.

9. Provide some good means of temperature control.

10. In a hot water heating system keep the temperature of the water down to correspond with existing outdoor temperatures.

11. In a vacuum system maintain a high vacuum. If this is not possible, locate and eliminate all leaks.

12. Install separate lines for those parts of the building that require long-hour or all-night heating. It is much cheaper than heating the entire building all night.

13. See that the entire system responds rapidly when steam is turned on. Locate and eliminate the cause of any sluggish circulation. Balance the radiation, provide adequate air elimination, and correct any trapped run-outs to provide quick system drainage.

14. Keep the system in good repair. Worn, damaged, or defective valves and traps will not function properly.

15. Insulate all steam pipes not used as heating surface.

16. Do not obstruct radiators or prevent the free circulation of air around them; to do so seriously reduces the heating capacity of a radiator.

17. Extract the heat in the condensate for hot water or some other useful purpose.

18. Provide thermometers and recording pressure gages so that the engineer can operate the system with full knowledge of what he is accomplishing.

19. Make all valves and controls convenient and accessible, either direct or through remote control. It is only human nature to delay and avoid doing that which is inconvenient.

20. Keep a daily record consistently, based on weather requirements, and watch it every day.

21. Control the heat supplied to water tanks located on or above the roof. Such tanks require heat to prevent freezing. No heat is required when the temperature in the tank is above 32 F.

22. Investigate every complaint of *no heat* by tenants; find the cause and correct it. Do not overheat an entire building to correct a local condition in one room.

AUTOMATIC TEMPERATURE CONTROL

As stated in Chapter 33, Automatic Control, properly applied to heating, ventilating and air conditioning systems makes possible the maintenance of desired conditions with maximum operating economy.

In addition to the large possibilities for economy, the use of adequate temperature control provides more healthful, comfortable, and efficient working conditions in the buildings because through its use the building is uniformly heated with correct temperatures, and drafts from open windows and overheating are eliminated.

There are many types of temperature control available, each adaptable to a particular type of building, but all require uniform distribution of steam and proper venting.

Before the installation of any type of modern temperature control equipment, it is necessary to see that the heating system is put in good operating condition. In general, the heating system in a building is not given the attention that other mechanical equipment is given because it will continue to function, after a fashion, even though changes in piping, location of radiation, settlement of piping, and the normal wear and tear or other changes have taken place. Through all this depreciation of the system, it becomes more and more costly to operate and parts of the building have to be greatly overheated in order to prevent underheating in a small section of the building. Vents, traps, vacuum pumps, and valves should be given a careful inspection and replaced or repaired if required. The piping should be of adequate size and graded properly. The return piping should have a careful inspection, and any pockets or lifts removed and properly vented. These inspections and repairs are not costly and prevent a much greater outlay in future years. In most cities district heating companies will be willing to make a survey of heating systems and offer recommendations as to operation and changes in piping layout.

The selection of control equipment depends upon the type and size of building and the degree of saving possible.

Chapter 17

PIPE, FITTINGS, WELDING

Pipe Material, Types of Pipe Used, Dimensions of Pipe Commercially Available, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Welding as Applied to Erection of Piping, Valves, Corrosion of Piping

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning work are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought iron, copper and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installation. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe*, the latter as *seamless pipe*.

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld, resistance-weld, or butt-weld process. While the lap-weld and resistance-weld processes produce a better weld than the butt type, lap-weld and resistance-weld pipe is seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operation. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is claimed to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher

first cost is said to be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast Ferrous Pipe. There are now available several types of cast-ferrous metal pipe made of a good grade of cast-iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft with external and internal diameters closely approximating those of extra strong wrought pipe. Cast ferrous pipe may be obtained coupled, beveled for welding, or with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element and iron pipe bearing a small alloy of copper and molybdenum have been claimed to possess more resistance to corrosion than plain steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to its inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The *IPS* dimensions of commercial pipe universally used at the present time conform to the recommendations made by a Committee of the *A.S.M.E.* in 1886. Pipe up to 12 in. in diameter is made in certain definite sizes designated by nominal internal diameter which is somewhat different from the actual internal diameter, depending on the wall thickness required. There are three weights of wrought iron and steel pipe commonly used, known as *standard-weight*, *extra-strong*, and *double extra-strong*. Because of the necessity of maintaining the same external diameter in all three weights for the same nominal size, the added wall thickness is obtained by decreasing the internal diameter. The term *full-weight*, when applied to sizes below 8 in., means that the pipe is up to the nominal weight per foot. When applied to sizes between 8 and 12 in., inclusive, it often indicates that the pipe has the heaviest of several wall

thicknesses listed. In sizes 14 in. and upward, pipe is designated by its outside diameter (O.D.) and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by the *standard-weight* and *extra-strong* pipe, demands for pipe for higher pressures and temperatures in industry resulted in the use of a multiplicity of wall thicknesses for all sizes. Even in heating installations, the erection of piping by welding was deemed to

TABLE 1. DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE

Nominal Pipe Size	Outside Diam.	Nominal Wall Thicknesses for Schedule Numbers									
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160
1/8	0.405				0.068*		0.095*				
1/4	0.540				0.088*		0.119*				
3/8	0.675				0.091*		0.126*				
1/2	0.840				0.109*		0.147*				0.187
3/4	1.050				0.113*		0.154*				0.218
1	1.315				0.133*		0.179*				0.250
1¼	1.660				0.140*		0.191*				0.250
1½	1.900				0.145*		0.200*				0.281
2	2.375				0.154*		0.218*				0.343
2½	2.875				0.203*		0.276*				0.375
3	3.500				0.216*		0.300*				0.437
3½	4.000				0.226*		0.318*				
4	4.500				0.237*		0.337*		0.437		0.531
5	5.563				0.258*		0.375*		0.500		0.625
6	6.625				0.280*		0.432*		0.562		0.718
8	8.625		0.250	0.277*	0.322*	0.406	0.500*	0.593	0.718	0.812	0.906
10	10.75		0.250	0.307*	0.365*	0.500*	0.593	0.718	0.843	1.000	1.125
12	12.75		0.250	0.330*	0.406†	0.562†	0.687	0.843	1.000	1.125	1.312
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 O. D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 O. D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 O. D.	30.0	0.312	0.500	0.625							

All dimensions are given in inches.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions. For tolerances on wall thicknesses, see appropriate material specification.

*Thicknesses marked with asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

†Owing to a necessary departure from the old *standard-weight* and *extra-strong* thicknesses in these two sizes, the new thicknesses are not as yet stocked by all manufacturers and jobbers, thence, where agreeable to the purchaser and suitable for the service conditions, the old *standard-weight* 0.375 in. wall pipe corresponding to a 1000 P/S value of 37.7 is still available and can be substituted for the 0.406 in. wall, and the old *extra-strong* 0.500 in. wall pipe corresponding to a 1000 P/S value of 55 can be substituted for the 0.562 in. wall.

warrant the use of pipe lighter than standard weight. For these reasons, a *Sectional Committee on Standardization of Wrought Iron and Wrought Steel Pipe and Tubing* functioning under the procedure of the *American Standards Association* was appointed to standardize the dimensions and materials of pipe.

The proposed pipe standard recommended by that sectional committee has set up several schedules of pipe including standard-weight and extra-strong thicknesses which are now included in Schedules 40 and 80, re-

spectively. The schedules approved by the Sectional Committee are given in Tables 1 and 3 and the corresponding weights in Tables 2 and 4.

Standard-weight pipe is generally furnished with threaded ends in random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent of the total number of lengths ordered may be *jointers* which are two pieces coupled together. Extra-strong pipe is generally furnished with plain ends in

Table 2. Nominal Weights of Welded and Seamless Steel Pipe

Nominal Pipe Size Inches	Sched. 10 Plain Ends	Sched. 20 Plain Ends	Schedule 30		Schedule 40		Sched. 60 Plain Ends	Sched. 80 Plain Ends	Sched. 100 Plain Ends	Sched. 120 Plain Ends	Sched. 140 Plain Ends	Sched. 160 Plain Ends
			Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings						
1/8					0.25*	0.25*		0.32*				
1/4					0.43*	0.43*		0.54*				
3/8					0.57*	0.57*		0.74*				
1/2					0.86*	0.86*		1.09*				1.31
3/4					1.14*	1.14*		1.48*				1.94
1					1.68*	1.69*		2.18*				2.85
1 1/4					2.28*	2.29*		3.00*				3.77
1 1/2					2.72*	2.74*		3.64*				4.86
2					3.66*	3.68*		5.03*				7.45
2 1/2					5.80*	5.82*		7.67*				10.0
3					7.58*	7.62*		10.3*				14.3
3 1/2					9.11*	9.21*		12.5*				
4					10.8*	10.9*		15.0*		19.0		22.6
5					14.7*	14.9*		20.8*		27.1		33.0
6					19.0*	19.2*		28.6*		36.4		45.3
8		22.4	24.7*	25.0*	28.6*	28.8*	35.7	43.4*	50.9	60.7	67.8	74.7
10		28.1	34.3*	35.0*	40.5*	41.2*	54.8*	64.4	77.0	89.2	105.0	116.0
12		33.4	43.8*	45.0*	53.6	55.0	73.2	88.6	108.0	126.0	140.0	161.0
14 O. D.	36.8	45.7	54.6		63.3		85.0	107.0	131.0	147.0	171.0	190.0
16 O. D.	42.1	52.3	62.6		82.8		108.0	137.0	165.0	193.0	224.0	241.0
18 O. D.	47.4	59.0	82.0		105.0		133.0	171.0	208.0	239.0	275.0	304.0
20 O. D.	52.8	78.6	105.0		123.0		167.0	209.0	251.0	297.0	342.0	374.0
24 O. D.	63.5	94.7	141.0		171.0		231.0	297.0	361.0	416.0	484.0	536.0
30 O. D.	99.0	158.0	197.0									

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*The weights marked with asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

random lengths of 12 to 22 ft, although 5 per cent may be in lengths of 6 to 12 ft.

In addition to *IPS* copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tubing intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S. Government and the *American Society for Testing Materials*. There are three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Class *K*—Designed for underground services and general plumbing service.
Class *L*—Designed for general plumbing purposes.
Class *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and Types *L* and *M* where such conditions may be considered normal as, for instance, in heating work. Types *K* and *L* are available in both hard and

TABLE 3. DIMENSIONS OF WELDED WROUGHT-IRON PIPE

Nominal Pipe Size	Outside Diameter	Nominal Wall Thicknesses for Schedule Numbers					
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80
1/8	0.405	....	....	....	0.070*	....	0.098*
1/4	0.540	....	....	....	0.090*	....	0.122*
3/8	0.675	....	....	....	0.093*	....	0.129*
1/2	0.840	....	....	....	0.111*	....	0.151*
3/4	1.050	....	....	....	0.115*	....	0.157*
1	1.315	....	....	....	0.136*	....	0.183*
1 1/4	1.660	....	....	....	0.143*	....	0.195*
1 1/2	1.900	....	....	....	0.148*	....	0.204*
2	2.375	....	....	....	0.158*	....	0.223*
2 1/2	2.875	....	....	....	0.208*	....	0.282*
3	3.5	....	....	....	0.221*	....	0.306*
3 1/2	4.0	....	....	....	0.231*	....	0.325*
4	4.5	....	....	....	0.242*	....	0.344*
5	5.563	....	....	....	0.263*	....	0.383*
6	6.625	....	....	....	0.286*	....	0.441*
8	8.625	....	....	0.283*	0.329*	....	0.510*
10	10.75	....	....	0.313*	0.372*	0.510*	0.606
12	12.75	....	....	0.336*	0.414†	0.574†	0.702
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.625	0.750
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.687	....
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.750	....
20 O. D.	20.0	....	0.375	0.500	0.562	....	....

All dimensions are given in inches.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions. For tolerances on wall thicknesses, see appropriate material specification.

*Thicknesses marked with an asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

†Owing to a necessary departure from the old *standard-weight* and *extra-strong* thicknesses in these two sizes, the new thicknesses are not as yet stocked by all manufacturers and jobbers. Hence, where agreeable to the purchaser and suitable for the service conditions, the old *standard-weight* 0.382 in. wall pipe corresponding to a 1000 P/S value of 38.7 is still available and can be substituted for the 0.414 in. wall and the old *extra-strong* 0.510 in. wall pipe corresponding to a 1000 P/S value of 56.3 can be substituted for the 0.574 in. wall.

soft tempers; Type *M* is available only in hard temper. Where flexibility is essential as in hidden replacement work or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. New or exposed work generally employs copper pipe of a hard temper. All three classes are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall thickness tolerances for these classes of copper tubing are given in Table 5. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment

also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 F or more above room temperature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the

TABLE 4. NOMINAL WEIGHTS OF WELDED WROUGHT-IRON PIPE

Nominal Pipe Size (Inches)	Sched. 10	Sched. 20	Schedule 30		Schedule 40		Schedule 60	Schedule 80
	Plain Ends	Plain Ends	Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings	Plain Ends	Plain Ends
1/8	....	....	....	....	0.25*	0.25*	....	0.32*
1/4	....	....	....	....	0.43*	0.43*	....	0.54*
3/8	....	....	....	....	0.57*	0.57*	....	0.74*
1/2	....	....	....	....	0.86*	0.86*	....	1.09*
3/4	....	....	....	....	1.14*	1.14*	....	1.48*
1	....	....	....	....	1.68*	1.69*	....	2.18*
1 1/4	....	....	....	....	2.28*	2.29*	....	3.00*
1 1/2	....	....	....	....	2.72*	2.74*	....	3.64*
2	....	....	....	....	3.66*	3.68*	....	5.03*
2 1/2	....	....	....	....	5.80*	5.82*	....	7.67*
3	....	....	....	....	7.58*	7.62*	....	10.3*
3 1/2	....	....	....	....	9.11*	9.21*	....	12.5*
4	....	....	....	....	10.8*	10.9*	....	15.0*
5	....	....	....	....	14.7*	14.9*	....	20.8*
6	....	....	....	....	19.0*	19.2*	....	28.6*
8	....	....	24.7*	25.0*	28.6*	28.8*	....	43.4*
10	....	....	34.3*	35.0*	40.5*	41.2*	54.8*	54.4
12	....	....	43.8*	45.0*	53.6	55.0	73.2	88.6
14 O. D.	36.0	44.8	53.6	....	62.2	....	87.6	104.0
16 O. D.	41.3	51.4	61.4	....	81.2	....	111.0	....
18 O. D.	46.5	57.9	80.5	....	103.0	....	136.0	....
20 O. D.	....	77.0	103.0	....	115.0	....	....	....

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*Weights marked with an asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

case of refrigeration lines) per unit length of material per degree change in temperature is termed the *coefficient of linear expansion* of that material, or commonly, the *coefficient of expansion.* This coefficient varies with the material.

The linear expansion of cast-iron, steel, wrought iron, and copper pipe, the materials most frequently used in heating and ventilating work, can be determined from Table 6.

The elongation values in Table 6 were computed from the following formula:

$$L_t = L_0 \left[1 + a\left(\frac{t - 32}{1000}\right) + b\left(\frac{t - 32}{1000}\right)^2\right] \tag{1}$$

where

L_t = length at temperature t degrees Fahrenheit, feet.
L_0 = length at 32 F, feet.
t = final temperature, degrees Fahrenheit.
a and b are constants as given on the next page.

Metal	a	b
Cast-Iron	0.005441	0.001747
Steel	0.006212	0.001623
Wrought Iron	0.006503	0.001622
Copper	0.009278	0.001244

The three methods by which the elongation due to thermal expansion may be taken care of are:

1. Expansion joints.
2. Swivel joints.
3. Inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

TABLE 5. STANDARD DIMENSIONS, WEIGHTS, AND DIAMETER AND WALL THICKNESS TOLERANCES FOR COPPER WATER TUBES*

(All Tolerances Plus and Minus)

Nominal Size, In.	Actual Outside Diameter, In.	Permissible Variation in Mean Outside Diameter, In.		Wall Thickness, In.						Weight per Ft Lb		
				Class *K*		Class *L*		Class *M*				
		Annealed	Hard Drawn	Nominal	Permissible Variation	Nominal	Permissible Variation	Nominal	Permissible Variation	Class *K*	Class *L*	Class *M*
⅜	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.144
½	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.203
¾	0.875	0.003	0.001	0.065	0.0045	0.045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0.0015	0.065	0.0045	0.050	0.004	0.035	0.0035	0.839	0.655	0.464
1¼	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0.884	0.681
1½	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.94
2	2.125	0.005	0.002	0.083	0.005	0.070	0.005	0.058	0.0045	2.06	1.75	1.46
2½	2.625	0.005	0.002	0.095	0.005	0.080	0.005	0.065	0.0045	2.92	2.48	2.03
3	3.125	0.005	0.002	0.109	0.005	0.090	0.005	0.072	0.0045	4.00	3.33	2.68
3½	3.625	0.005	0.002	0.120	0.005	0.100	0.005	0.083	0.005	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.006	0.110	0.005	0.095	0.005	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.006	0.125	0.006	0.109	0.005	9.67	7.61	6.65
6	6.125	0.005	0.002	0.192	0.006	0.140	0.006	0.122	0.005	13.87	10.20	8.91

*From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials, A.S.T.M.* Designation B88-33.

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used extensively in low-pressure steam and hot water heating systems and in hot water supply lines. The swivel joints absorb the expansive movement of the pipe by the turning of threaded joints. In many cases the straight pipe in the offset of a swivel joint is sufficiently flexible to take up the expansion without developing enough thrust to produce swiveling in the threaded joint. This is preferable since continued turning in the threaded joint may in time result in a leak, particularly when the pressure is high. The amount of elongation which a swivel joint can take up is controlled by the length of the swing piece employed and by the lateral displacement which is permissible in the long pipe runs.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion is relatively complicated[1]. The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating work.

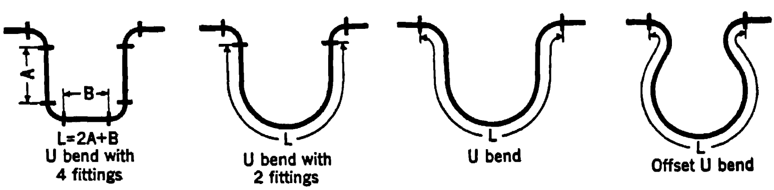


FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

Fig. 1 shows several types of expansion bends commonly used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from the following formula:

$$L = 6.16\sqrt{D\,\Delta} \tag{2}$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches.

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 lb per square inch. When square type bends are used, the width of the bend should not exceed about two times the height. It is

[1]Piping Handbook, by Walker and Crocker, and A Manual for the Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S. Smith, and E. T. Cope, published by The Detroit Edison Company.

further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter. Use of welding elbows with radii of 1½ times the pipe diameter will decrease the end thrusts somewhat but will raise the fiber stress correspondingly.

All risers must be anchored and safeguarded so that the difference in length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

Proper anchoring of piping is especially necessary with light-weight radiators, to allow for freedom of expansion in order that no pipe strain

TABLE 6. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 ft from −20 F up				Saturated Steam		Elongation in Inches per 100 ft from −20 F up			
Vacuum Inches of Hg.	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe
		−20	0	0	0	0	664.3	500	3.847	4.296	4.477	6.110
		0	0.127	0.145	0.152	0.204	795.3	520	4.020	4.487	4.677	6.352
		20	0.255	0.293	0.306	0.442	945.3	540	4.190	4.670	4.866	6.614
		40	0.390	0.430	0.465	0.655	1115.3	560	4.365	4.860	5.057	6.850
29.39	....	60	0.518	0.593	0.620	0.888	1308.3	580	4.541	5.051	5.268	7.123
28.89	....	80	0.649	0.725	0.780	1.100	1525.3	600	4.725	5.247	5.455	7.388
27.99	....	100	0.787	0.898	0.939	1.338	1768.3	620	4.896	5.437	5.660	7.636
26.48	....	120	0.926	1.055	1.110	1.570	2041.3	640	5.082	5.627	5.850	7.893
24.04	....	140	1.051	1.209	1.265	1.794	2346.3	660	5.260	5.831	6.067	8.153
20.27	....	160	1.200	1.368	1.427	2.008	2705	680	5.442	6.020	6.260	8.400
14.63	....	180	1.345	1.528	1.597	2.255	3080	700	5.629	6.229	6.481	8.676
6.45	....	200	1.495	1.691	1.778	2.500		720	5.808	6.425	6.673	8.912
	2.5	220	1.634	1.852	1.936	2.720		740	6.006	6.635	6.899	9.203
	10.3	240	1.780	2.020	2.110	2.960		760	6.200	6.833	7.100	9.460
	20.7	260	1.931	2.183	2.279	3.189		780	6.389	7.046	7.314	9.736
	34.5	280	2.085	2.350	2.465	3.422		800	6.587	7.250	7.508	9.992
	52.3	300	2.233	2.519	2.630	3.665		820	6.779	7.464	7.757	10.272
	74.9	320	2.395	2.690	2.800	3.900		840	6.970	7.662	7.952	10.512
	103.3	340	2.543	2.862	2.988	4.145		860	7.176	7.888	8.195	10.814
	138.3	360	2.700	3.029	3.175	4.380		880	7.375	8.098	8.400	11.175
	180.9	380	2.859	3.211	3.350	4.628		900	7.579	8.313	8.639	11.360
	232.4	400	3.008	3.375	3.521	4.870		920	7.795	8.545	8.867	11.625
	293.7	420	3.182	3.566	3.720	5.118		940	7.989	8.755	9.089	11.911
	366.1	440	3.345	3.740	3.900	5.358		960	8.200	8.975	9.300	12.180
	451.3	460	3.511	3.929	4.096	5.612		980	8.406	9.196	9.547	12.473
	550.3	480	3.683	4.100	4.280	5.855		1000	8.617	9.421	9.776	12.747

[a]From *Piping Handbook*, by Walker and Crocker. This table gives the expansion from −20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

will distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters they usually emit sounds of distress which are exceedingly troublesome.

PIPE THREADS

All threaded pipe for heating and ventilating installations uses the American Standard taper pipe thread which is made with a taper of 1 in

16 measured on the diameter of the pipe so as to secure a tight joint. Threads of fittings are tapped to the same taper. The number of threads per inch varies with the different pipe sizes. All threaded pipe should be made up with a thread paste suitable for the service under which the pipe is to be used.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support when expansion takes place, or the bottom support should be designed to withstand the entire load.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this malleable iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure.

As in the case of pipe, there are several weights of fittings manufactured. Recognized American Standards for the various weights are as follows:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure.

Steel flanged fittings for 150 and 300 lb maximum steam service pressure.

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings, usually of wrought iron only; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close

or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared, or compression types. Illustrations of each of these types are shown in Fig. 2. Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in both large and small sizes.

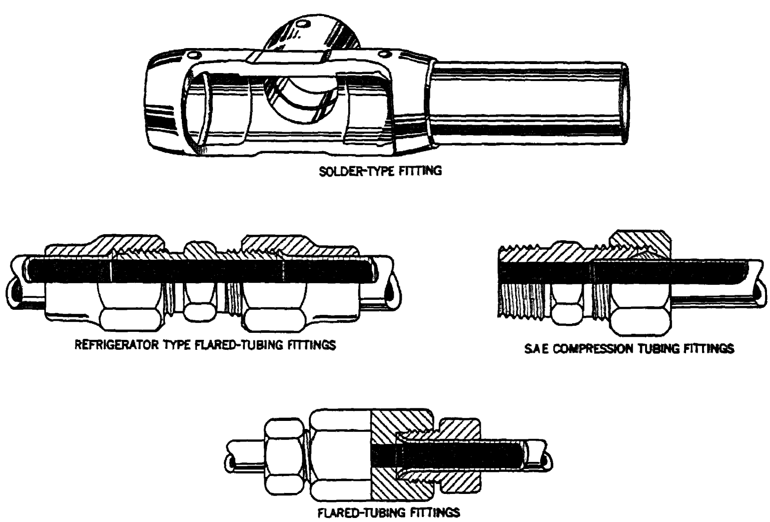


FIG. 2. COPPER OR BRASS TUBING FITTINGS

An American Standard, *ASA* A40.2-1936 has been prepared to standardize dimensions for brass fittings for flared copper water tubes. Flared tube fittings are widely used in refrigerating work where *SAE* dimensions and a 45-deg flare render most fittings interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits. Brass fittings with *SAE* dimensions are not interchangeable with the American Standard fittings for water tubes.

Ammonia pipe fittings made of cast-iron were formerly used extensively in handling refrigerants in larger installations. Replacement of ammonia by other refrigerants operating at lower pressures has seriously curtailed the market for these fittings, and the use of copper tubing with special fittings and welded steel piping has further rendered ammonia fittings obsolete. For these reasons formulation of an American Standard for these fittings was abandoned by the *ASA* in 1936.

Thread Connections

Threads used for fittings are the same American Standard taper pipe threads as those used for pipe, and unless otherwise ordered, right-hand threads are used. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are known to the trade as pitched elbows and are commercially available. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively.

Flanged fittings are generally used in the best practice for connecting all piping above 4 in. in diameter. While screwed fittings may be used for the larger sizes and are satisfactory under the proper working conditions, it will be found difficult either to make or to break the joints in these large sizes.

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 lb pressure and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a $\frac{1}{16}$-in. raised face. The standard facing for steel flanged fittings for 150 and 300 lb is a $\frac{1}{16}$-in. raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 7, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Tables 8 and 9.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a relatively narrow recessed facing.

WELDING

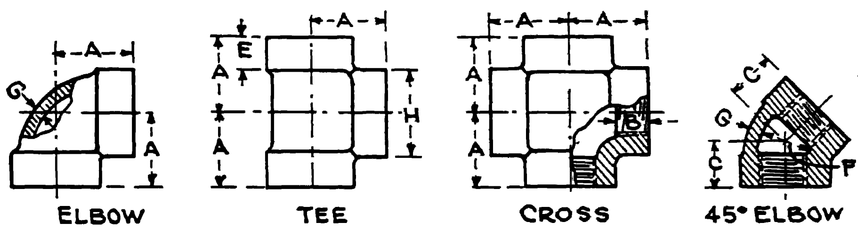
Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as a competitive method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding on low to medium pressure heating jobs.

Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding

embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds.

Welding application requires the same basic knowledge of design as do the other types of assembly, but in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to

TABLE 7. TENTATIVE AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45-DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR 125-LB CAST-IRON SCREWED FITTINGS



Nominal Pipe Size	*A* Center to End, Elbows, Tees and Crosses	*C* Center to End, 45 Deg Elbows	*B* Length of Thread Min.	*E* Width of Band, Min.	*F* Inside Diameter of Fitting Min.	*F* Inside Diameter of Fitting Max.	*G* Metal Thickness, Min.	*H* Outside Diameter of Band, Min.
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50	5.97	1.88	1.88	12.750	12.850	0.800	15.47
14 O.D.	10.40	....	2.00	2.00	14.000	14.100	0.880	16.94
16 O.D.	11.82	....	2.20	2.20	16.000	16.100	1.000	19.30

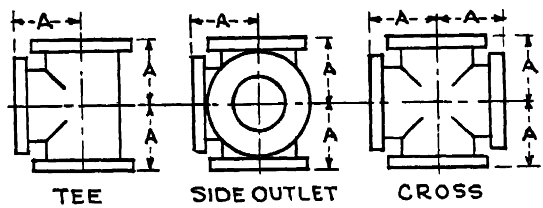
All dimensions given in inches.

determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests. Industry will not accept the employment of mechanics of undetermined ability nor on the basis of past experience. Neither does industry accept

the statement that a weld is only as good as the workman who makes it. The control Codes now in process of adoption will be the law governing the use of the welding process. These Codes prohibit individual practices contrary to their specified procedure and rules of control, and this is predicated upon the sound requirement that the employer must assume full responsibility for the deposited weld.

It is advisable that this management responsibility be included in all welding specifications and that authoritative standards of workmanship also be specified. The standards of workmanship for this industry are as

Table 8. American Standard Dimensions of Tees and Crosses (Straight Sizes) for Class 125 Cast-Iron Flanged Fittings



Nominal Pipe Size a-b	A	AA	Diameter of Flange	Thickness of Flange, Min.	Metal[d] Thickness of Body
	Center to Face Tees and Crosses b-c	Face to Face Tees and Crosses b-c			
1	3½	7	4¼	7/16	5/16
1¼	3¾	7½	4⅝	½	5/16
1½	4	8	5	9/16	5/16
2	4½	9	6	⅝	5/16
2½	5	10	7	11/16	5/16
3	5½	11	7½	¾	⅜
3½	6	12	8½	13/16	7/16
4	6½	13	9	15/16	½
5	7½	15	10	15/16	½
6	8	16	11	1	9/16
8	9	18	13½	1⅛	⅝
10	11	22	16	1 3/16	¾
12	12	24	19	1¼	13/16
14 O.D.	14	28	21	1⅜	⅞
16 O.D.	15	30	23½	1 7/16	1
18 O.D.	16½	33	25	1 9/16	1 1/16
20 O.D.	18	36	27½	1 11/16	1⅛
24 O.D.	22	44	32	1⅞	1¼
30 O.D.	25	50	38¾	2⅛	1 7/16
36 O.D.	28	56	46	2⅜	1⅝
42 O.D.	31	62	53	2⅝	1 13/16
48 O.D.	34	68	59½	2¾	2

All dimensions given in inches.

[a]Size of all fittings listed indicates nominal inside diameter of port.

[b]Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

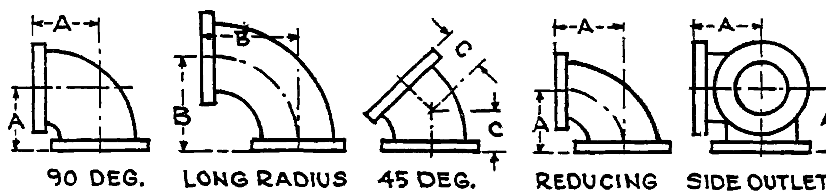
[c]Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

[d]Body thickness at no point shall be less than 87½ per cent of the dimensions given in the table.

set forth in the *Standard Manual on Pipe Welding* of the *Heating, Piping and Air Conditioning Contractors National Association.*

A complete line of manufactured steel welding fittings is now available and a dimensional standard is being prepared under the procedure of the *American Standards Association* to unify heretofore divergent dimensions for the same type welding fittings as produced by different manufacturers. Proposed standard dimensions for elbows, tees, caps, and lapped-joint stub ends are given in Table 10. Dimensions for eccentric and concentric reducers, and 180-deg return bends are not shown in Table 10 but will

TABLE 9. AMERICAN STANDARD DIMENSIONS OF ELBOWS FOR CLASS 125 CAST-IRON FLANGED FITTINGS



Nominal Pipe Size[a]	A	B	C	Diameter of Flange	Thickness of Flange, Min.	Metal[e] Thickness of Body
	Center to Face Elbow[b-c-d]	Center to Face Long Radius Elbow[b-c-d]	Center to Face 45 Deg Elbow[c]			
1	3½	5	1¾	4¼	7/16	5/16
1¼	3¾	5½	2	4⅝	½	5/16
1½	4	6	2¼	5	9/16	5/16
2	4½	6½	2½	6	⅝	5/16
2½	5	7	3	7	11/16	5/16
3	5½	7¾	3	7½	¾	⅜
3½	6	8½	3½	8½	13/16	7/16
4	6½	9	4	9	15/16	½
5	7½	10¼	4½	10	15/16	½
6	8	11½	5	11	1	9/16
8	9	14	5½	13½	1⅛	⅝
10	11	16½	6½	16	1 3/16	¾
12	12	19	7½	19	1¼	13/16
14 O.D.	14	21½	7½	21	1⅜	⅞
16 O.D.	15	24	8	23½	1 7/16	1
18 O.D.	16½	26½	8½	25	1 9/16	1 1/16
20 O.D.	18	29	9½	27½	1 11/16	1⅛
24 O.D.	22	34	11	32	1⅞	1¼
30 O.D.	25	41½	15	38¾	2⅛	1 7/16
36 O.D.	28	49	18	46	2⅜	1⅝
42 O.D.	31	56½	21	53	2⅝	1 13/16
48 O.D.	34	64	24	59½	2¾	2

All dimensions given in inches.

[a] Size of all fittings listed indicates nominal inside diameter of port.

[b] Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[c] Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45-deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90-deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

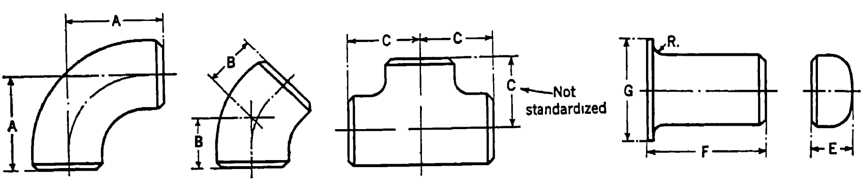
[d] Side outlet elbows shall have all openings on intersecting center-lines.

[e] Body thickness at no point shall be less than 87½ per cent of the dimensions given in the table.

be included in the American Standard. Larger sizes also are available in some types of fittings. The welding bevel which is a straight 37½-deg V for wall thicknesses ¾ in. and below, and a U-bevel for thicknesses heavier than ¾ in., conforms to the recommended practice of *ASA* Standard B16e-1939, American Standard for Steel Pipe Flanges and Flanged Fittings. The latter also contains dimensions for steel welding-neck flanges for pressures up to 2500 lb per square inch. Tables 11 and 12 give these dimensions for welding-neck flanges suitable for 150 and 300 lb per square inch gage pressure.

Socket welding fittings are also commercially available. These fittings have a machined recess into which the pipe slips. A fillet weld between the pipe and socket edge provides a pressure-tight joint. This type of fitting has gained rapid acceptance due to its ease of installation, low cost, and ability to make a pressure tight joint without weakening the pipe as is the case with threading. Standard dimensions for socket welding fittings are being formulated under the procedure of the *American Standards Association.*

Table 10. Proposed American Standard Dimensions for Butt-Welding Elbows, Tees, Caps, and Lapped-Joint Stub Ends



Nominal Pipe Size	Outside Diameter	Center-to-End			Caps E[b-c]	Lapped-Joint Stub Ends		
		90-Deg Elbows A	45-Deg Elbows B	Of Run Tee C[a]		Length F[b]	Radius of Fillet R	Diam. of Lap G[d]
1	1.315	1½	⅞	1½	1½	4	⅛	2
1¼	1.660	1⅞	1	1⅞	1½	4	3/16	2½
1½	1.900	2¼	1⅛	2¼	1½	4	¼	2⅞
2	2.375	3	1⅜	2½	1½	6	5/16	3⅝
2½	2.875	3¾	1¾	3	1½	6	5/16	4⅛
3	3.500	4½	2	3⅜	2	6	⅜	5
3½	4.000	5¼	2¼	3¾	2½	6	⅜	5½
4	4.500	6	2½	4⅛	2½	6	7/16	6 3/16
5	5.563	7½	3⅛	4⅞	3	8	7/16	7 5/16
6	6.625	9	3¾	5⅝	3½	8	½	8½
8	8.625	12	5	7	4	8	½	10⅝
10	10.750	15	6¼	8½	5	10	½	12¾
12	12.750	18	7½	10	6	10	½	15

All dimensions given in inches.

[a]The dimensions of welding tees cover those which have side outlets from one size less than half the size of the run-way opening of the tees to full size.

[b]Dimensions E and F are applicable only to these fittings in schedules up to and including Schedule 80, *ASA* Standard B36.10-1939.

[c]The shape of these caps shall be ellipsoidal and shall conform to the requirements of the *ASME* Boiler Construction Code.

[d]This dimension is for a regular lapped joint in accordance with *ASA* Standard B16e-1939. For ring-joint facing dimensions, see B16e-1939.

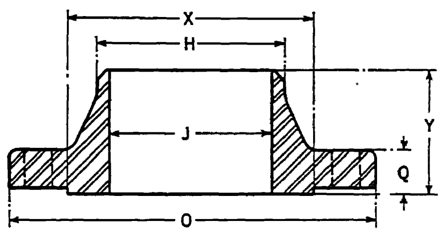
VALVES

Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger sizes either cast-iron, cast-steel or some of the steel alloys are employed. Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up

TABLE 11. AMERICAN STANDARD DIMENSIONS OF STEEL WELDING NECK FLANGES FOR STEAM SERVICE PRESSURE RATING OF 150 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 500 F, AND 100 LB PER SQ IN. (GAGE) AT 750 F



Nominal Pipe Size	Diameter of Flange	Thickness of of Flg.[a] Min.	Diameter of Hub	Hub Diam. Beginning of Chamfer[b-c]	Length Thru Hub[a]	Inside Diam. of Pipe Schedule 40[c]	Diam. of Bolt Circle	No. of Bolts	Size of Bolts
	O	*Q*	*X*	*H*	*Y*	*J*			
½	3½	7/16	1 3/16	0.84	1⅞	0.62*	2⅜	4	½
¾	3⅞	½	1½	1.05	2 1/16	0.82*	2¾	4	½
1	4¼	9/16	1 15/16	1.32	2 3/16	1.05*	3⅛	4	½
1¼	4⅝	⅝	2 5/16	1.66	2¼	1.38*	3½	4	½
1½	5	11/16	2 9/16	1.90	2 7/16	1.61*	3⅞	4	½
2	6	¾	3 1/16	2.38	2½	2.07*	4¾	4	⅝
2½	7	⅞	3 9/16	2.88	2¾	2.47*	5½	4	⅝
3	7½	15/16	4¼	3.50	2¾	3.07*	6	4	⅝
3½	8½	15/16	4 13/16	4.00	2 13/16	3.55*	7	8	⅝
4	9	15/16	5 5/16	4.50	3	4.03*	7½	8	⅝
5	10	15/16	6 7/16	5.56	3½	5.05*	8½	8	¾
6	11	1	7 9/16	6.63	3½	6.07*	9½	8	¾
8	13½	1⅛	9 11/16	8.63	4	7.98*	11¾	8	¾
10	16	1 3/16	12	10.75	4	10.02*	14¼	12	⅞
12	19	1¼	14⅜	12.75	4½	....	17	12	⅞
14 OD	21	1⅜	15¾	14.00	5	To Be	18¾	12	1
16 OD	23½	1 7/16	18	16.00	5	Specified	21¼	16	1
18 OD	25	1 9/16	19⅞	18.00	5½	by	22¾	16	1⅛
20 OD	27½	1 11/16	22	20.00	5 11/16	Purchaser	25	20	1⅛
24 OD	32	1⅞	26⅛	24.00	6		29½	20	1¼

All dimensions given in inches.

[a]A raised face of 1/16 in. is included in *thickness of flange minimum* and in *length through hub.*

[b]The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

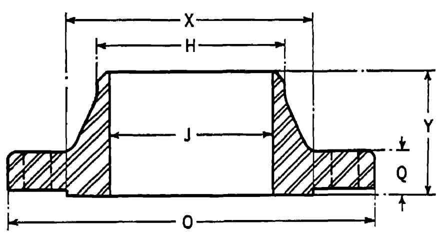
[c]Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in *ASA* B36.10-1939, Schedule 40.

*These diameters are identical with the diameters of what was formerly designated as *Standard Weight Pipe* of the corresponding sizes.

to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum. These valves may be

TABLE 12. AMERICAN STANDARD DIMENSIONS OF STEEL WELDING NECK FLANGES FOR STEAM SERVICE PRESSURE RATING OF 300 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 750 F



Nominal Pipe Size	Diam. of Flange	Thickness of Flg.[a] Min.	Diam. of Hub	Hub Diam. Beginning of Chamfer[b-c-d]	Length Thru Hub[a]	Inside Diam. of Pipe Schedule 40[c-d]	Inside Diam. of Pipe Schedule 80[c-d]	Diam. of Bolt Circle	No. of Bolts	Size of Bolts
	O	Q	X	H	Y	J	J			
½	3¾	9/16	1½	0.84	2 1/16	0.62*	0.55†	2⅝	4	½
¾	4⅝	⅝	1⅞	1.05	2¼	0.82*	0.74†	3¼	4	⅝
1	4⅞	11/16	2⅛	1.32	2 7/16	1.05*	0.96†	3½	4	⅝
1¼	5¼	¾	2½	1.66	2 9/16	1.38*	1.28†	3⅞	4	⅝
1½	6⅛	13/16	2¾	1.90	2 11/16	1.61*	1.50†	4½	4	¾
2	6½	⅞	3 5/16	2.38	2¾	2.07*	1.94†	5	8	⅝
2½	7½	1	3 15/16	2.88	3	2.47*	2.32†	5⅞	8	¾
3	8¼	1⅛	4⅝	3.50	3⅛	3.07*	2.90†	6⅝	8	¾
3½	9	1 3/16	5¼	4.00	3 3/16	3.55*	3.36†	7¼	8	¾
4	10	1¼	5¾	4.50	3⅜	4.03*	3.83†	7⅞	8	¾
5	11	1⅜	7	5.56	3⅞	5.05*	4.81†	9¼	8	¾
6	12½	1 7/16	8⅛	6.63	3⅞	6.07*	5.76†	10⅝	12	¾
8	15	1⅝	10¼	8.63	4⅜	7.98*	7.63†	13	12	⅞
10	17½	1⅞	12⅝	10.75	4⅝	10.02*	....	15¼	16	1
12	20½	2	14¾	12.75	5⅛	....	....	17¾	16	1⅛
14 OD	23	2⅛	16¾	14.00	5⅝	To Be	To Be	20¼	20	1⅛
16 OD	25½	2¼	19	16.00	5¾	Speci-	Speci-	22½	20	1¼
18 OD	28	2⅜	21	18.00	6¼	fied by	fied by	24¾	24	1¼
20 OD	30½	2½	23⅛	20.00	6⅜	Pur-	Pur-	27	24	1¼
24 OD	36	2¾	27⅝	24.00	6⅝	chaser	chaser	32	24	1½

All dimensions given in inches.

[a]A raised face of 1/16 in. is included in *thickness of flange minimum* and in *length through hub.*

[b]This outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg

[c]Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in *ASA* B36.10-1939, Schedules 40 and 80. Purchaser's order must specify which of these two inside diameters is desired.

[d]These flanges are regularly bored to match inside diameter of Schedule 40 pipe, but are bored to Schedule 80 pipe when so ordered.

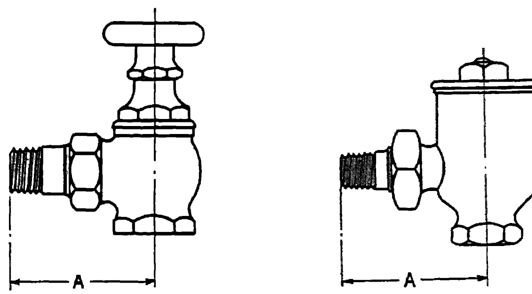
*These diameters are identical with the diameters of what was formerly designated as *Standard Weight Pipe* of the corresponding sizes.

†These diameters are identical with the diameters of what was formerly designated as *Extra-Strong Pipe* of the corresponding sizes.

secured with either a rising or a non-rising stem, although in the smaller sizes the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between

TABLE 13. STANDARD ROUGHING-IN DIMENSIONS ANGLE TYPE VALVES



SIZE OF VALVE	DIMENSION A STEAM AND HOT WATER ANGLE VALVES AND UNION ELBOWS EFFECTIVE JANUARY 1, 1926	DIMENSION A MODULATING VALVES EFFECTIVE JANUARY 1, 1926	DIMENSION A RETURN LINE VACUUM VALVES EFFECTIVE JANUARY 1, 1925
½	2¼	2¾	3¼
¾	2¾	2¾	------
1	3	3	------
1¼	3½	3½	------
1½	3¾	3¾	------
2	4¼	4¼	------
Tolerance	±⅛	±⅛	------

All dimensions given in inches.

Connecting ends shall be threaded and gaged as to threading according to the American (Taper) Pipe Thread Standard, *A.S.A.* No. B2—1919.

The standardization of the Roughing-in Dimensions of Angle Steam and Hot Water, and Modulating Radiator Valves was made possible by the cooperation of the *Manufacturers Standardization Society of the Valves and Fittings Industry*.

the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever handles are often supplied to indicate the relative opening of the valve in any position. Standard roughing-in dimensions for angle-type valves are given in Table 13.

Automatic control of steam supply to individual radiators can be

effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a 1/16-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made particularly for use in hot water heating systems are of less complex design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when the pressure dies down and a vacuum tends to be formed the air is drawn back into the radiator.

A system operating either continuously or intermittently and supplied with vacuum valves will generally hold heat longer and warm up more quickly than one provided with non-vacuum air valves; thus, it will effect considerable economy of fuel because the idle period during which no heat is delivered is shortened. In those cases, where a system is equipped with vacuum air valves and which has been cold for several hours, the system will probably have an internal pressure within the radiator closely approaching atmospheric. At such times, the vacuum valve will not vent the system any more rapidly than the ordinary type. Automatic air valves are provided with a float to close them in case the radiator becomes flooded with water because it does not drain properly.

CORROSION[2]

Corrosion is sometimes encountered in heating work on the outside of buried pipes or the inside of steam heating systems; it is seldom experienced in hot water heating systems unless the water is frequently renewed. Piping buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford some protection.

Internal corrosion in steam heating systems occurs principally in the

[2]New Light on Heating System Corrosion, by J. H. Walker (*Heating and Ventilating*, May, 1933). A.S.H.V.E. RESEARCH REPORT No. 983—Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg. (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 253). A.S.H.V.E. RESEARCH REPORT No. 1037—Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 263). A.S.H.V.E. RESEARCH REPORT No. 1071—Corrosion Studies in Steam Heating Systems, by R. R. Seeber and Margaret R. Holley (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 461).

condensate return pipes and is nearly always caused by oxygen or carbon dioxide, or both, in solution in the condensate. Oxygen may enter the heating system with the steam, owing to its presence in the boiler-feed water, or it may enter as air through small leaks, particularly in systems which operate at sub-atmospheric pressures. When a steam heating system is operated intermittently, air rushes in during each shutdown period and oxygen is absorbed by the condensate which clings to the interior surfaces of the pipes and radiators. The rate of corrosion depends upon the amounts of oxygen and carbon dioxide present in solution, upon the operating temperature, and upon the length of time that the pipe surfaces are in contact with gas-laden condensate.

Another possible cause of corrosion is a flow of electric current sometimes resulting from faulty electrical circuits which should be corrected. Electrolytic corrosion also may occur because of the presence of two dissimilar metals, such as brass and iron, but the condensate in practically all steam heating systems is such a weak electrolyte that this cause of corrosion is very infrequent.

If trouble is experienced from corrosion, oxygen should be eliminated from the feed water by proper deaeration with commercial apparatus. The elimination of the oxygen due to air leakage is more difficult because of the multitude of small leaks which exist around valve stems and in pipe joints. In vacuum systems, however, an attempt should be made to minimize such leakage.

Carbon dioxide in varying amounts is contained in steam produced from the majority of water supplies. It is formed from the breaking down of carbonates and bicarbonates which are present in nearly all natural waters. It can be partly removed by chemical treatment and deaeration, but there is no simple method whereby it can be entirely eliminated.

These gases cause corrosion only when in solution in the condensate; when they are mixed with dry steam their corrosive effect is negligible. The amount of gas in solution depends upon the partial pressure of that gas in the atmosphere above the surface of the solution, in accordance with the well known physical law of Henry and Dalton[3]. The exact application of this law, however, assumes equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

Distinction should be made between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam, such as water heaters, kitchen equipment, and sterilizers. Experience has shown that in heating systems the partial pressures of the gases do not reach such magnitudes as to cause harmful amounts of gas to become dissolved in the condensate when steam supplies are of reasonable purity. In other kinds of steam-using apparatus which are not ordinarily well vented, the gases tend to accumulate in the steam space and to become dissolved in the condensate in appreciable concentrations. Consequently, corrosion is frequently observed in the condensate discharge lines from such apparatus, but this does not necessarily indicate that equally serious

[3]Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 121).

corrosion is taking place in the heating system supplied with steam from the same source.

When corrosive conditions are believed to exist, their seriousness should be determined by actual measurement, rather than by inference from isolated instances of pipe failures. The *National District Heating Association* has perfected a corrosion tester for measuring the inherent corrosiveness of existing conditions. This corrosion tester consists of a frame supporting three coils of wire which are carefully weighed. After the tester has been inserted in the pipe line for a definite length of time, the loss of weight of the coils, referred to an established scale, indicates the relative corrosiveness of the condensate. Accompanying such corrosion measurements, a careful chemical analysis should be made of the condensate, and the findings will serve as a basis for an intelligent study of the problem.

Corrosion, if found to exist, can be lessened or overcome by several means. If the steam supply is found to be definitely contaminated, proper chemical treatment of the water, followed by deaeration, is an obvious remedy. The leaks in the piping system, particularly in vacuum systems, should be stopped so far as is practicable.

Some success has been reported with the use of inhibitors, chief among which are oil, and sodium silicate. Oil may be fed into the main steam-supply pipe by means of a sight-feed lubricator. The type of oil known as 600-W is usually recommended. In the present state of knowledge on this point, the quantity to be fed can best be determined by trial. The use of sodium silicate, fed in a similar manner, is reported to be successful but it has not been widely used.

In view of the fact that corrosion is most frequently found in the return lines from special equipment, which constitute a relatively small part of the total piping in a building, a simple solution of the corrosion problem may be to use non-corroding materials in those certain portions of the piping system, since the higher cost will usually be an unappreciable portion of the total. Brass and copper are undoubtedly less subject to this type of corrosion than the ferrous metals, and considerable attention is now being given to corrosion-resistant linings for ferrous pipe. Cast-iron pipe, sometimes alloyed with other metals, also deserves consideration.

Chapter 18

GRAVITY WARM AIR FURNACE SYSTEMS

Design Procedure, Estimating Heating Requirements, Leader Pipe Sizes, Proportioning Wall Stacks, Register Selections, Recirculating Ducts and Grilles, Furnace Return Connection, Furnace Capacity, Examples, Booster Fans

WARM air heating systems of the gravity type are described in this chapter[1], and those of the mechanical type are described in Chapter 19. In the gravity type, the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal or brick, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard.

The air supply to the furnace may be taken (1) entirely from inside the building through one or more recirculating ducts, (2) entirely from outside the building, in which case no air is recirculated, or (3) through a combination of the inside and the outside air supply systems.

DESIGN PROCEDURE

The design of a furnace heating system involves the determination of the following items:

1. Heat loss in Btu from each room in the building.
2. Area and diameter in inches of warm-air pipes in basement (known as leaders).
3. Area and dimensions in inches of vertical pipes (known as wall stacks).
4. Free and gross area and dimensions in inches of warm-air registers.
5. Area and dimensions of recirculating or outside air ducts, in inches.
6. Free and gross area and dimensions in inches of recirculating registers.

[1]All figures and much of the engineering data which follow are from University of Illinois, *Engineering Experiment Station Bulletins* Nos. 141, 188, 189 and 246; Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz, V. S. Day, and S. Konzo.

7. Size of furnace necessary to supply the warm air required to overcome the heat loss from the building. This *size* should include square inches of leader pipe area which the furnace must supply. It is also desirable to call for a minimum bottom fire-pot diameter in inches, which is the nominal grate diameter.

8. Area and dimensions in inches of chimney and smoke pipe. If an unlined chimney is to be used, that fact should be made clear.

The heat loss calculations should be made in accordance with the procedure outlined in Chapter 5, taking into consideration the transmission losses as well as the infiltration losses.

LEADER PIPE SIZES

In a gravity circulating warm-air furnace system the size of the leader to a given room depends upon the temperature of the warm air entering the room at the register. A reasonable air temperature at the registers must, therefore, be chosen before the system can be designed. The *National Warm Air Heating and Air Conditioning Association* has approved an air temperature of 175 F at the registers as satisfactory for design purposes. At this temperature, the heat-carrying capacity (heat available above 70 F) per square inch of leader pipe per hour for first, second or third floors is shown by Fig. 1 at 175 F to be 105, 170 and 208 Btu, respectively. For average calculations, the values 111, 166 and 200 will simplify the work and may be satisfactorily substituted for these heat-carrying capacities. If H represents the total heat to be supplied any room, the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{111} = \text{approximately } 0.009H \quad (1)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \quad (2)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{200} = \text{approximately } 0.005H \quad (3)$$

In designing for a lower warm-air register temperature, say 160 F, the factors 111, 166 and 200 become 80, 140 and 166 (Fig. 1 at 160 F), and the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{80} = \text{approximately } 0.012H \quad (4)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{140} = \text{approximately } 0.007H \quad (5)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \quad (6)$$

These equations are applicable to straight leaders from 6 to 8 ft in length. Longer leaders must be thoroughly insulated or the vertical stacks must be increased in area as discussed under wall stacks. If some provision is not made for these longer leaders, the air temperature may be much lower than anticipated and the room will not be properly heated.

The values shown in Fig. 1 apply only to the case where the straight, leader pipe is 8 ft in length and is connected to stacks whose cross-sectional area is approximately 75 per cent of that of the leader pipe.

Any deviation from these conditions requires a modification of the constants used in Equations 1, 2, and 3. The temperature drop in leaders of various lengths at three different register temperatures is shown in Fig. 2, and should be used to obtain new register temperatures, lower than 175 F, on which to base selections from the curves of Fig. 1, and thereby new constants for Equations 1, 2 and 3.

Leader sizes should in general be not less than those obtained by Equations 1 to 3 nor should leaders less than 8 in. in diameter be used. In residences requiring a leader pipe area of 650 sq in. or less, it is advisable

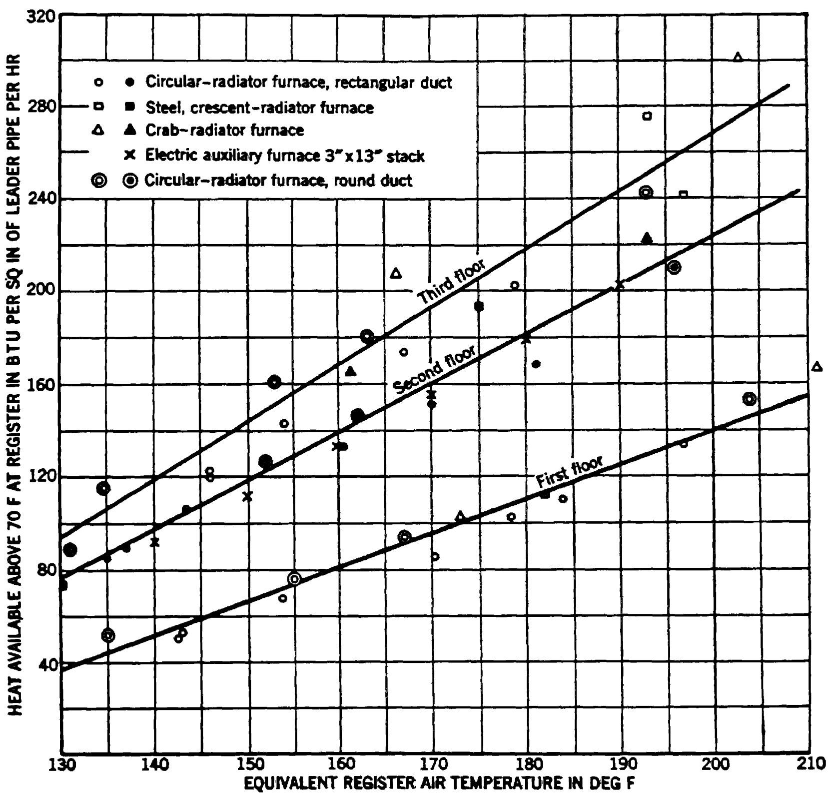


FIG. 1. VALUE OF SQUARE INCH OF LEADER PIPE AREA FOR FIRST, SECOND, AND THIRD FLOORS FOR SIMPLE SYSTEM HAVING LEADERS 8 FT IN LENGTH

to use two or more leader pipes to rooms requiring more than the capacity of a 12 in. round pipe. It is not considered good commercial practice to specify diameters except in whole inches. The tops of all leaders should be at the same elevation as they leave the furnace bonnet, and from this point there should be a uniform up-grade of 1 in. per foot of run in all cases. Leaders over 12 ft in length should be avoided if possible. In cases where such leaders are required, the use of a larger size pipe, than is required by the application of the equations, smooth transition fittings, and duct insulation are recommended.

PROPORTIONING WALL STACKS

The wall stack for an upper floor should be made not less than 70 per cent of the area of the leader. In cases where the leader is short and straight as was the case for Fig. 1, such a practice is probably justified, since the loss (Fig. 3) in capacity occasioned by the smaller stack is not serious for stacks having areas in excess of 70 per cent of the leader area. For leaders over 8 ft in length or for leaders which are not straight, the ratio of stack area to leader area should be greater than 70 per cent in

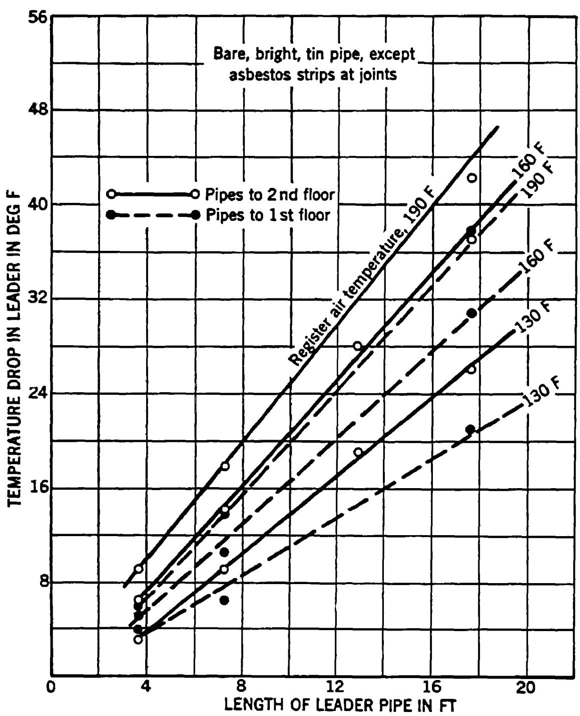


FIG. 2. INFLUENCE OF LEADER PIPE LENGTH ON TEMPERATURE LOSS IN AIR FLOWING THROUGH PIPE

order to offset the greater temperature losses (Fig. 2) in the longer leader. In gravity circulating systems, this ratio of stack to leader area is a very important matter.

The curves in Figs. 4 and 5 indicate that for rooms having a heat requirement exceeding approximately 9000 Btu per hour, exceedingly high register temperatures are required for stacks whose width is less than 3½ in. For such requirements either multiple stacks, or stacks having larger cross-sectional area (placed in 6 in. studding spaces) will be required.

REGISTER SELECTIONS

The registers used for discharging warm air into the rooms should have a free or net area not less than the area of the leader in the same run of piping. The free area should be at least 70 per cent of the gross area of the register. No upper-floor register should be wider horizontally than the wall stack, and it should be placed either in the baseboard or side wall, if this can be done without the use of offsets. First floor registers may be of the baseboard or floor type, with the former location preferred. High

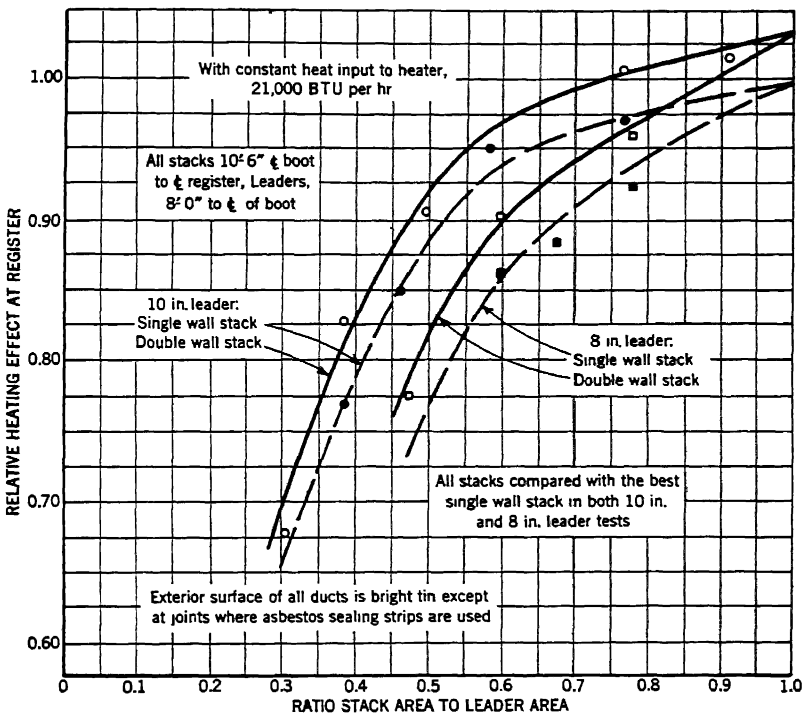


FIG. 3. RELATIVE HEATING EFFECT OF STACKS AT CONSTANT HEAT INPUT TO FURNACE

side wall locations for warm air registers in gravity circulating systems are not recommended on account of the tendency for stratification of the air in the room, resulting in high temperatures at the ceiling.

RECIRCULATING DUCTS AND GRILLES

The ducts through which air is returned to the furnace should be designed to minimize friction and turbulence. They should be of ample area, equal to or slightly in excess of the total area of warm-air pipes, and

at all points where the air stream must change direction or shape, streamline fittings should be employed. Horizontal ducts should pitch at least ½ in. per foot upward from the furnace.

The recirculating grilles (or registers) should have a free area at least equal to the ducts to which they connect, and their free area should never be less than 50 per cent of their gross area.

The location and number of return grilles will depend on the size, details and exposure of the house. Small compactly built houses may frequently be adequately served by a single return effectively placed in a central hall. More often it is desirable to have two or more returns, provided, however, that in two-story residences one return is placed to effectively receive the cold air returning by way of the stairs.

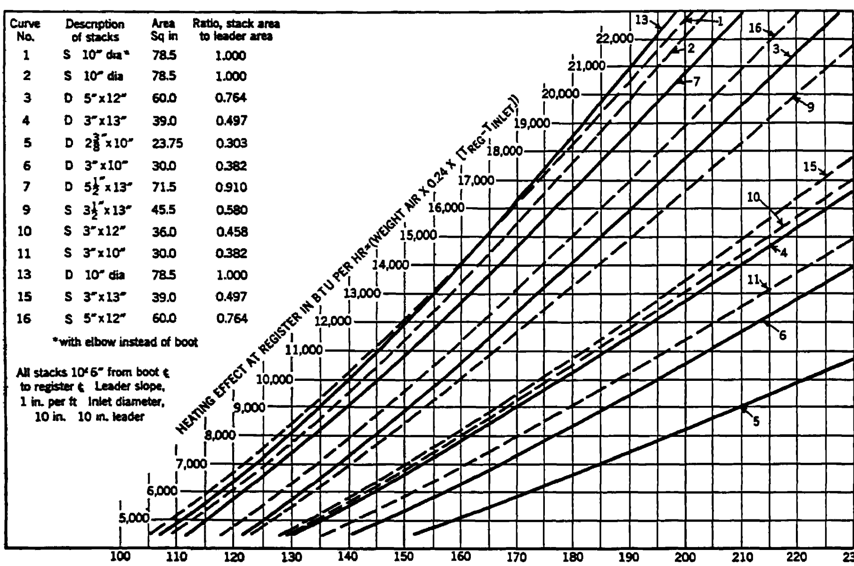


FIG. 4. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 10-IN. LEADER

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus in rooms having only small windows the grille should be brought as close to the furnace as possible, but if the room has a bay window, French doors, or other large sources of cooling or leakage of cold air, the grille should be placed close by, so as to collect the cool air and prevent drafts. When long ducts of this type are employed they must be made oversize. This precaution is particularly important when long ducts and short ducts are used in the same system. The long ducts must be oversize, if they are to operate satisfactorily in parallel with short ducts.

Return ducts from upstairs rooms may be necessary in apartments or other spaces which are closed off or badly exposed. Metal linings are

advisable in such ducts. It is important that these ducts be free from unnecessary friction and turbulence, and that they be located to prevent preheating of the air before it reaches the furnace.

Furnace Return Connection

Circulation of the air is accelerated if the return connection to the furnace is through a round inclined pipe connected to two 45 deg elbows rather than through a vertical pipe connected to two 90 deg elbows. The top of the return shoe should enter the casing below the level of the grate in the furnace. In order to accomplish this the shoe must be wide as is indicated in Fig. 6, No. 1 arrangement.

Tests of six different systems of cold air returns, Fig. 6, made at the University of Illinois[2], resulted in the following conclusions:

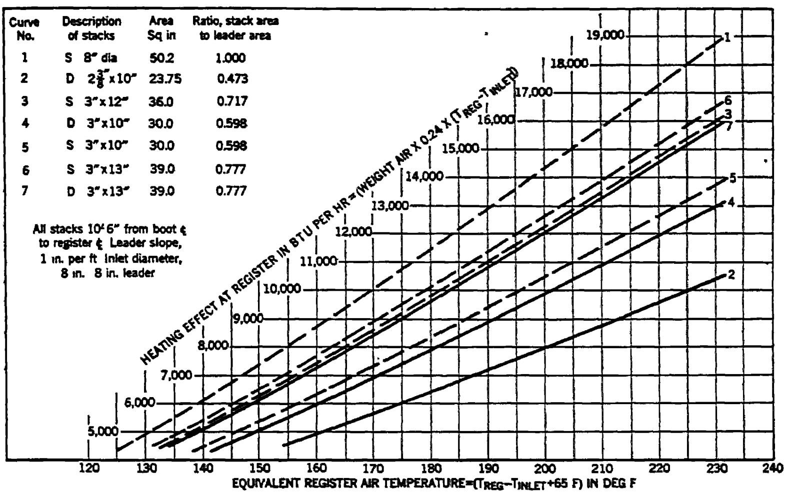


FIG. 5. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 8-IN. LEADER

1. In general, somewhat better room temperature conditions may be obtained by returning the air from positions near the cold walls.

2. Friction and turbulence in elaborate return duct systems retard the flow of air, and may seriously reduce furnace efficiency, and lessen the advantages of such a design.

3. The cross-sectional duct area is not the only measure of effectiveness. Friction and turbulence may operate to make the air flow out of all proportion to the various duct areas.

FURNACE CAPACITY

The size of furnace should, of course, be such as will provide the necessary air heating capacity, usually expressed in square inches of leader pipe area, and at the same time provide a grate of the proper area to burn the necessary fuel at a reasonable chimney draft. The total leader pipe area required is obtained by finding the sum of the leader pipe areas as already designated.

[2]Investigation of Warm Air Furnaces and Heating Systems, Part IV, by A. C. Willard, A. P. Kratz, and V. S. Day (University of Illinois, *Engineering Experiment Station Bulletin* No. 189).

The grate area will depend on several factors of which four are very important. First of all, the air temperature at the register for which the plant has been designed must be determined. Usually, this temperature is taken at 175 F. Second in importance is the combustion rate, *which must always correspond with the register air temperature*, as is shown by a set of typical furnace performance curves (Fig. 7) for a cast-iron, circular radiator furnace with a 23 in. diameter grate and 50 in. diameter casing. The third factor is efficiency, which is a function of the combustion rate, and varies with it as shown by the efficiency curve of Fig. 7. The fourth factor is the heat value per pound of fuel burned, which was 12,790 Btu. This is not shown on the curves since it was constant for all combustion rates.

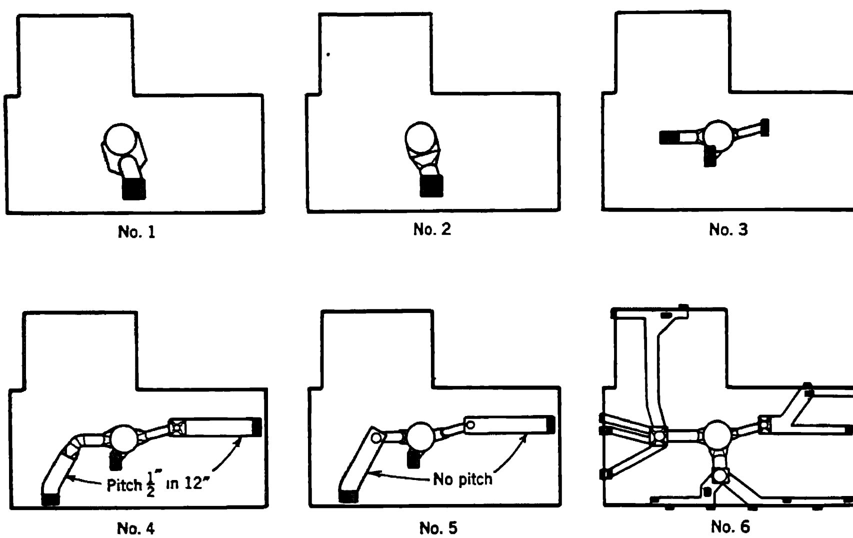


FIG. 6. ARRANGEMENT OF COLD AIR RETURNS FOR SIX INSTALLATIONS

It may be noted from Fig. 7 that for this particular furnace a register temperature of 175 F was accompanied by a combustion rate of approximately 7.5 lb per square foot per hour, a capacity at the bonnet of 152,000 Btu per hour and a furnace efficiency of 58 per cent. Under these conditions the capacity at the bonnet per square foot of grate was equivalent to a value of 52,800 Btu per hour and per square inch of grate was equivalent to 367 Btu per hour. If it is desired to use these curves to select a furnace to deliver air at 175 F register temperature in a house where the total heat loss is H Btu per hour and the loss between the furnace and the registers is 0.25 H Btu per hour, the area of the grate in square inches will be $\frac{1.25\,H}{367} = 0.0034\,H$.

If, on the other hand, it is desired to select a furnace to deliver air at 160 F register temperature, the combustion rate is 5.5 lb and the efficiency

of the furnace is 62 per cent. Under this condition the capacity at the furnace bonnet per square foot of grate is 43,200 Btu per hour and per square inch of grate is 300 Btu per hour, the required area of the grate in square inches in this case will be $\frac{1.25\ H}{300} = 0.0042\ H$. It should be noted that a larger grate area is required if the furnace is to deliver air at a lower register temperature.

The typical performance curves shown in Fig. 7 are not applicable to

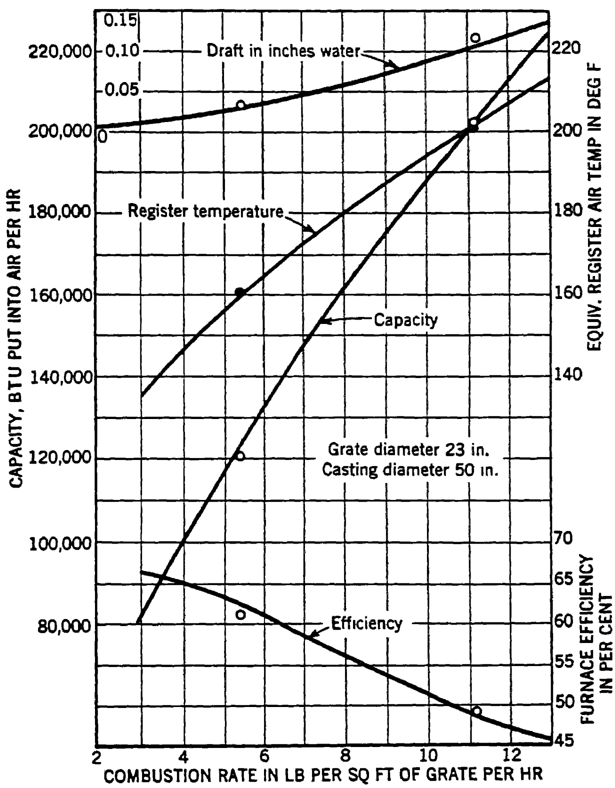


FIG. 7. TYPICAL PERFORMANCE CURVES FOR A WARM-AIR FURNACE AND INSTALLATION IN A THREE-STORY TEN LEADER PLANT, OPERATING ON RECIRCULATED AIR

all furnaces and hence for ordinary design purposes the values recommended in the Standard Code[3] should be used. The equation for a furnace having a ratio of heating surface to grate area of 20 to 1 is equal to:

$$H = \frac{G \times p \times f \times E_1 \times E_2 \times 0.866}{144} \quad (7)$$

[3]Standard Gravity Code for the Design and Installation of Gravity Warm Air Heating Systems in Residences. This code has been sponsored by the *National Warm Air Heating and Air Conditioning Association*, the *National Association of Sheet Metal Contractors*, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It is recommended that the installation of all gravity warm air heating systems in residences be governed by the provisions of this code, the tenth edition of which may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 5 E. Long St., Columbus, Ohio.

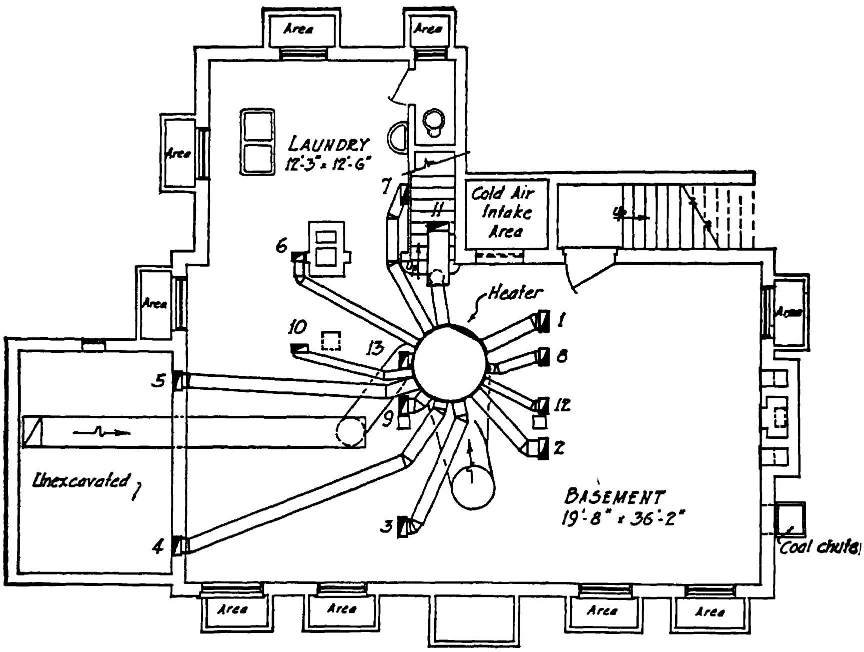


FIG. 8. BASEMENT PLAN, RESEARCH RESIDENCE

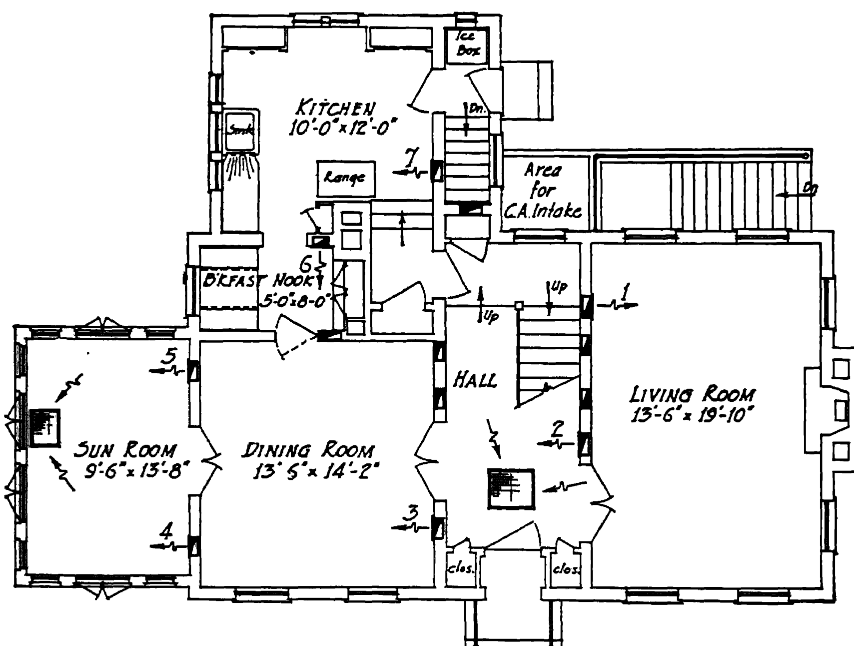


FIG. 9. FIRST FLOOR PLAN, RESEARCH RESIDENCE

where

G = grate area, square inch.
p = combustion rate, pound coal per square foot of grate per hour.
f = heating value of the coal, Btu per pound.
E_1 = efficiency at bonnet, ratio of heat delivered at bonnet to heat developed in furnace.
E_2 = efficiency of duct transmission, ratio of heat delivered at register to heat delivered at bonnet.
0.866 = factor of safety to allow for contingencies under service conditions such as accumulations of soot and ashes, ineffective firing methods, etc.
H = total heat loss from structure.

An addition of 2 per cent of the furnace capacity is proposed for each unit when the ratio of heating surface to grate area exceeds 20. This addition is based on tests[4] conducted at the University of Illinois on seven types of furnaces having varying ratios of heating surface to grate area. This correction does not, however, apply to values of the ratio less than 15 nor greater than 30.

By transposing the terms in Equation 7 and adding the correction term for ratios of heating surface to grate area other than 20 to 1, the following equation is obtained:

$$G = \frac{144 \times H}{p \times f \times E_1 \times E_2 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \tag{8}$$

in which R = ratio of heating surface to grate area.

In the case of the Standard Code[5] the numerical values used in Equation 8 were based on those determined from the tests conducted on the different types of furnaces.

$$G = \frac{144 \times H}{7.5 \times 12{,}790 \times 0.55 \times 0.75 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \tag{9}$$

$$G = 0.004205 \frac{H}{[1 + 0.02\ (R\text{-}20)]} \tag{10}$$

As used in these calculations, H = Btu heat loss from the entire house per hour = summation of all room losses $H_1 + H_2$ + etc. + the Btu necessary to heat the outside air, if any, at intake. This outside air loss in Btu per hour will be approximately 1.27 times the cubic feet of air admitted through the intake per hour on a zero day. For systems which recirculate all the air this value will be zero. For systems which have an outside air intake, controlled by damper, this value might well be approximated, since this loss will probably be reduced to a minimum on a zero day. Assume for such cases that the building loss is increased by 25 per cent, and that there is the usual 25 per cent loss between furnace and registers.

TYPICAL DESIGN

The application of the preceding data to an actual example may be of assistance to the designer. Figs. 8, 9, 10 and 11 represent the plans of

[4]University of Illinois, *Engineering Experiment Station Bulletin* No. 246, by A. C. Willard, A. P. Kratz, and S. Konzo, Chapter X, pp. 126-146.
[5]Loc. Cit. Note 3.

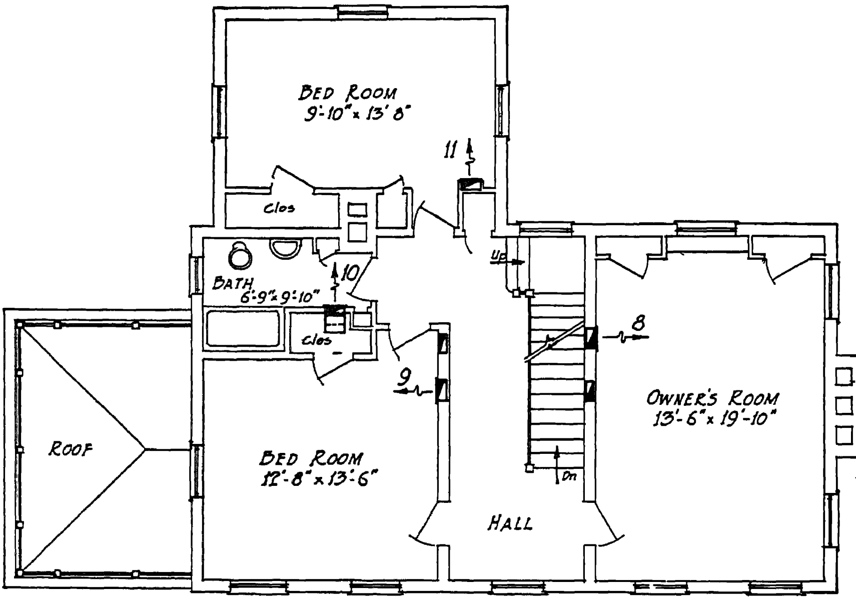


FIG. 10. SECOND FLOOR PLAN, RESEARCH RESIDENCE

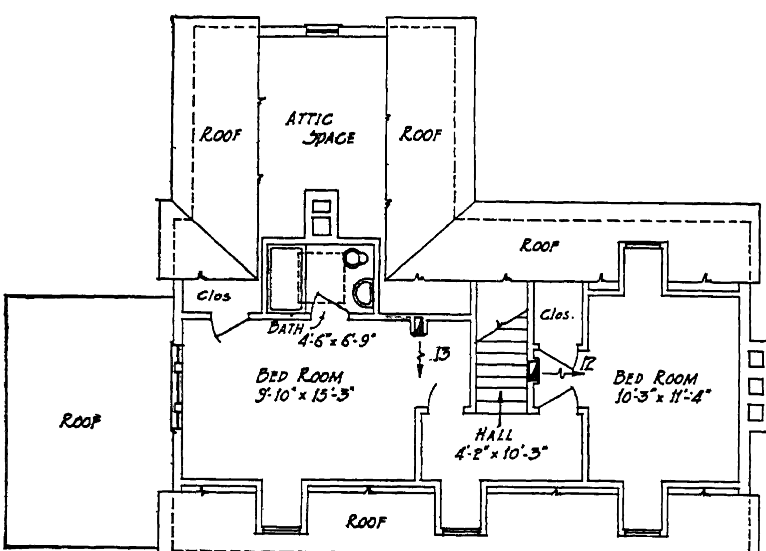


FIG. 11. THIRD FLOOR PLAN, RESEARCH RESIDENCE

the Warm Air Research Residence of the *National Warm Air Heating and Air Conditioning Association* erected at the University of Illinois[6].

Leaders, Stacks and Registers. (Direct Method)

Living Room, 1st floor:

17,250 ÷ 111 = 155 sq in. leader area. See summary Table 1; also see Art. 3 Sec. 1 of the Standard Gravity Code[7].

Leader diameter = 14 in.

Register size = 155 sq in. net area. Gross area = net area ÷ 0.7 = 14 in. × 16 in.

Owner's Room, 2nd floor:

15,030 ÷ 167 = 90 sq in. leader area. See summary Table 1; also see Art. 3 Sec. 2 of the Standard Gravity Code[7].

Leader diameter = 11.4, say 12 in.

Stack area = 0.7 × 90 = 63 sq in. = say 5 in. × 12 in.

Register area = 90 sq in. net area. Gross area = net area ÷ 0.7 = 12 × 12 or 12 in. × 14 in.

In like manner the leaders, stacks and registers are calculated for each room in the house.

Leaders, Stacks and Registers. (Code[7] Method. See Art. 3, Sec. 1, 2, 3)

Living Room (Glass = 90, Net wall = 405, Cubic contents = 2405)

$$\text{Leader} = \left(\frac{90}{12.6} + \frac{405}{57} + \frac{2405}{800}\right) 9 = 155 \text{ sq in.}$$

Register, same as Direct Method.

Owner's Room (Glass = 68, Net wall = 394, Cubic contents = 2275)

$$\text{Leader} = \left(\frac{68}{12.6} + \frac{394}{57} + \frac{2275}{800}\right) 6 = 91 \text{ sq in.}$$

Stack and Register, same as Direct Method.

Assuming all air recirculated, the minimum furnace for the plant will be:

Grate area = 0.0042 × 132,370 = 556 sq in.
Use 27 in. diameter grate. (Equation 10.)

If provision should be made for certain outside air circulation, then increase the building heat loss by, say, 25 per cent and obtain by Equation 10 a 30 in. grate.

Experiments at the University of Illinois[8] have shown that the capacity of a furnace may be increased nearly three times by an adequate fan, with a constant register or delivery temperature maintained, *provided that the rate of fuel consumption can be increased to provide the necessary heat.* In other words, the capacity of a forced circulation system *is limited by the ability of the chimney to produce a sufficient draft, and the ability of the fan to deliver an adequate amount of air.*

---

[6]Plans used with permission. Bathroom on third floor not heated.
[7]Loc. Cit. Note 3.
[8]University of Illinois, *Engineering Experiment Station Bulletin* No. 120, p. 129.

TABLE 1. SUMMARY OF DATA APPLIED TO WARM AIR RESEARCH RESIDENCE

Rooms	From Chapter 7 Estimating Heat Losses Btu Heat Losses H	Leader Area Sq In.	Stack Area Sq In. $0.7 \times LA$	Leader Diameter Inches	Stack Size Net	Register Size Gross
First Floor		$= 0.009H$				
Living	17250	155	—	14	....	14×16
Dining	6810	61	—	9	....	8×12
Breakfast	2300	21	—	8	....	8×10
Kitchen	9210	83	—	11 or 12	....	12×14
Sun	25710	230	—	Two 12	....	Two 12×14
Hall and stair	12570	113	—	12	....	12×14
Second Floor		$= 0.006H$				
Owner's	15030	90	63	11 or 12	5×12	12×14
S. W. Bed	9800	59	41	9	$3\frac{1}{2} \times 12$	8×12
Bath	2450	15	10	8	3×10	8×10
N. Bed	14800	89	62	11 or 12	5×12	12×14
Third Floor		$= 0.005H$				
E. Bed	8220	41	29	8	3×10	8×10
W. Bed	8220	41	29	8	3×10	8×10

BOOSTER FANS

Booster fans often may be arranged to operate when gas or oil burners are running and to stop automatically when the burners shut down. The booster equipment is most effective in increasing output at low operating temperatures. According to tests, efficiencies may be advanced from 60 per cent for gravity to 70 per cent with boosters at low operating temperatures, but at high operating temperatures gravity and booster efficiencies are almost identical[9].

[9]University of Illinois, *Engineering Experiment Station Bulletin* No. 141, p. 79, and No. 246.

Chapter 19

MECHANICAL WARM AIR FURNACE SYSTEMS

Furnaces, Fans and Motors, Sound Control, Sprays and Filters, Air Distribution Design, Automatic Controls, Design of Heating System, Selecting the Furnace, Selecting the Fan, Heavy Duty Fan Furnaces, Humidification, Cooling Methods, Cooling System Design

MECHANICAL warm air or fan furnace heating systems[1], which are a special type of central fan systems, are particularly adapted to residences, small office buildings, stores, banks, schools, and churches. Circulation of air is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering its bottom, as in gravity systems described in Chapter 18. The advantages of mechanical systems, as compared with gravity systems are:

1. The furnace can be installed in a corner of the basement, leaving more basement room available for other purposes.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view except in the furnace room.

3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by sprays or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

Much of the equipment used in central fan systems is the subject matter of other chapters. It is the purpose of this chapter to discuss the coordinated design and to deal in detail only with problems not covered elsewhere which refer particularly to the whole problem of fan warm air furnace heating and air conditioning.

---

[1]See University of Illinois *Engineering Experiment Station Bulletin* No. 266 by A. P. Kratz and S. Konzo for details of tests conducted in Warm Air Research Residence.

FURNACES

Furnaces for mechanical warm air systems may be made of cast-iron, steel, or alloy. Cast-iron furnaces are usually made in sections and must be assembled and cemented or bolted together on the job. Steel furnaces are made with welded or riveted seams. The proper design of the furnace depends largely on the kind of fuel to be burned. Accordingly, various manufacturers are making special units for coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as follows:

1. Coal Burning:
   a. *Bituminous*—Large combustion space with easily accessible secondary radiator or flue travel.
   b. *Anthracite or coke*—Large fire box capacity and liberal secondary heating surfaces.
2. Oil Burning:
   a. Liberal combustion space.
   b. Long fire travel and extensive heating surface.
3. Gas Burning:
   a. Extensive heating surface.
   b. Close contact between flame and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential heaters. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. It is recommended that the system be designed for blow-through installations, so that the furnace shall be under external pressure in order to minimize the possibility of leakage of the products of combustion into the air circulating system.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of any number of furnaces, using one or more fans.

Furnace Casings

Casings are usually constructed of galvanized iron, 26-gage or heavier, but they may also be constructed of brick. Galvanized iron casings should be lined with sheet iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. Casings for commercial or heavy duty furnaces, if built of galvanized iron, should be insulated with fireproof insulating material at least 2 in. thick. It is generally believed that either brick or sheet metal casing should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

For furnace casings sized for gravity flow of air, where a fan is to be

used, some form of baffling must be employed if the desired results are to be expected. Many manufacturers recommend the use of special baffles to restrict the free area within the casing and to force impingement of the air against the heating surfaces. A method for making these baffles for furnaces with top horseshoe radiators and for furnaces with back crescent radiators is illustrated in Fig. 1.

Either square or round casings may be used. Where square casings are

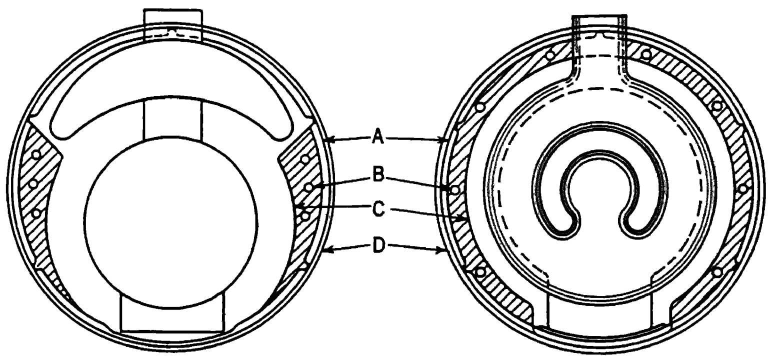


FIG. 1. USUAL METHOD OF BAFFLING ROUND CASINGS FOR FAN FURNACE WORK

A. Liner, 1 in. from casing. *B*. Hole to vent baffle. *C*. Baffle, closed top and bottom. *D*. Outer casing.

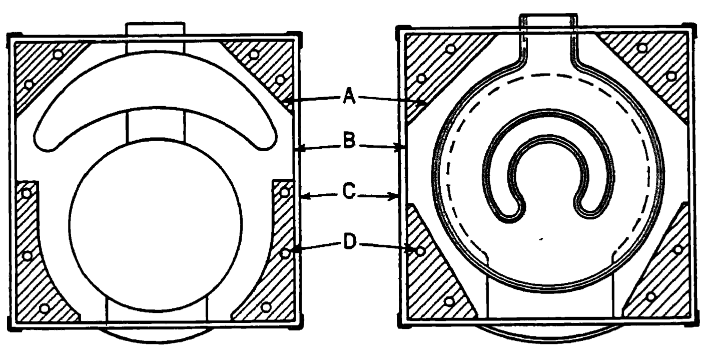


FIG. 2. METHOD OF BAFFLING SQUARE FURNACE CASING FOR FAN FURNACE WORK

A. Baffle, closed top and bottom. *B*. Liner, 1 in. from casing. *C*. Outer casing. *D*. Hole to vent baffle.

used, the corners must be baffled to reduce the net free area and to force impingement of air against the heating surfaces. Fig. 2 shows a satisfactory method of baffling square furnace casings for fan furnace work.

The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet

and thus provides a larger plenum chamber. Fig. 3 illustrates a complete residence fan furnace installation showing location of fan, furnace, filters, plenum chamber and method of take-off of warm air pipe.

FANS AND MOTORS

Centrifugal type fans are most commonly used, and these may be equipped with either backward or forward curved blades. Motors may be mounted on the fan shaft or outside of the fan with belt connection. Multi-speed motors or pulleys are desirable to provide a factor of safety and to allow for increased air circulation. For additional information on fans and motors, see Chapters 29 and 35.

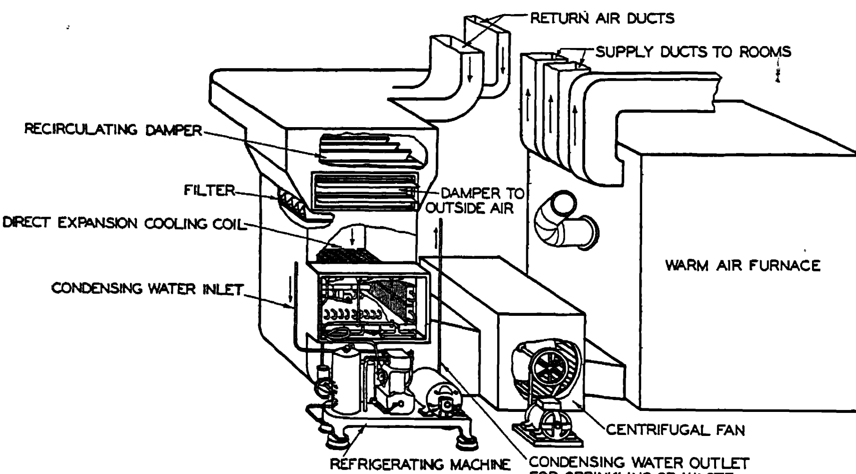


Fig. 3. Complete Residence Fan Furnace Installation for Winter Heating and Summer Cooling

SOUND CONTROL

Special attention should be given to the problem of noise elimination. The fan housing should not be directly connected with metal, either to the furnace casing or to the return air piping. It is common practice to use canvas strips in making these connections. Motors and their mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing. The installation of a fan directly under a cold air grille is not recommended on account of the noise objection. (See also Chapter 32.)

FILTERS

There are many satisfactory types of filters on the market. These include dry filters, viscous filters, oil filters and other types, some of which must be cleaned, some of which must be cleaned and recharged with oil, and some of which are inexpensive and may be discarded when they become dirty, and replaced with new ones.

The resistance of a filter must be considered in the design of the system since the resistance rises rapidly as the filter becomes dirty, thus impairing the heating efficiency of the furnace, in fact, endangering the life of the furnace itself. Manufacturers' ratings of filters must be carefully regarded, and ample filter area must be provided. Filters must be replaced or cleaned when dirty. (See also Chapter 28.)

AIR DISTRIBUTION

The conditions of comfort obtained in a room are greatly influenced by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. Due to the economic considerations involved, it is common practice to locate the supply openings on the inside walls of a residence and the return openings nearest the greatest outside exposure. Many designers prefer, however, to locate the supply registers so that the warm air from the registers *blankets* a cold wall, and mixes with the cold air dropping off from the exposed walls. This may be accomplished by the use of a supply register placed close to an outside wall in such a position that the warm air sweeps the cold wall surface. The ducts leading to supply registers which are located on exposed walls should be adequately insulated to reduce the heat loss from the ducts.

Register and Grille Openings

Supply registers located in the floor are effective, but as they require frequent attention to keep them clean they should be avoided where another effective register location can be found. Tests conducted in the Warm Air Research Residence[2] have indicated that excellent results are obtainable with either high side wall or baseboard registers, providing a reasonable amount of precaution is employed. Baseboard registers should be of a deflecting-diffuser type which throws the air downward toward the floor and diffuses it at the same time. Register air temperatures under 125 F and air velocities over 500 fpm should be avoided as they may cause drafts.

High side wall registers must be of such type that the air is delivered horizontally or in a slightly downward direction, and must be so located as to avoid impingement of air on ceiling or wall. Directional flow diffusing type should be used to insure best results. Register air velocities should be such that the air stream carries to the opposite exposure. Velocities under 500 fpm are not recommended. Register air temperatures under 125 F are not objectionable. In fact, when cooling is desired, better air distribution is obtained with high side wall registers.

Unless registers, regardless of their location, are well proportioned and designed as well as decorated to harmonize with the trim, they may be unsightly. All registers should be equipped with dampers and must be sealed against leakage around the borders or margins.

---

[2]Loc. Cit. Note 1.

Velocities through registers may be reduced by the use of registers larger than the connecting pipes. Some suggestions for equalizing velocities over the face area of the register by means of diffusers are illustrated in Fig. 4. Merely to use a larger register may not result in materially reduced velocities unless diffusers are used.

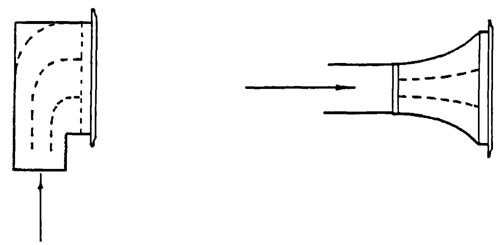


FIG. 4. DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

Dampers

Suitable dampers are essential to any trunk or individual duct system, as it is virtually impossible to so lay out a system that it will be absolutely in balance without the use of dampers. Special care must be used in the design of any system to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. (See Fig. 4, Chapter 31.)

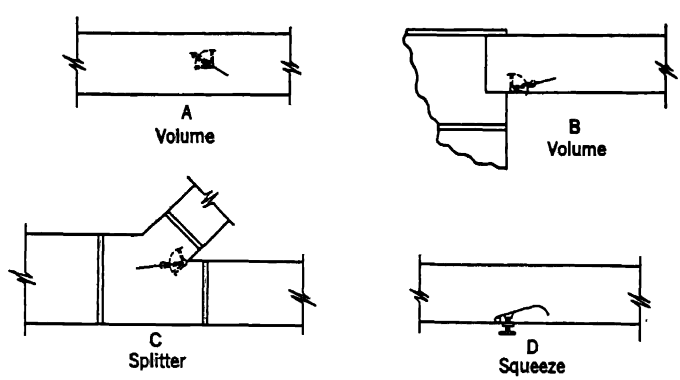


FIG. 5. THREE TYPES OF DAMPERS COMMONLY USED FOR TRUNK AND INDIVIDUAL DUCT SYSTEMS

Three types of dampers are commonly used in trunk and individual duct systems. *Volume dampers* are used to completely cut off or reduce the flow through pipes. (See *A* and *B*, Fig. 5.) *Splitter dampers* are used where a branch is taken off from a main trunk. (See *C*, Fig. 5.) *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. (See *D*, Fig. 5.) It is essential that a damper be provided for each main or duct branch. A positive locking device should be used with each type of damper.

Ducts

The ducts may be either round or rectangular. The radii of elbows should be not less than one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in a properly designed system can be largely eliminated through proper care in the planning and installation of the control system[3]. The essential requirements of the control are:

1. To keep the fire burning when using solid fuel regardless of the weather.
2. To avoid excessive bonnet temperatures with resultant radiant heat losses into the basement.
3. To avoid the overheating of certain rooms through gravity action during off periods of blower operation.
4. To have a sufficient supply of heat available at all times to avoid lag when the room thermostat calls for heat.
5. To prevent cold air delivery when heat supply is insufficient.
6. To avoid heat loss through the chimney by keeping stack temperatures low.
7. To provide quick response to the thermostat, with protection against overrun.
8. To provide for humidity control.
9. To provide a means of summer control of cooling.
10. To protect against fire hazards.

The following controls are desirable:

1. A *thermostat* located at a point where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. This location would be near an outside wall but not upon it, in a sun room, or in a room with some unusual exposure. The thermostat, of course, should not be located where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct or register.

2. A thermostatic blower switch located in the bonnet to permit blower operation only between the temperatures of 100 F and 150 F. In certain extreme cases it may be necessary, or weather conditions may make it advisable, to adjust the high limit to a higher temperature than that given. Another location sometimes used for the blower switch is in the main duct near the frame opening from the bonnet.

3. A *protective limit control* located in the bonnet to shut down the system independently of the thermostat if the bonnet temperature exceeds 200 F.

4. On oil and gas burner installations, a control should be included which will shut down the system if the fire goes out or if there is a failure of the ignition system.

5. A *humidistat* to regulate the moisture supplied to the rooms.

6. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying, or a time interval contactor is used that will start the stoker to run a predetermined length of time at predetermined intervals.

METHOD OF DESIGNING FORCED-AIR HEATING SYSTEMS

1. Determine heat loss from each room in Btu per hour. (See Chapter 5.)

2. Locate warm air registers and return registers on plans of house, beginning with the upper story rooms.

---

[3]Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37).

3. Sketch in duct layout to connect all registers and grilles with the central unit.

4. Determine equivalent length of duct for each register, allowing 10 diameters of straight pipe as equivalent to each 90 deg elbow having an inner radius not less than the diameter of the round pipe or the depth of the rectangular pipe.

5. Select a value for temperature of the air at the furnace bonnet. It is customary to use some value lying between 150 to 165 F. Use lower value if larger number of air recirculations is desired. It is recommended that the number of air recirculations should be in excess of 5 per hour.

6. Determine approximate value of temperature reduction in each duct caused by heat loss from the ducts. A value of from 0.3 to 0.6 F per foot of duct has been obtained from tests conducted in the Research Residence installation for uninsulated duct lengths up to approximately 60 ft.

7. Subtract this temperature reduction from the assumed bonnet air temperature to obtain an approximate value of the register air temperature for each register.

8. Determine the required air volume for each room from the following equation, or from the values listed in Table 1:

$$Q = \frac{H}{60 \times 0.24 \times d\,(t_r - 65)} \tag{1}$$

where

Q = required air volume, cubic feet per minute.
H = heat loss of room, Btu per hour.
d = density of air at register temperature, pounds per cubic foot.
t_r = register temperature, degrees Fahrenheit.
0.24 = specific heat of air.
65 = return air temperature.

For any given register temperature the solution of this equation simplifies to the following form:

$$Q = H \times \text{Factor} \tag{2}$$

in which the values of the Factor may be obtained from Table 1.

9. Determine register size from the air volume delivered to each room by the following formula:

$$\text{Free area of register, square feet} = \frac{Q}{V} \tag{3}$$

$$\text{Gross area of register, square feet} = \frac{\text{Free Area}}{R} \tag{4}$$

where

Q = required air volume, cubic feet per minute.
V = velocity at register face, feet per minute.
R = ratio of free area to gross area of register.

TABLE 1. FACTORS CORRESPONDING TO REGISTER TEMPERATURE FOR EQUATION 2

Register Temperature	Factor
110	0.0221
120	0.0184
130	0.0158
140	0.0140
150	0.0125
160	0.0114
170	0.0105

Allowable register velocities to be used in Equation 3 are approximately as follows:

Baseboard, non-deflecting type, maximum = 300 fpm.
Baseboard, deflecting toward floor, maximum = 500 fpm.
Baseboard, deflecting and diffusing = up to 800 fpm.
High side wall = not less than 500 fpm.

10. Duct systems for forced-air installations may consist of either trunk systems or individual duct systems.

Trunk Systems. Determine duct sizes and friction losses as outlined in Chapter 31, except that for residence applications the velocities in the main duct and in the various parts of the system should approximate the values recommended in Table 2.

Individual Duct Systems. An individual duct system is one having separate ducts extending from the heating unit to each register. In designing such a system select first the duct having the greatest equivalent length. Select a reasonable velocity using Table 2 as a guide. From friction chart in Chapter 31 determine unit friction loss per 100 ft of run, and from this the total friction loss in the duct selected. If this total friction loss exceeds a reasonable value a lower velocity should be used.

The remaining ducts are proportioned so that the total pressure in each duct is the same as that calculated for the longest duct. The added resistance necessary in the shorter ducts is accomplished by increasing the velocity in these ducts. No duct should be less than 6 in. in diameter, nor should the velocity in any duct exceed approximately 1200 fpm. The final adjustment in a duct system may be made by employing dampers.

TABLE 2. RECOMMENDED VELOCITIES THROUGH DUCTS AND REGISTERS

Description	Low Velocity System (fpm)	Medium Velocity System (fpm)	High Velocity System (fpm)
Main ducts	500	750	1000
Branch ducts	450	600	750
Wall stacks	350	500	600
Baseboard registers (max.)	300	350	400
Wall registers above 5 ft (min.)	500	550	600

Instead of proportioning the ducts as outlined in the preceding paragraph it is more usual in practice to proportion all the ducts so that they have the same velocity as that used in the longest duct and to balance the system by employing dampers in the shorter ducts.

Return duct systems are designed making use of the same principles as those used in the design of supply duct systems. In this case the design may be based on the volume of air corresponding to the density of air existing in the return ducts, or in order to provide a factor for air leakage, it may be based on the same volume as used for the supply ducts.

11. Determine frictional resistance in:

a. Supply side of system as outlined in Item 10.
b. Return side of system as outlined in Item 10.
c. Furnace units, casing or hood, which is usually considered as equivalent to 0.03 to 0.10 in. of water.
d. Accessories such as washers or air filters, from manufacturer's data.
e. Inlet and outlet registers and grilles, from manufacturer's data.
f. Other accessory equipment such as cooling coils, from manufacturer's data.

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the items listed in the preceding discussion. In practice it is recommended that liberal allowances should be made so that the

fan will be capable of delivering air against pressures that may not have been foreseen during the design of the duct system.

12. Select a furnace capable of delivering heat at the register outlets equal to the total heat loss of the structure to be heated.

The following formula may be used for coal burning furnaces:

$$G = \frac{H}{f \times p \times E_1 \times E_2 \left[1 + 0.02\,(R - 20)\right]} \tag{5}$$

where

G = required grate area, square feet.
H = total heat loss from building, Btu per hour.
f = calorific value of coal, Btu per pound.
p = combustion rate in pounds of fuel per square foot of grate per hour.
E_1 = furnace efficiency based on heat available at bonnet.
E_2 = efficiency of transmission based on ratio of heat delivered at register to heat available at bonnet.
R = ratio of heating surface to grate area.

In practice it is customary to use the following constants:

f = 12,000 (for specific values, see Table 5, Chapter 7).
p = 7.5 lb.
E_1 = 0.65 lower efficiency must be used with highly volatile solid fuel.
E_2 = 0.85.

The foregoing procedure for determining the size of the furnace to be used applies to continuously heated buildings.

13. Although intermittently heated buildings usually have their heat losses computed according to the standard rules for determining such losses, these rules do not take into account the heat which will be absorbed by the cold material of the building after the air is raised in temperature. This heat absorption must be added to the normal heat loss of the building to determine the load which the heating plant must carry through the warming-up process. It is customary to increase the normal heat loss figure by from 50 to 150 per cent depending upon the heat capacity of the construction material, the higher percentage applying to materials of high heat capacity such as concrete and brick. Fan furnace systems are well adapted for heating intermittently heated buildings as these systems do not require the warming of intermediate piping, radiators, or convectors, the generation of steam, or the heating of hot water.

14. Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected. Use the manufacturer's Btu ratings of furnaces designed for exclusive use with oil, and select a burner with liberal excess capacity.

15. The selection of the proper size gas furnace for a constantly heated building can be easily made by using the following *American Gas Association* formula:

$$R = \frac{H}{0.9} \tag{6}$$

where

H = total heat loss from building, Btu per hour.
R = official *A.G.A.* output rating of the furnace, Btu per hour.

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output. The proper sizing may be done by means of the following formula:

$$I = 1.59\,H \tag{7}$$

where

I = Btu per hour input.

The factor 1.59 is the multiplier necessary to care for a 10 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

16. Specify location and type of all dampers in both supply air and return air sides of system. Specify controls including location of all thermostats. Arrange for proper control of humidifying equipment.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings are available in sizes ranging from 400,000 to 3,000,000 Btu per hour per unit. Heavy duty heaters may be arranged in combinations of one or more units in a battery. One typical arrangement is shown in Fig. 6.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area on furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

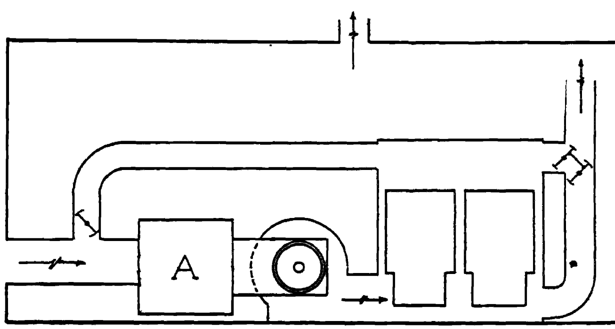


FIG. 6. HEATER ARRANGED FOR USE OF AIR WASHER OR FILTER (A) WITH HEATED AIR TO MIX WITH OUTSIDE AIR FOR TEMPERING, SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR AND EXHAUST TO ATMOSPHERE

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 20. Ducts are designed by the method outlined in Chapter 31.

HUMIDIFICATION

Mechanical warm air systems offer a means of proportioning and distributing moisture-bearing air; consequently, during the winter months

humidifiers may be employed to deliver water vapor to the fan-driven air stream in proper amounts to produce a more humid atmosphere, with increased comfort for people and increased life for household furnishings. Temperatures and relative humidities should be governed within the limits of the generally accepted standards. See Chapters 2 and 26 for more detailed information on this point.

In earlier types of furnaces, water evaporating pans were usually placed in the cool portions of the air stream, but modern types usually locate them in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. Without the addition of this heat, termed the latent heat of evaporation, water injected into the air will be carried along in the form of tiny globules until it falls out of the stream or is deposited upon some surface. Furthermore, when dry air is in contact with water for a sufficient length of time without the presence of a sizable body of water or a source other than air from which this latent heat of evaporation can be taken, such heat is supplied from the air. There is, therefore, a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. (See Chapter 26.)

Sprays for residence systems may be provided in separate housings to be installed on the inlet or outlet side of the fan, or they may be integral with the fan construction. They operate at water pressures of from 10 to 30 lb and use two or more spray nozzles for washing and humidification. The sprays should be adjusted to completely cover the air passages.

Sprays are usually controlled by solenoid valves wired in parallel with the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Sprays used in connection with commercial or heavy duty plants should be a regulation type of commercial spray.

Residence Requirements

The principles underlying humidity requirements and limitations for residences are summarized in *University of Illinois Bulletin* No. 230[4], as follows:

1. Optimum comfort is the most tangible criterion for determining the air conditions within a residence.

2. An effective temperature of 65 deg[5] represents the optimum comfort for the majority of people. Under the conditions in the average residence a dry-bulb temperature of 69.5 F with relative humidity of 40 per cent is the most practical for the attainment of 65 deg effective temperature.

---

[4]See Humidification for Residences, by A. P. Kratz (*University of Illinois, Bulletin* No. 230).

[5]66 deg is the optimum winter effective temperature recommended by the A.S.H.V.E. Committee on Ventilation Standards.

3. Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air inleakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.

4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight-fitting storm sash or the equivalent is installed.

5. The problems of humidity requirements and limitations cannot be separated from condensations of good building construction, and the latter should receive serious attention in the installation of humidifying apparatus.

The following conclusions were drawn from the experimental results reported in the aforementioned bulletin:

1. None of the types of gravity warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except only in moderately cold weather.

2. The water pans used in the radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the use of basement air. A more positive cooling effect may be obtained through air washers where the temperature of the water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew-point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew-point. Ample coil surface is necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with the fan and duct system for the heating installation, and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. (See also Chapters 20 and 23.)

Results at Research Residence

The following conclusions may be drawn from the studies thus far completed in the Research Residence, subject to the limitations of the conditions under which the tests were run[6]:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hours on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

[6]A.S.H.V.E. Research Report No. 947—Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 95). A.S.H.V.E. Research Report No. 979—Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 167).

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10 year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the frictional resistance of the coil to flow of air must be given careful consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

METHOD OF DESIGNING COOLING SYSTEM

The general procedure for the design of a cooling system in a forced-air installation is as follows:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 3 and 6.) Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets. In Research Residence tests[7] a value of from 65 to 70 F was found satisfactory.

3. Determine indoor conditions to be maintained. In Research Residence 80 F dry-bulb and 45 per cent relative humidity were found satisfactory.

4. Determine the quantity of air to be introduced into each room. (See Chapter 20.)

5. Estimate heat loss in duct system between cooling unit and supply registers.

6. Calculate the heat to be removed by the cooling unit, in the form of sensible heat and latent heat.

7. Determine size of ducts in duct system and size of registers, as explained in this chapter under the heading of Method of Designing Forced-Air Heating Systems.

8. Determine pressure loss in duct system and select fan as also explained in the same section.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. (See Chapter 25.)

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

---

[7]Loc. Cit. Note 6.

Chapter 20

CENTRAL SYSTEMS FOR COMFORT AIR CONDITIONING

Types of Systems for Heating, Humidifying, Cooling, Dehumidifying with Modifications, Design Details, Load Calculations, Selection of Equipment

THE purpose of this chapter is to offer a summary of the procedure generally followed in the design of central systems for comfort air conditioning. Detailed information for making the various calculations is found elsewhere in THE GUIDE, and reference to the proper chapter follows in this text.

The reader is also referred to the Code of Minimum Requirements[1], which was prepared by a joint Committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers*, and which was adopted by the Society in January 1938. Engineers should also be familiar with any state or municipal codes which may apply to the problem under consideration.

A central system consists of a fan with complete supply and return ducts designed to serve one or more conditioned spaces, together with some or all of the following equipment: heating coils, cooling coils, humidifiers, dehumidifiers, air cleaning devices, and control equipment. These various items of equipment are assembled into a properly balanced system which has for its purpose the control of temperature and humidity within the conditioned space.

The principal types of central systems are illustrated in Figs. 1 to 5, but space does not allow showing all of the modifications to which these types may be subjected by the conditions existing in the building or by the preferences of the designing engineer. Also, a central system need not provide for year-round conditioning, but may be designed only for heating, cooling, or simply for ventilation.

CLASSIFICATION OF SYSTEMS

Central systems in which dehumidification is accomplished by cooling may be classified as follows:

[1]Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27). Reprints of this code are available at $.10 a copy.

1. Central system with heating and cooling coils and humidifying sprays (Fig. 1). This system is very common and is used extensively for summer and winter conditioning.

2. Central system with preheating coil, washer and reheating coil (Fig. 2). This is widely used for winter conditioning on larger installations. Summer conditioning is accomplished by using cold water in the washer.

3. Blow-through system with heating and cooling coils and mixing dampers (Fig. 3). This is used where several different zones are served from one central system and the conditioning of the entering air for each zone is obtained by mixing varying quantities of air after passing through the heating and cooling coils.

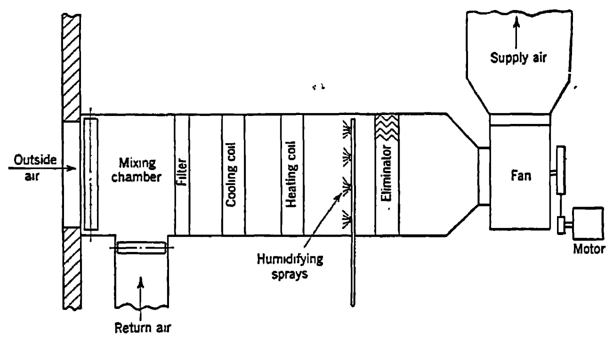


FIG. 1. CENTRAL SYSTEM WITH COILS AND SPRAYS

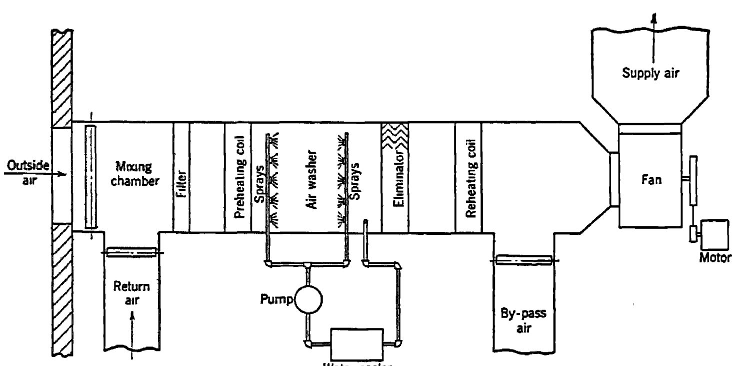


FIG. 2. CENTRAL SYSTEM WITH WASHER AND PREHEATING AND REHEATING COILS

4. Central outside air conditioning system with zone recirculating conditioners (Fig. 4). This is used on installations having large zones requiring independent control to meet local requirements for office and apartment buildings, hotels, etc.

5. Central conditioning system using either washer or coils with booster reheating coils (Fig. 5). This is used for special zone control where there is great variation in heating requirements of different spaces, as may be caused by different exposures to wind and sun.

6. Residential system. In Fig. 6 is shown a combination of unit equipment used for residential air conditioning. The arrangement is similar to Fig. 1, but the coil, humidifier, and fan section may be purchased as a unit. The sketch shows a system which permits the use of radiators or convectors in kitchens, baths, garages, and similar rooms.

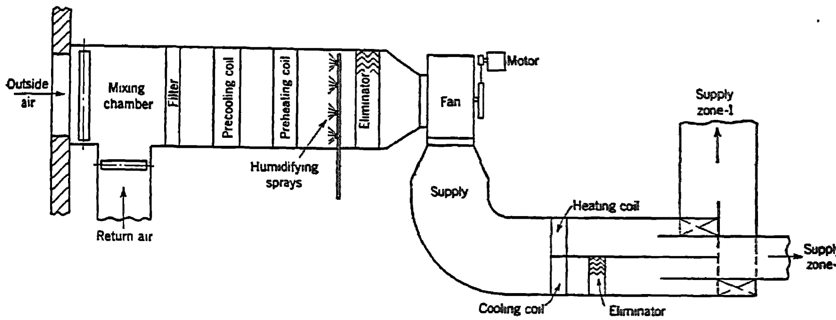


FIG. 3. BLOW-THROUGH SYSTEMS WITH COILS AND MIXING DAMPERS

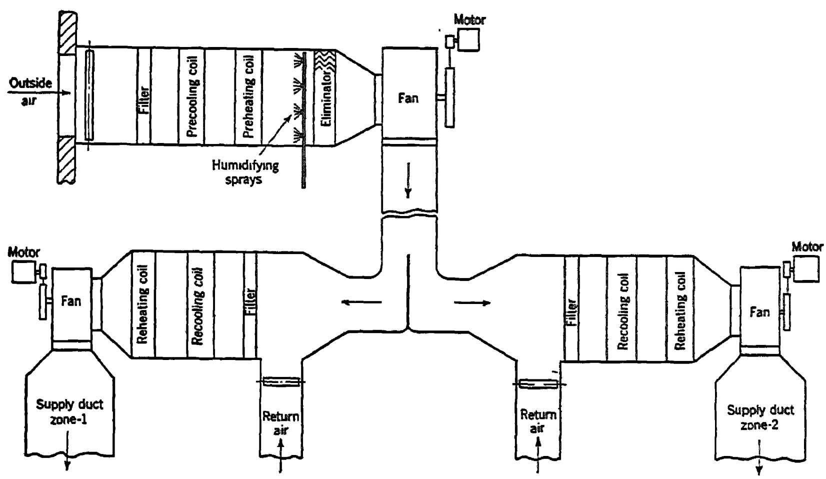


FIG. 4. CENTRAL SYSTEM WITH RECIRCULATING CONDITIONERS

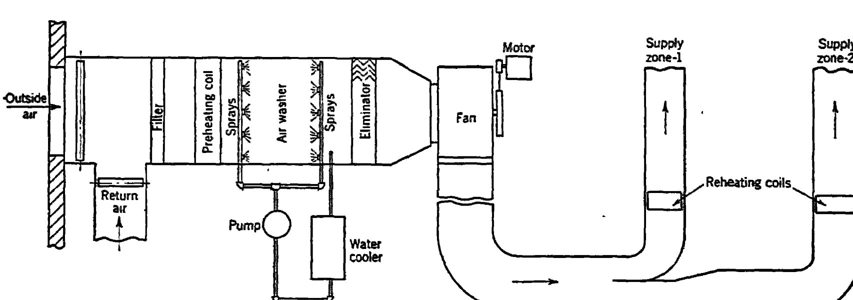


FIG. 5. CENTRAL SYSTEM WITH BOOSTER COILS

The addition of a cooling coil and compressor turns the system into a complete year-round conditioner.

Reheating

When air is dehumidified by cooling, the dry-bulb temperature leaving the dehumidifier is frequently lower than desired at the supply grille. In these cases, the dehumidified air must be reheated, and such reheating may be accomplished either by by-passing some air around the dehumidifier[2] (Fig. 2), or by using reheating coils.

Study of Figs. 1 and 2 will make it obvious that without the by-pass,

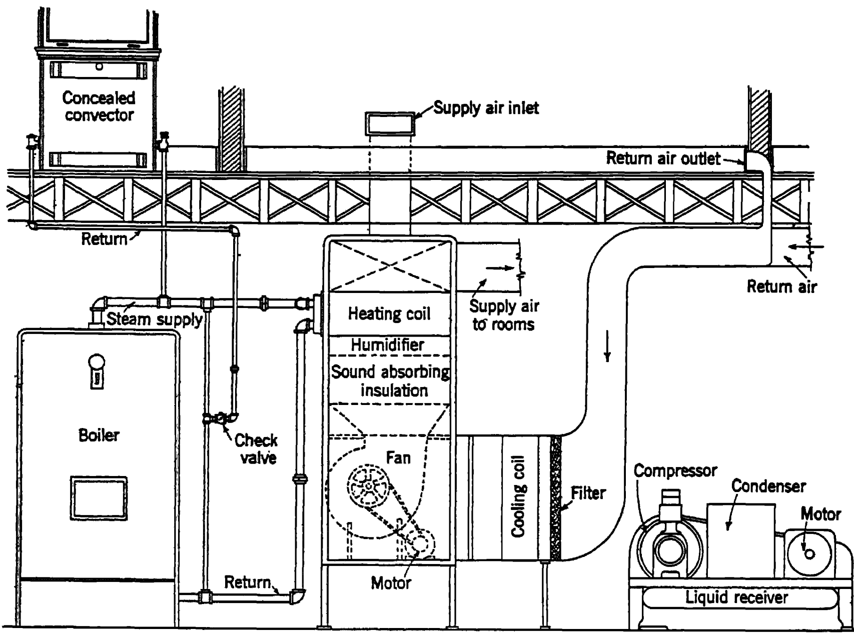


FIG. 6. RESIDENTIAL CONDITIONING SYSTEM WITH STEAM BOILER

all of the air is first cooled and then subsequently reheated, but with the by-pass, only part of the air is cooled, heat required for reheating being supplied by by-passed air. The by-pass thus reduces the cooling load and makes a separate reheating source unnecessary.

DESIGN OF SYSTEM

The factors which affect the design of an air conditioning system and the steps in the design are enumerated herewith.

Item 1. Design conditions.

a. Outside dry-bulb temperature in winter.
b. Outside dry- and wet-bulb temperatures in summer.
c. Inside dry- and wet-bulb temperatures (winter and summer).

[2]Patents exist covering the by-pass method.

Item 2. Design heating load.
- *a.* Heat transfer through windows, walls, partitions, doors, floors, skylights, roofs.
- *b.* Heat loss resulting from infiltration.
- *c.* Heat required to warm ventilation air.
- *d.* Heat required to evaporate moisture for humidification.
- *e.* Heat loss through ducts and coil housings.
- *f.* Allowances for heat emitting sources.

Item 3. Design cooling load.
- *a.* Heat transfer through windows, walls, partitions, doors, floors, skylights, roofs, including solar radiation.
- *b.* The sensible and latent heat emission of occupants.
- *c.* Heat emission of electrical, chemical, gas, steam or hot water apparatus, or lights (divide into sensible and latent heat).
- *d.* The sensible and latent heat gains resulting from infiltration.
- *e.* Sensible and latent heat to be removed from ventilation air.
- *f.* Heat gain through ducts and coil housings.

Item 4. Design of the air conditioning system.
- *a.* Establish the air temperatures at supply grilles for winter and summer.
- *b.* Calculate the air quantities for winter and summer.
  1. Adjust factors to get most satisfactory balance between winter and summer air quantities.
- *c.* Select coils, washers, heat exchangers, etc., with capacities equal to heating and cooling loads established in *Items 2* and *3*.
- *d.* Select air cleaning equipment.
- *e.* Select fans.
- *f.* Design duct system including supply and return grilles.
- *g.* Consider noise reduction problems.
- *h.* Design the control system.
- *i.* Calculate static pressure loss of the complete system.
- *j.* Select fan motors and drives and other auxiliary equipment.

Design Conditions

Outside design temperatures for principal cities are found in Chapter 5 for the heating season and in Chapter 6 for the cooling season. Recommended inside design temperatures for various types of buildings and for the seasons are found in Chapters 2 and 5.

Load Calculations for Heating

Complete tabular information is given in Chapter 3 for determining the heat loss through windows, walls, partitions, doors, floors, skylights, ceilings and roofs.

The heating capacity required to warm infiltration air is determined by methods shown in Chapter 4.

The minimum quantity of outside air brought in for ventilation is sometimes fixed by law; but when this is not the case, the A.S.H.V.E. Code[3] should be used as a standard. Outside air in sufficient quantities provides the best method of controlling objectionable odors, and the design should err on the safe side.

Code requirements state that the assumed rate at which air is to be positively introduced into the enclosure per occupant, when the contamination of air within the enclosures results entirely from respiratory process, shall not be less than 10 cfm per stated number of occupants

[3] Loc. Cit. Note 1.

which is indicated as being the maximum number of people within the enclosure when the sum of the remaining loads is a maximum. The Code further states that the assumed ventilation rates shall not be less than 15 cfm per stated number of occupants in enclosures where smoking is customarily permitted, and that provision shall be made for air removal from the enclosure either by natural or mechanical means at not less than the assumed ventilation rate. For the purpose of the Code, air quality or purity are assumed to be met if means are provided for the positive introduction of outside air in the amounts previously mentioned and for removal of 95 per cent by count of all dust particles over 10 microns in diameter from all air delivered to the enclosure.

The heat required to warm ventilation air is determined in the same manner as for infiltration air. It should be kept in mind, however, that infiltration increases the amount of heat required to warm the conditioned space, while ventilation air adds to the load on the heating coil.

The heat required to evaporate the necessary water required for winter humidification and superheat the resulting vapor in order to raise the moisture content of the outside air, assumed to enter the enclosure by infiltration or positively introduced for ventilation, should be calculated according to the information included in Chapter 1.

Information is given in Chapter 42 for calculating heat transfer through ducts and housings.

The heat emission of occupants, lights, and other sources is treated later in this chapter. This heat gain may be used as a credit against the heat loss calculations. In general, however, the design for heating disregards these gains as in most cases these values are not a continuous or uniform source of heat and the heating system must be adequate to maintain the required temperature at all times including nights, Sundays and holidays, when the space is not in normal use.

The sum of the several losses (*Items 2a* and *2b*) will give the amount of heat to be supplied to the conditioned space. The above quantity plus ventilation and humidification load and heat loss in ducts represents load on conditioner.

Load Calculations for Cooling

The heat gain through the windows, partitions, doors, floors, skylights, ceilings or roofs of the enclosure due to the air dry-bulb temperature difference assumed to exist between the air on the opposite sides of the construction may be determined from data given in Chapter 3. Charts and tables are given in Chapter 6 for the determination of sun effect on walls, roofs, and windows.

The heat gain from occupants may be calculated from data in Chapter 2, which give the metabolic rate for people engaged in various activities. In addition charts are included which give a separation of the sensible and latent heat losses from the body which should be itemized separately in all calculations.

The heat emission from various appliances should be calculated according to the information and data given in Chapter 6, with special consideration being given to the division of latent and sensible heat requirements of the apparatus. Complete details may also be found in

Chapter 6 for determining the heat gain resulting from electric lights and motors within the enclosure.

The heat gain from infiltration air and ventilation air is determined by the method shown in Chapter 4. Separate sensible and latent heat. (See Chapter 42 for heat gain in ducts.)

Air Distribution System for Heating

The total heating load to be supplied by the central system is determined from the several components of the load listed under *Item 2*. The quantity, air motion, and temperature of the treated air and the method of introducing it to the conditioned space should be designed so as to limit the variation in dry-bulb temperature to 3 F or less at a 5 ft level throughout that portion of the enclosure which is normally frequented by persons. It is desirable to avoid air velocities exceeding 50 linear feet per minute in the occupied zone between the floor and the 5 ft level. When architectural or other construction requirements necessitate the location of a supply or return grille below the 5 ft level in an occupied space, special consideration should be given to the air velocities in that region to avoid uncomfortable drafts.

It is desirable to use reasonably low temperature differences between the entering air and room conditions where possible. Air temperatures from 80 to 90 F will generally be satisfactory, although where the quantity of air to be circulated is kept at a minimum and where the arrangement of air inlets permits adequate mixing with the room air before reaching the breathing zone, higher temperatures from 100 to 120 F can be used. Having selected the desired temperature of the entering air, the quantity of air is determined as follows:

$$Q = \frac{H}{60d \times 0.24\,(t_2 - t_1)} \tag{1}$$

where

Q = volume of air to be introduced, cubic feet per minute.
H = sensible heat loss of space to be conditioned, Btu per hour.
d = density of air, pounds per cubic foot.
t_2 = outlet temperature at the grille, degrees Fahrenheit.
t_1 = design room temperature, degrees Fahrenheit.

If the air quantity calculated is excessive, it may be decreased by using a higher entering temperature. If the quantity is too small to provide adequate distribution and ventilation, it may be increased by using a lower entering temperature. Air motion has a cooling effect on the individual, and ordinarily the air quantity circulated should provide an overall air change in the conditioned space in not less than 5 min or more than 12 min. Best results are secured when the entering air temperature and method of distribution permit uniform mixture of air without excessive motion in the occupied zone.

After air temperatures are established, the heat loss from ducts may be approximated (see Chapter 42). The resulting temperature drop can be calculated by solving Equation 2.

$$\Delta t = \frac{H_d}{60\,d \times 0.24 \times Q} \tag{2}$$

where

Δt = temperature drop, degrees Fahrenheit.
H_d = heat loss in duct, Btu per hour.

Unless the duct passes through an unheated portion of the building, heat loss from ducts can frequently be neglected. Final decision rests on careful analysis of local conditions.

The duct distribution system is designed using velocities as recommended in Chapter 31, and grille locations as discussed in Chapter 30. In most installations it is advisable in order to permit economical heating prior to occupancy to design the return duct system of sufficient area to convey 100 per cent of the air handled by the fan. Also in mild weather certain economies of operation may be affected by designing the outside air duct of sufficient area to convey approximately the total quantity of air handled by the fan and means should be provided for the escape of this air quantity. In every case, however, the outside air duct must be of sufficient area to pass minimum ventilation air.

Air Distribution System for Cooling

The total cooling and dehumidifying load to be supplied by the central system is determined from the several components of the design load listed under *Item 3*. The entering air temperature is determined by selecting the proper relationship between the quantity of air to be handled, the heat gain in the conditioned space, and the location of the air inlets. In cooling applications it is desirable that the difference between the temperature of air currents in the space frequented by occupants and the average temperature in such space be not greater than 2 F for air velocities of 40 linear feet per minute and over and not greater than 3 F for velocities of less than 40 linear feet per minute.

There is a fairly wide range of permissible entering air temperatures. With high velocity jets or diffusing nozzles, located at some distance from the occupied space, entering air temperatures may be as much as 30 F below room temperature. Where the air is introduced through supply inlets fairly close to the occupied zone the entering air should be within 10 to 15 F of a desired room temperature. The problem of preventing drafts in summer air conditioning is important as air, cooler than room air, tends to fall without diffusing and proper design must consider the relationship between temperature and diffusion to secure satisfactory results.

The relation between heat gain, air quantity, and air temperatures is given by Equation 1. For cooling, t_1 and t_2 are reversed. At this stage in the design, the only known quantities are H and t_1. The maximum and minimum limits for Q are the same as given for heating, and t_2 may be from 10 to 30 F below t_1, depending on room size and shape and type and location of supply grilles.

Moisture added within the conditioned space does not increase the dry-bulb temperature in the space, and therefore only the total sensible heat gain to the space is used for H in Equation 1. Any reasonable value may be assigned to t_2 and a trial calculation made. The value of Q obtained should lie within the proper limits; and if coils are to be used, the

air quantity should be investigated to determine the most economical coil size.

To absorb the moisture load, the supply air must enter the space with a lower dew-point than that desired in the conditioned space. A method for determining this dew-point reduction follows:

Total all the latent heat gains in the room and convert them to equivalent grains of moisture. Divide the total grains of moisture by the number of pounds of air delivered to the room which will give the difference in weight of moisture between the entering air and room air conditions. Subtract this amount from the grains of moisture corresponding to the dew-point temperature in the room and refer to psychrometric charts or tables to establish the required dew-point temperature of the entering air. The intersection of this new dew-point condition with the dry-bulb temperature line of the entering air at the supply inlet to the room will establish the entering wet-bulb temperature condition.

When the supply duct passes through an unconditioned space, the dry-bulb temperature rise in the duct should be estimated (see Chapter 42). This temperature rise usually ranges from 1 to 3 deg and is deducted from t_2 to establish dry-bulb temperature leaving conditioner. Ducts intended for low temperature air must frequently be insulated to prevent condensation on outer surfaces, even though heat saving is not large enough to justify the cost of covering.

It is seldom that standard equipment will produce exactly the combination of dry-bulb and dew-point temperatures indicated by preceding calculations, and it becomes necessary to revise assumptions regarding grille temperature, air quantity, refrigerant temperature, and air velocity through conditioner until the proper balance is obtained. When such alterations fail to produce desired results, reheating is indicated. Reheating methods have been previously mentioned in this chapter.

The design of a duct distribution system for cooling is accomplished in the same manner as that previously described for heating installations. For cooling spaces prior to occupancy, it is also desirable to design the return air ducts of sufficient area to convey 100 per cent of the air handled by the fan and the same recommendations with regard to the outside air duct as referred to in the heating design would be applicable for summer air conditioning.

CORRELATION OF SUMMER AND WINTER DESIGN

Frequently the quantity of air required for the central system in summer conditioning is considerably greater than the quantity required for winter conditioning. In practice, volume control should be provided using a speed regulator on the fan, or dampers, so that the air quantity may be changed for the cooling and heating cycles. Sometimes a recalculation using different entering air temperatures will permit using the same quantity of air all year round. There is no fixed rule or method for determining the most practical design for air quantity and the engineer should use discretion to a large extent in working out a balanced system.

If the system is to be used for heating only, good results will be obtained with supply grilles located in the baseboard. For cooling systems, it is better to locate them in sidewalls above heads of occupants or in the ceiling. This same location will be satisfactory for year-round systems, especially if the supply temperature in winter does not exceed 100 to 110 F. Air distribution is discussed in more detail in Chapter 30.

SELECTION OF EQUIPMENT

The type of system to be used is controlled by the existing load conditions, the degree of perfection expected from the system, and the designer's knowledge of the operating characteristics of the various types of equipment. Information about equipment is available in other chapters and in the *Catalog Data Section.*

It is considered good practice to provide for total recirculation in order to reduce operating costs during unoccupied periods. And it is also less expensive to use all outdoor air for cooling when the outdoor wet-bulb temperature is lower than that of the return air.

The process of evaporative cooling may, in some localities, provide interior conditions which are adequate for certain occupancies. Outdoor air is drawn through a spray of recirculated water which is not mechanically cooled. The equilibrium temperature for heat exchange is the wet-bulb temperature of the entering air. In actual washers (see Chapter 26), the dry-bulb temperature does not fall to the wet-bulb temperature but only approaches it. To get much cooling effect, it is necessary to have a large differential between dry- and wet-bulb temperatures of outdoor air. Hence, this method is only feasible in those sections where the air temperature is high and the relative humidity is low. Also because the wet-bulb temperature of the air does not change in passing through the spray, this method is not recommended for cooling loads with a high proportion of latent heat.

In making the selection between spray and surface dehumidifiers, certain characteristics of each should be considered. A spray dehumidifier, (*i.e.*, dehumidifying air washer) will deliver practically saturated air, the temperature of which is determined by the temperature of the air washer water. The control in this case is the control of the water temperature and cooling effect can be accomplished by an external water cooler, a natural cold water supply, or coils installed directly in the washer. The washer system is used in the winter time for humidifying in the same way; that is, by controlling the water temperature, water can be evaporated into the air. This may require preheating the air before entering the washer, or the use of an external water heater, or steam coils directly in the washer. Air washers also have the ability to eliminate certain kinds of dirt and dissolved gases and some odors.

In common practice, the air leaving a cooling coil is not saturated. But as most comfort-conditioning systems require some differential between delivery dry-bulb and dew-point temperatures, it frequently happens that by careful selection the right combination can be produced by the coil, eliminating the need of reheating. Strictly speaking, the desired performances will be obtained only at full load, there being some fluctuation in room relative humidity at light loads. Such variation is usually within the allowable limits for comfort work. It is essential that a good filter be placed ahead of a cooling coil, otherwise sufficient dust will soon adhere to the wet coil to completely block the air passage.

The relation between sensible and latent heat is of great importance in the performance of cooling coils. Unfortunately, no uniformity exists at present in stating this ratio; and the several possible methods are given in Table 1 together with typical values for various occupancies. It must be

remembered that these values are typical and should in no case be arbitrarily assumed to be correct for any given problem.

The procedure for selecting both heating and cooling coils is given in Chapter 25, and coil performance tables may be found in the manufacturers' catalogs. Some manufacturers have standardized on 4-row cooling coils and offer them at lower prices per square foot of surface than for other depths. By changing face velocity within permissible limits, 4-row coils can be adapted to many cooling jobs.

The characteristics of washers are discussed in Chapter 26. Dimensions will be found in manufacturers' catalogs.

TABLE 1. ROOM HEAT LOAD RATIOS FOR TYPICAL SUMMER COMFORT CONDITIONING

Room Heat Load Ratios[a]	Typical Classes of Room Service or Load				
	No. Occupants or Sources of Vapor	Private Office or Residence	Restaurant or Crowded Office	Auditorium at Capacity or Crowded Restaurant	Ballroom at Capacity
Sensible Heat / Total Heat	1.00	0.90	0.80	0.70	0.60
Total Heat / Sensible Heat	1.00	1.11	1.25	1.43	1.67
Latent Heat / Total Heat	0	0.10	0.20	0.30	0.40
Total Heat / Latent Heat	....	10.00	5.00	3.33	2.50
Sensible Heat / Latent Heat	....	9.00	4.00	2.33	1.50
Latent Heat / Sensible Heat	0	0.11	0.25	0.43	0.67

[a]The overall heat load ratio for the dehumidifier will be different from the heat load ratio for the room. The extent of the difference will depend on the quantity and condition of the outside air used, upon the magnitude of the duct losses, and upon whether or not reheat or by-pass is used.

If coils are used, humidification may be accomplished in winter by use of a separate humidifier. On small installations, the simplest method is to use a warm water spray through atomizing nozzles, using a pressure of 15 to 25 lb and sufficient nozzles to atomize about twice the amount of water needed for humidifying. The grains of moisture to be added to the incoming air at outside design temperature to bring it up to the required room dew-point are calculated. This is converted to total pounds of water per hour for the system and sprays designed for twice this amount. Cold water will not vaporize as completely as warm water, and water temperatures from 120 to 150 F are commonly used. Another method is to install a water tank in the air stream with a steam coil submerged in the tank, the humidification being accomplished by the steam boiling the water into vapor, which in turn will be absorbed by the passing air. In both of these systems there is a tendency to deposit lime and other impurities on any surface the water touches, and this scale should be

cleaned regularly before it becomes excessive. Water spray should not touch the steam heating coils as they will quickly become coated with scale. The preferred practice is to install sprays between coils and eliminator plates.

Sound Control, Filters, Fans and Motors

Problems of sound control should be jointly considered by the acoustical and air conditioning engineer for satisfactory results. Many installations require noise levels which are relatively low and for that reason equipment must be selected having a very low noise rating. In central systems consideration should also be given to the lining of ducts for the reduction of noise levels within an enclosure. Often reduced speeds of equipment and low air velocities are helpful in eliminating undesirable noise conditions. Information is given in Chapter 32 with regard to acceptable noise levels for various types of rooms and methods are outlined for computing length of duct lining materials.

For a discussion of air cleaning devices and for the selection of all types of air filtering equipment refer to Chapter 28. The selection of fans, motors and their control may be based on data available in Chapters 29 and 35.

Automatic Control

The control of an air conditioning system is very important. A simple comfort cooling or heating installation requires a minimum of control, whereas a more complex installation justifies a more complete control. In this connection, there are many patents allowed and pending on air conditioning equipment including control, and the designer should consider these factors in selecting equipment. Refer to Chapter 33.

Static Pressure

The static pressure against which the fan must operate is the sum of all pressure losses through all parts of the complete system. Resistances of equipment such as coils, washers, filters, and grilles are obtained from data given in the manufacturers' catalogs. Pressure drop in the duct is calculated as shown in Chapter 31.

SPECIAL CONSIDERATIONS

In designing a central system for air conditioning there are a number of special considerations not referred to previously which must be considered. Certain features of building constructions are important. The building must be suitable in construction so that the desired conditions can be maintained economically.

For instance, excessive sun load on roofs or glass windows may not only cause excessive heat gain to be absorbed by refrigeration, but direct sun radiation heating converts a surface into a panel heater. This radiant heat is not absorbed until it strikes another mass such as a building wall, or a person. It is therefore possible to have comfortable air conditions surrounding a person, and yet have him uncomfortably warm from radiant heat from a hot wall, window, or ceiling near him. As another example, it is much better to provide hoods over steam tables, coffee urns, etc., than to try to remove this heat by mechanical refrigeration.

Another problem is presented with winter conditioning for maintaining satisfactory relative humidity. If 30 to 40 per cent is desired at all times, no matter how cold it is outside, excessive condensation may collect on

single glass windows, or on hardware which connects to the outside such as latches and hinges. Condensation on windows can sometimes be prevented by applying a small amount of local heat under the window. In other cases double glass or storm window construction is used. Refer to Chapters 3 and 5 for condensation temperatures.

In many cases, different zones of a building will require entirely different treatment. In some buildings where offices are exposed on all four sides to sun and wind effect, cooling is required on the sunny side and heating is required on the shady side simultaneously, in spring and fall seasons. The central system must be carefully zoned to apply cooling or heating as they may be needed for each zone, independent of other zones.

In many existing buildings, the central system will be added to a radiation system of heating. The designer should take full avantage of this radiation for heating the outside walls and windows. At the same time, it must be controlled to prevent overheating the whole system. A few uncontrolled radiators will often change the heat balance and cause excessive overheating of the whole zone, without occupants near the radiators realizing the source of the trouble. All radiation used locally in connection with the control system should be equipped with automatic control to prevent overheating.

The apparatus should be placed for minimum piping and duct work but it must be accessible for maintenance, repair, and cleaning. The air conditioning unit should be designed to provide access for cleaning coils, drip pans, eliminators and for very easy maintenance on filters. This equipment will be in operation for many years, probably for the life of the building, and a little thought spent in the plan will simplify maintenance and assure successful results.

Lastly, an air conditioning system should be designed from the standpoint of safety so that the fire hazard will be kept at a minimum, and the designer should acquaint himself with existing local regulations.

Example 1. With the assumed values as indicated, perform the essential calculations to determine the design loads for heating and cooling and the necessary factors for designing the distribution system.

Solution: Design Conditions.

Outside air dry-bulb, winter	0 F
Outside air dry-bulb, summer	95 F
Outside air wet-bulb, summer	75 F
Inside air dry-bulb, winter	72 F
Inside air wet-bulb, winter	56 F
Inside air relative humidity, winter	35 per cent
Inside air dry-bulb, summer	80 F
Inside air wet-bulb, summer	66.7 F
Inside air relative humidity, summer	50 per cent

200 people—4 kw light load.

Design Heating Load.

Sensible heat loss through walls, etc.	200,000 Btu per hour
Outside air—2000 cfm $\times$ 0.075 $\times$ 60 $\times$ 0.24 $\times$ (72 − 0) =	155,500 Btu per hour
Humidification— $\dfrac{2000 \times 0.075 \times 60 \times (40-5) \times 1040}{7000}$ =	46,900 Btu per hour
Heat loss through ducts	33,200 Btu per hour
Heat gain from lights, etc.	Disregard
Total heating load	435,600 Btu per hour

Design Cooling and Dehumidifying Load.

	Sensible	Latent
Heat gain through walls, etc.	128,350	
Heat gain from occupants (200)	44,000	36,000
Heat emission from appliances	2,000	
Heat gain from lights (4 kw)	13,650	
Heat gain from solar radiation	12,000	
Totals	200,000	36,000 Btu per hour
Outside air:		
$2000 \times 0.075 \times 60 \times 0.24 \times (95 - 80) =$	32,350	
$\frac{2000 \times 0.075 \times 60 \times (96 - 77.2) \times 1060}{7000} =$		25,600
Heat gain through ducts	22,200	
Totals	254,550	61,600 Btu per hour

Total Cooling Load = 316,150 Btu per hour

Tons Cooling Effect = 26.4

Ratio S.H. to T.H. $= \frac{200,000}{236,000} = 0.85$ (Room) $= \frac{254,550}{316,150} = 0.805$ (Conditioner)

Design Distribution System for Heating

a. Total heat loss in space = 200,000 Btu per hour

Assume grille temperature = 90 F

$$Q = \frac{200,000}{60 \times 0.075 \times 0.24 \times (90 - 72)} = 10,300 \text{ cfm}$$

b. Total heat loss in ducts between unit and grilles = 33,200 Btu per hour

$$\Delta t = \frac{33,200}{60 \times 0.075 \times 0.24 \times 10,300} = 3 \text{ F}$$

c. 90 F plus 3 F = 93 F temperature leaving coil

Design Distribution System for Cooling

a. Total sensible heat gain in space = 200,000 Btu per hour

Assume grille temperature = 62 F

$$Q = \frac{200,000}{60 \times 0.075 \times 0.24 \times (80 - 62)} = 10,300 \text{ cfm} = 770 \text{ lb per minute}$$

b. Latent heat gain in space = 36,000 Btu per hour = 600 Btu per minute

$$= \frac{600 \times 7000}{1060} = 3970 \text{ gr per minute}$$

3970 ÷ 770 = 5.2 gr per pound air supplied

Room condition = 80 F DB, 50% RH, 60 F DP = 77.2 gr per pound

Gain in conditioned space = 5.2 gr per pound

Specific humidity leaving conditioner = 72.0 gr per pound

Grille temperature = 62 F DB and 58 F DP

c. Duct gain = 22,200 Btu per hour

$$\Delta t = \frac{22,200}{60 \times 0.075 \times 0.24 \times 10,300} = 2 \text{ F temperature rise}$$

Leaving coil = 60 F DB, 58.7 F WB, 58 F DP

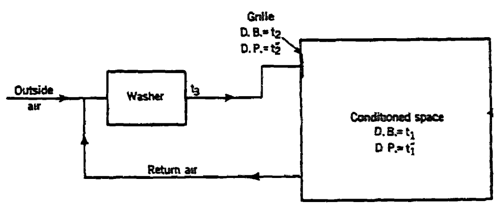
These calculations are based on maximum load conditions as set forth in the design. For intermediate loads the calculations may show entirely different relationship. For example, the entering air temperature will approach the space temperature as the sensible heat gain or loss decreases, due to outside temperature change, entrance or exit of people, use of artificial lighting, direction and intensity of sun's rays. The entering dew-point will remain much more uniform as it is affected by changes in room moisture

gain only and this does not fluctuate greatly. If intermediate load conditions are important the calculations should be repeated for those loads.

Example 2. Determine the cooling load for a theater if the following design conditions are assumed. A dehumidifying air washer is to be used with and without the by-pass.

Outside air dry-bulb	94 F
Outside air wet-bulb	75 F
Inside air dry-bulb	80 F
Inside air wet-bulb	67 F
Minimum outdoor air	6300 cfm
People	600
Lights	4 kw
Transmission gain	110,000 Btu per hour

Solution. Without By-Pass.



where

t_1 = dry-bulb temperature, room or return air, degrees Fahrenheit.
t_2 = dry-bulb temperature at supply grille, degrees Fahrenheit.
t_3 = saturation temperature leaving washer, degrees Fahrenheit.
t_1'' = dew-point in room, degrees Fahrenheit.
t_2'' = dew-point at supply grille, degrees Fahrenheit.
h_1 = enthalpy of room air, Btu per pound.
h_0 = enthalpy of outside air, Btu per pound.
h_3 = enthalpy of air leaving washer, Btu per pound.

Design Cooling Load.

	Sensible	Latent
Transmission	110,150	
People (600)	132,000	720,000
Lights (4 kw)	13,650	
Totals	255,800 Btu per hour	720,000 gr per hour

Determine Air Quantity

Sensible heat gain in space			=	255,800 Btu per hour
Assume grille temperature			=	68 F
$Q = \dfrac{255{,}800}{60 \times 0.24\,(80 - 68)}$	=	1,480 lb per min	=	19,700 cfm
Minimum outdoor air	=	470 lb per min	=	6,300 cfm
Return air	=	1,010 lb per min	=	13,400 cfm
Moisture gain in space	=	720,000 gr per hour	=	12,000 gr per min

12,000 ÷ 1,480 = 8.11 gr per pound air supplied

Room conditions = 80 F DB, 67 F WB, 60 F DP	=	77.21 gr per lb
Gain in conditioned space	=	8.11 gr per lb
Specific humidity leaving conditioner	=	69.1 gr per lb

Grille conditions = 68 F DB and 57 F DP

Determine Cooling Load on Air Washer

Heat removed, outside air $= 470\ (h_0 - h_3) = 470\ (38.46 - 24.4) = 6600$ Btu per min
Heat removed, return air $= 1010\ (h_1 - h_3) = 1010\ (31.51 - 24.4) = 7180$ Btu per min

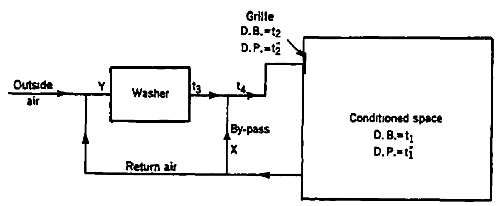
Total 13,780 Btu per min

Refrigerating effect $= 13{,}780 \div 200 = 68.9$ tons.

Determine Reheating Required

$$H = Wc_p\ (t_2 - t_3)$$
$$= 1480 \times 0.24\ (68 - 57) = 3910 \text{ Btu per minute} = 234{,}600 \text{ Btu per hour.}$$

Solution. With By-Pass



where

x = air to be by-passed, pound per pound total air.
y = air passed through washer, pound per pound total air.
t_4 = dry-bulb temperature after mixing with by-passed air, degrees Fahrenheit.
$t_4 = t_2$ if no temperature rise in duct is assumed.

Determine Amount of By-Pass Air and Saturation Temperature Leaving Washer[4]

$$\begin{aligned} t_1x + t_3y &= t_4 \\ t_1''x + t_3y &= t_2'' \\ 80x + t_3y &= 68 \\ 60x + t_3y &= 57 \\ \hline 20x &= 11 \\ x &= 0.55 \\ y &= 1 - 0.55 = 0.45 \\ 80 \times 0.55 + 0.45t_3 &= 68 \\ t_3 &= 53.3 \text{ F} \end{aligned}$$

Determining Cooling Load on Washer

Air through washer $= 0.45 \times 1480$ = 665 lb per min
Outside air = 470 lb per min

Return air through washer = 195 lb per min
Heat removed, outside air $= 470\ (h_0 - h_3) = 470\ (38.46 - 22.13) =$ 7680 Btu per min
Heat removed, return air $= 195\ (h_1 - h_3) = 195\ (31.51 - 22.13) = 1830$ Btu per min

Total 9510 Btu per min

Refrigerating effect $= 9510 \div 200 = 47.55$ tons
Reheating—None.

Note: The Revised Bulkeley Psychrometric Chart and Table 6, Chapter 1 were used in the solution of these problems.

[4]For derivation of these simultaneous equations see page 190 of THE GUIDE 1937.

Chapter 21

UNIT HEATERS, UNIT VENTILATORS, UNIT HUMIDIFIERS

Unit Heaters, Ratings, Unit Ventilators, Applications, Window Ventilators, Unit Humidifiers, Types of Units

IN other chapters, descriptions are given of heating, cooling, ventilating humidifying, and dehumidifying systems. The success of such systems has led to the production of factory-assembled equipment employing a majority of the principles of these complete systems. As a result, present day practice involves the use of unitary equipment in the majority of installations where capacity and application demands are within the limits of such units. Thus unit heaters, unit ventilators, and unit humidifiers described in this chapter, and cooling units, unit air conditioners, and attic fans described in Chapter 22 have come to occupy a place of their own in the industry.

Unitary Equipment

Unitary equipment was first applied in units of small capacity but increased experience in this field has led to an ever widening range of capacities and applications. In general a *unit* may be defined as a factory-made encased assembly of the functional elements indicated by its name, such as unit heater, unit ventilator, etc. These units are shipped substantially complete or built and shipped in sections so that the only field work necessary is the assembling together of the sections, without resorting to any field fabrication.

A unit may be complete in itself, employing its own direct means of air distribution and source of heating, in which case it thus represents a complete self-contained unit. Or it may be coupled with separate means of air distribution such as duct work and outlets, in which case it will still be considered as a unit system, as contrasted with the generally accepted term of a field fabricated central station system. The manufacturer of the unit is responsible for the output and performance of the unit under rated conditions, whereas the contractor installing the complete unitary system is normally held responsible for the performance of the complete system.

Unit equipment justifies its existence due to the following features:

1. Lower cost per unit capacity. Standardized design and volume production makes possible low cost factory assembly thereby eliminating individual design and handling of every part for each installation.

2. Flexibility and mobility of equipment. Unitary equipment can be readily located in existing buildings without the necessity of running large ducts through floors and many partitions. Such equipment can be shifted to meet changing requirements. Tenants may obtain the advantages of conditioning when the entire building is not equipped with a conditioning system. In industrial process work, the flexibility of unitary equipment is also advantageous.

3. Lower installation costs. The fact that the equipment arrives on the job in an assembled condition, coupled with the lesser problems of duct work and connecting piping, materially reduces installation costs.

4. Small capacities. The small capacities available in unitary equipment have brought the advantages of controlled air conditions to a number of small offices, stores, shops, and individual rooms where specially designed and built central system equipment would have been uneconomic.

Definitions

With the growth of the unit equipment industry, it becomes increasingly evident that there is no sharp line of demarcation, on the basis of capacity, between a unit and a central plant system. The definitions contained in a code, Standard Method of Rating and Testing Air Conditioning Equipment[1], have helped to clarify and identify the various types of equipment since the definitions are given on a purely functional basis. The following definitions are taken from the code:

1. A *Heating Unit* is a specific air treating combination consisting of means for air circulation and heating within prescribed temperature limits.

2. A *Heating Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for heating and maintaining humidity within prescribed limits.

3. A *Humidifying Unit* adds water vapor to and circulates air in a space to be humidified.

4. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

5. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

UNIT HEATERS

A unit heater consists of the combination of a heating element and fan or blower having a common enclosure and placed within or adjacent to the space to be heated. Generally no ducts are attached to inlets or outlets, although it is common practice with many unit heater applications to equip the heaters with directional outlets or adjustable louvers. While unit heaters are designed primarily to handle all recirculated air, they may be installed to handle either partial or total outdoor air.

Features

A wide variety of structural designs is available. All employ some form of heat transfer surface, supplied with steam, hot water, gas or electric heat. Air is always forced over or drawn through the heat transfer surface by a fan of either the propeller or centrifugal type. Heating surfaces may be in the form of non-ferrous or steel pipe coils, non-ferrous or steel pipe with extended surfaces, cast-iron, or pressed or built-up sections of the cartridge or automotive type.

[1]Prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association* and *Air Conditioning Manufacturers' Association*.

Compared with the older method of heating by radiation, properly designed and applied unit heaters should:

1. Circulate air in the building at a rapid rate but without objectionable draft.
2. Reduce the temperature differential between the floor and ceiling.
3. Direct the heated air so that uniform temperature distribution will be obtained throughout the heated space.
4. Prevent or remove the cold stratum of air commonly found at the floor level.
5. Reduce the number of heating elements required and thereby decrease the cost and extent of the piping necessary.
6. Maintain a closer control of room temperature either manually or by means of simple thermostats.
7. Produce an economy in heating costs resulting from the sum total of the above advantages.
8. Provide a means of saving floor area or room space due to the compactness of the equipment and flexibility of application.

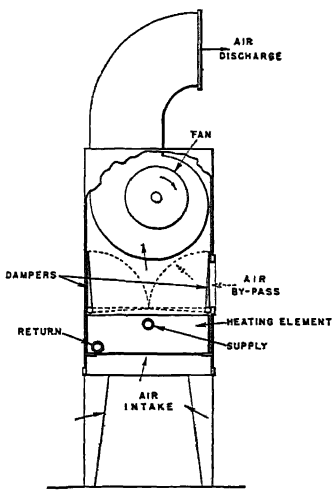


FIG. 1. FLOOR MOUNTED UNIT HEATER, HOUSED TYPE FAN

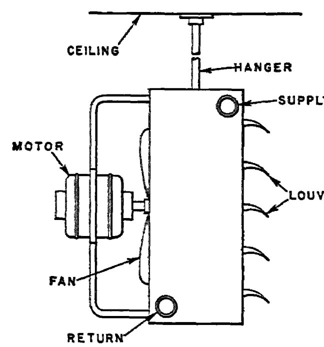


FIG. 2. SUSPENDED UNIT HEATER, PROPELLER TYPE FAN

Types of Units

There are two major types of unit heaters, centrifugal housed fan type and propeller fan type. The housed fan high velocity (1500 to 2500 fpm) discharge units, with outlets adjustable to deliver air in several directions, are able to project their heating effect over distances of from 30 ft to as much as 200 ft from the unit. This makes possible the location of these units at considerable distances from each other, thus reducing greatly the piping and loss of floor space due to the heating equipment. Figs. 1 and 3 illustrate the housed fan type of unit.

Propeller fan type of unit heater is used extensively to heat the small commercial establishment, although this type of unit is also available in very large sizes. Fig. 2 illustrates a propeller fan type of heater. A code[2] governing the number of sizes of propeller fan type units as well as

[2]Standards for Propeller Type Unit Heaters prepared and adopted by the *Industrial Unit Heater Association*, June, 1938.

standardization of fan motor types and the method of specifying outlet velocities has been adopted.

Ratings

Standard practice is to rate unit heaters in Btu per hour at a given temperature of air entering the heater and at a given steam pressure maintained in the coil. Steam at 2 lb pressure and air entering at 60 F are used as the standard basis of rating[3]. The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heating capacity for any condition of steam pressure and entering air temperature may be calculated approximately from any given rating by the use of factors in Tables 1 and 2. Table 1 is used for the blow-through type and Table 2 for the draw-through type of unit. The formulae given under *unit ventilators* for calculating capacities also apply to unit heaters.

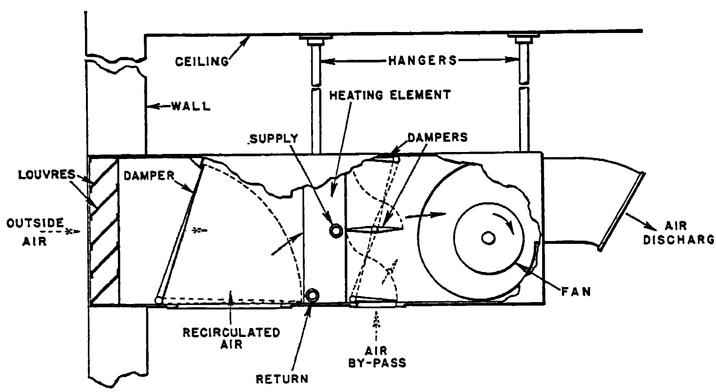


FIG. 3. SUSPENDED TYPE UNIT HEATER, HOUSED TYPE FAN

The temperature to be maintained[4] in the room, for recirculating heaters with intakes at the floor level, should be considered as the temperature of the air entering the heater. Where outside air is introduced, the temperature of the mixture must be calculated and used as the entering air temperature to the heater. Unit heaters taking in recirculated air at the floor level should maintain temperature differentials of less than 0.5 deg per foot of elevation when the maximum capacity of the heaters is required. This temperature difference per foot of elevation is less than the corresponding variations for spaces heated by direct radiation.

The temperature variation from floor to ceiling with suspended unit heaters taking air at some distance above the floor may reach as much as 1 deg per foot of elevation during the periods when the maximum capacity of the heaters is required. Thus this allowance should be made in calcu-

[3]A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165).

[4]A.S.H.V.E. RESEARCH REPORT No. 958—Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson, and O. C. Cromer (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 243).

A.S.H.V.E. RESEARCH REPORT No. 1011—Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson, and John James (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

lating the capacity of suspended heaters. High velocity discharge units (blower type illustrated in Fig. 3) will maintain slightly lower temperature differences than will low velocity units (propeller fan type illustrated in Fig. 2).

Unit heaters are customarily rated as free delivery type units. If outside air intakes, filters, or ducts on the discharge side are used with the heater, proper consideration should be given to the reduction in air and heating capacity that will result because of this added resistance.

The percentage of this reduction in capacity will depend upon the charactertistics of the heater and on the type, design and speed of the fans so that no specific percentage reduction can be assigned for all heaters at a given added resistance. In general, however, disc or propeller fan type units will experience a larger reduction in capacity than housed centrifugal fan units for a given added resistance and a given heater will have a larger reduction in capacity as the fan speed is lowered. When confronted with this problem the ratings under the conditions expected should be secured from the manufacturer.

Boiler Capacity

The capacity of the boiler should be based on the rated capacity of the heaters at the lowest entering air temperature that will occur, plus an allowance for line losses. Ordinarily for recirculating heaters the lowest entering temperature will occur at the beginning of the heating period and is usually taken as 40 F, while for ventilators taking air from outdoors the lowest entering temperature will be the extreme outdoor temperature expected in the district. No greater allowance in boiler capacity beyond the calculated heat demand need be added in order to supply unit heaters than for any other type of system.

It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attendance to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, however, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller heating units instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their coils are designed for the purpose and when proper provision is made for returning the condensate. If ventilators are to take in air that may be at a temperature below freezing, however, a steam pressure of not less than 5 lb should be maintained on the convector or a corresponding differential in pressure between the supply and returns be maintained by means of a vacuum.

Piping Connections

Piping connections for unit heaters are similar to those for other types of fan blast heaters. The piping around the unit heaters must strictly conform to the system requirements while at the same time permitting the heaters themselves to function as intended. The basic piping principles for steam systems are discussed in Chapter 14.

TABLE 1. CONSTANTS FOR DETERMINING THE CAPACITY OF Blow-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In.	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1.446	1.369	1.273	1.191	1.110	1.034	0.956	0.881	0.809	0.739	0.671
2	1.585	1.495	1.405	1.320	1.237	1.155	1.078	1.000	0.926	0.853	0.782	0.713
5	1.640	1.550	1.456	1.370	1.289	1.206	1.127	1.050	0.974	0.901	0.829	0.760
10	1.730	1.639	1.545	1.460	1.375	1.290	1.211	1.131	1.056	0.982	0.908	0.838
15	1.799	1.708	1.614	1.525	1.441	1.335	1.275	1.194	1.117	1.043	0.970	0.897
20	1.861	1.769	1.675	1.584	1.498	1.416	1.333	1.251	1.174	1.097	1.024	0.952
30	1.966	1.871	1.775	1.684	1.597	1.509	1.429	1.346	1.266	1.190	1.115	1.042
40	2.058	1.959	1.862	1.771	1.683	1.596	1.511	1.430	1.349	1.270	1.194	1.119
50	2.134	2.035	1.936	1.845	1.755	1.666	1.582	1.498	1.416	1.338	1.262	1.187
60	2.196	2.094	1.997	1.902	1.811	1.725	1.640	1.555	1.472	1.393	1.314	1.239
70	2.256	2.157	2.057	1.961	1.872	1.782	1.696	1.610	1.527	1.447	1.368	1.293
75	2.283	2.183	2.085	1.990	1.896	1.808	1.721	1.635	1.552	1.472	1.392	1.316
80	2.312	2.211	2.112	2.015	1.925	1.836	1.748	1.660	1.577	1.497	1.418	1.342
90	2.361	2.258	2.159	2.063	1.968	1.880	1.792	1.705	1.621	1.541	1.461	1.383
100	2.409	2.307	2.204	2.108	2.015	1.927	1.836	1.749	1.663	1.581	1.502	1.424

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

TABLE 2. CONSTANTS FOR DETERMINING THE CAPACITY OF Draw-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 lb Gage and Entering Air Temperature of 60 F)

STEAM PRESSURE LB PER SQ IN.	TEMPERATURE OF ENTERING AIR											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.483	1.405	1.329	1.253	1.178	1.105	1.032	0.962	0.892	0.822	0.754	0.688
2	1.520	1.442	1.363	1.290	1.215	1.141	1.069	1.000	0.930	0.861	0.792	0.728
5	1.565	1.485	1.410	1.334	1.260	1.187	1.114	1.045	0.975	0.906	0.838	0.771
10	1.637	1.558	1.480	1.403	1.328	1.253	1.182	1.112	1.042	0.973	0.903	0.838
15	1.688	1.610	1.533	1.458	1.382	1.310	1.239	1.168	1.099	1.028	0.960	0.895
20	1.728	1.649	1.572	1.498	1.421	1.350	1.278	1.208	1.138	1.070	1.002	0.936
30	1.803	1.725	1.648	1.572	1.497	1.423	1.352	1.281	1.212	1.145	1.078	1.010
40	1.864	1.787	1.710	1.637	1.563	1.491	1.420	1.350	1.282	1.215	1.148	1.081
50	1.927	1.850	1.773	1.700	1.628	1.554	1.483	1.416	1.347	1.278	1.211	1.145
60	1.973	1.897	1.820	1.748	1.673	1.601	1.531	1.463	1.394	1.325	1.260	1.194
70	2.018	1.943	1.869	1.795	1.722	1.651	1.582	1.512	1.443	1.377	1.310	1.243
75	2.043	1.970	1.895	1.822	1.750	1.680	1.609	1.540	1.471	1.402	1.333	1.268
80	2.064	1.988	1.914	1.841	1.770	1.698	1.629	1.560	1.491	1.422	1.354	1.288
90	2.102	2.028	1.951	1.878	1.804	1.732	1.661	1.590	1.523	1.457	1.387	1.321
100	2.150	2.071	1.994	1.919	1.845	1.770	1.700	1.630	1.560	1.492	1.425	1.359

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

Rapid condensation of steam, especially during heating-up periods, is characteristic of this type of equipment. The piping must be planned to accommodate this rapid condensation, must keep the surfaces free of water; while on the supply side the piping must be ample to carry a full supply of steam to the surfaces to take the place of that condensed.

Adequate size of pipe is thus essential to all heating surfaces over which there is a forced flow of air. Especially is this true where the fan is operated under start-and-stop control and where the air handled may be made up either wholly or partly of cold air from outside the building. In such installations the condensation rate may vary rapidly and the necessity for ample pipe capacity is especially acute.

A method of connecting a unit heater to a one-pipe gravity system is illustrated in Fig. 4. In those cases where the unit heater is to be con-

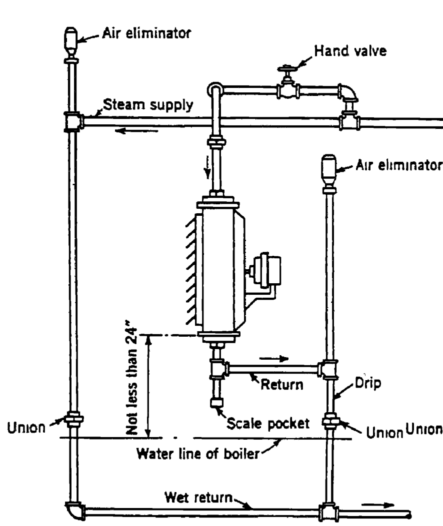


FIG. 4. UNIT HEATER CONNECTION TO ONE-PIPE GRAVITY STEAM SYSTEM

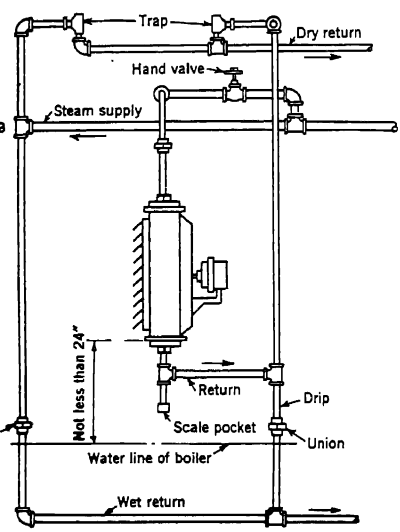


FIG. 5. UNIT HEATER CONNECTION TO GRAVITY SYSTEM WITH WET AND DRY RETURNS

nected to a dry return instead of a wet return it is necessary to provide a water pocket or loop about 5 ft in depth to prevent steam passing into the return and thus into other equipment.

A method of connection is shown in Fig. 5, where there is a wet return and a dry return. In this case the condensate from the heater and the drip from the supply main drop to the wet return by gravity, while the air passes upward through the traps to the dry return and is vented from the system at any suitable location.

A sketch of an arrangement where there is a dry return line through which both air and condensate pass to be handled by some suitable means, such as a condensate pump and receiver is given in Fig. 6. The return line is not subjected to vacuum, and consequently all arrangements must facilitate gravity flow of the condensate toward the receiver. Traps must pass air and condensate rapidly to keep the return piping only partially full of water.

Since unit heaters are often constructed with sufficient strength to resist high pressures, use of high pressure steam in them is a common practice. In Fig. 7 the condensate and air reach the return overhead through traps, and check valves are located in the return piping.

For two-pipe closed gravity return systems, the return from each unit should be fitted with a heavy duty or blast trap, and an automatic air valve should be connected into the return header of each unit. Pressure drop must be compensated for by elevation of the heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns the same as for vacuum systems and the entire return system be free-vented to the atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

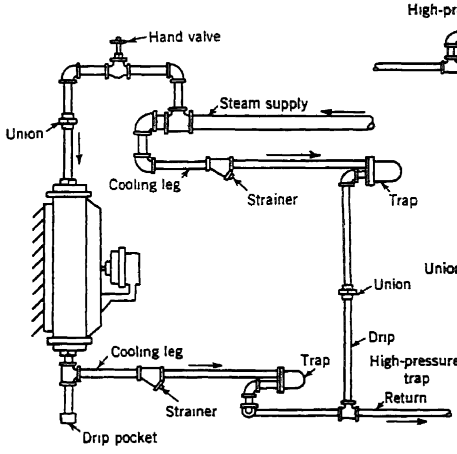


FIG. 6. UNIT HEATER CONNECTION FOR VACUUM OR VAPOR SYSTEM DISCHARGING CONDENSATION INTO DRY RETURN

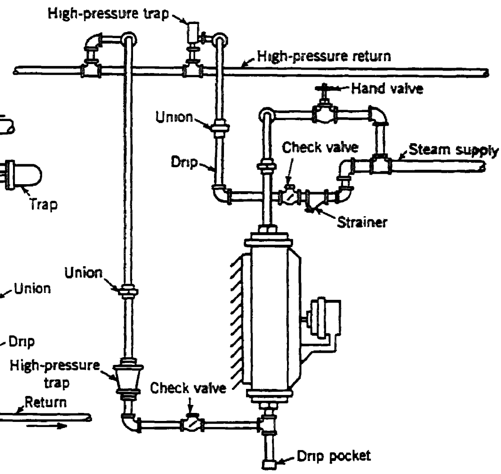


FIG. 7. METHOD OF CONNECTING UNIT HEATER TO HIGH PRESSURE RETURN

On vacuum or open vented systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, with some danger of return pipe corrosion and the problem of its elimination at some other point in the system.

Application

Unit heaters are used principally for commercial and industrial applications such as display rooms, garages, factories, factory offices and to

some extent for office applications where appearance is not a major factor.

Unit heaters may be adapted to a number of industrial processes, such as drying and curing, with which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is given off. When such conditions are severe, it is necessary that the heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. (See discussion of condensation in Chapter 3.)

There are three major factors to consider in the application of unit heaters, as follows:

1. Location of Unit.
2. Air Distribution.
3. Heating Medium.

Heaters may be distributed through the central portions of a room discharging toward exposed surfaces, or may be spaced around the walls, discharging along the walls and inward as well, especially when there are considerable roof losses.

Suspended type units are located in an elevated position withdrawing air from this higher level and discharging the heated air down into the working zone. This type of installation is illustrated in Figs. 2 and 3. Suspended type units provide excellent temperature distribution.

In closely occupied spaces where direct air drafts into the working zone are not permitted, the floor mounted unit will give more uniform temperature distribution. These units draw the cold air from the floor and discharge the heated air above the working zone.

In general, it is better to direct the discharge from the unit heaters in such fashion that rotational circulation of the entire room content is set up by the system rather than to have the heaters discharge at random and in counter-directions.

Various types and makes of unit heaters are illustrated in the *Catalog Data Section* of this edition. Usually hot blasts of air in working zones are objectionable, so heaters mounted on the floor should have their discharge outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close to the head line as possible, yet not into the working zone. In general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

Low or high pressure steam as well as hot water are generally used in unit heaters. Direct-fired units are also available. Superheated steam can be satisfactorily used in unit heaters provided the capacity is based on saturated steam temperature and not on the total temperature. If

unusually high superheat is used, trouble may be experienced from the excessive expansion and contraction of the heating elements.

Electric, Direct-Fired, and Turbine-Driven Units

The foregoing discussion relates generally to units in which steam or hot water is used as the heating medium. Electric unit heaters are applied where electric power is abundant and cheap and where other forms of fuel are scarce and expensive. The low first cost, easy control, and inexpensive installation of this type of heating have also accounted for many other installations in which electricity has conveniently provided heat for short periods of time. (See Chapter 43.)

A recent development in gas burning equipment is the direct-fired industrial unit heater. These heaters are of the warm-air type and are equipped with fans which cause the air to pass over the heating surfaces at a fairly high velocity and then direct the warm air into the space to be heated. As is the case with the steam-fed unit heaters, the gas-fired appliances may be used for heating stores, shops, and warehouses. They usually are suspended in the space to be heated and in most instances leave the entire floor and wall area free for commercial use. Partial or complete automatic control also may be secured on appliances of this type. This type of heater is often used for temporary heat during building construction or where the installation of a steam or hot water plant is for some reason not justified. For permanent installations, it is usually advisable to provide an exhaust duct from the gas-fired unit heaters to remove products of combustion from the occupied space. While this is not necessary in large open industrial plants, in smaller closed rooms, it becomes essential.

Where high pressure steam is available it is sometimes used to drive a steam turbine direct-connected to the unit heater. The exhaust from this turbine, reduced in pressure, is then passed into the heating coil where it is condensed and returned to the boiler.

UNIT VENTILATORS[5]

Unit ventilators are similar in principle to unit heaters since ventilators incorporate an encased heating surface through which outside air is forced by means of a blower or fan and may or may not have provision for recirculation of air. While unit heaters are largely used for commercial and industrial applications, unit ventilators are intended primarily for schools, offices, and semi-commercial establishments. A typical unit ventilator is illustrated in Fig. 8.

Specifications

Unit ventilators usually consist of a semi-decorative cabinet containing the following necessary or optional parts:

1. Outside air inlet.
2. Inlet damper for closing the opening to the outside air inlet when the unit is not in use.

[5]A roof ventilator is sometimes termed a *unit ventilator*. For information on roof ventilators, see Chapter 41.

3. Adhesive or dry type filters for cleaning the air (optional).
4. A heating element usually of special design and intended for low pressure steam.
5. Motor and fan assembly.
6. Mixing chamber where warm and cold air streams are brought together.
7. Outdoor air inlet and recirculating air mixing damper (optional).
8. Discharge grille or diffuser.
9. Temperature control arrangement.

Functions and Features

The primary functions and features of a unit ventilator are:

1. To supply a given quantity of outdoor air for ventilation or to mix indoor and outdoor air. (See A.S.H.V.E. Ventilation Standards, Chapter 46.)

2. To warm the air to approximately the room temperature if the unit is intended for ventilation only, or to a higher temperature if it is intended to take care of all or a part of the heat transmission losses from the room.

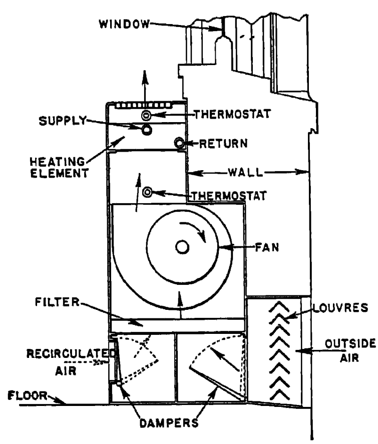


FIG. 8. TYPICAL UNIT VENTILATOR SHOWING ONE OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

3. To control the temperature of the air delivered so as to prevent both cold drafts and overheating. (See Chapter 33.)

4. To deliver air to the room in such a manner that proper distribution is obtained without drafts.

5. To recirculate room air for the purpose of heating or promoting comfort when ventilation is unnecessary. (Ordinances should be consulted.)

6. To perform all its functions without objectionable noise.

7. To clean the air properly.

In general the features of this type of unit are quite similar to those given for unit heaters.

Ratings

Unit ventilators are customarily furnished with two ratings, one established by anemometer readings and the other by condensation. The latter is for standard air. For the former, capacities vary from 750 to 10,000 cfm. Each size may be equipped with radiators for various rates

TABLE 3. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

Cubic Feet of Air Per Minute		Total Capacity in Square Feet, Equivalent Direct Radiation	Capacity Available for Heating the Room, Square Feet Equivalent Direct Radiation	Final Air Temperature (Deg Fahr)
Anemometer Rating	Condensate Rating			
750	500	214	56	95
1000	750	320	84	95
1260	1000	427	112	95
1560	1250	534	141	95

of condensation, to give different final temperatures for a given air capacity and entering temperature, thus enabling the engineer to select the unit best adapted to the heating and ventilating load. Relatively low final temperatures are conducive to the smallest temperature variation throughout a room. Table 3[6] shows the air handling capacities by the two methods of rating and also approximate heating data.

If no direct heating surface (radiation) is installed, the combined heating and ventilating requirements must be taken care of by the unit ventilators, and the total heat to be supplied is obtained by means of the following formulae:

When all of the air handled by the unit is taken from the outside,

$$H_t = 0.24\, W\, (t_y - t_o) \tag{1}$$

$$W = d\, 60\, Q \tag{2}$$

$$t_y = \frac{H}{0.24W} + t \tag{3}$$

where

d = density of air, pounds per cubic foot.
H = heat loss of room, Btu per hour.
H_v = heat required to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour $= H + H_v$.
Q = volume of air handled by the ventilating equipment, cubic feet per minute.
t = temperature to be maintained in the room.
t_o = outside temperature.
t_y = temperature of the air leaving the unit.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure.

From Equations 1, 2 and 3:

$$H_t = H + 0.24\, d\, 60\, Q\, (t - t_o). \tag{4}$$

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air is taken from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution. $H = 24{,}000$; $d = 0.075$; $Q = 1000$ cfm; $t = 70$ F; $t_o = 0$ F.

[6]A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25).

Substituting in Equation 4:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\ (70-0) = 99{,}600 \text{ Btu per hour}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F.}$$

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24 W_o\ (t_y - t_o) + 0.24\ W_i\ (t_y - t) \tag{5}$$

where

W_o = weight of air, pounds per hour taken from out-of-doors.
W_i = weight of air, pounds per hour taken from the room.

$$W_o = d_o\ 60\ Q_o \tag{6}$$

$$W_i = d_i\ 60\ Q_i \tag{7}$$

where

d_o = density of air, pounds per cubic foot at temperature t_o.
d_i = density of air, pounds per cubic foot at temperature t.
Q_o = volume of air taken in from the outside, cubic feet per minute.
Q_i = volume of air taken in from the room, cubic feet per minute.

$$t_y = \frac{H}{0.24\ (W_o + W_i)} + t \tag{8}$$

$$H_t = H + 0.24\ d_o\ 60\ Q_o\ (t - t_o) \tag{9}$$

Equations 5, 6, 7, 8, and 9 may be used in the same manner as is illustrated above for Equations 1, 2, 3, and 4. It may be noted in Equation 9, representing the total heat requirements, that as the quantity Q_o is diminished the heat requirements for the unit diminish very materially.

In Example 1, if the quantity of air taken in from the outside is reduced to zero, or all of the air handled by the unit is recirculated, the total heat requirements H_t reduce from 99,600 to 24,000 Btu per hour, or to about one fourth. Such a unit handling one third of its air volume from the outside and two thirds from the room would show a total heat requirement of $24{,}000 + \frac{99{,}600 - 24{,}000}{3} = 59{,}200$ Btu per hour. Units designed and operated on this principle show an average heat requirement and, therefore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24\ W\ (t_y - t) \tag{10}$$

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\ W\ (t_y - t_o) \tag{11}$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to exactly provide for the ventilating

requirements, the direct radiation would be selected on the usual basis. However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_y for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

Applications

Items to be considered in the application of unit ventilators include the following:

1. Combination with other means of heating.
2. Location of units.
3. Method of venting.

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at very near the room temperature, and enough separate direct heaters are placed in the room to warm it to the desired temperature, independently of the unit. Their principal advantage lies in offsetting the cooling effect of window and wall surfaces long before these can be heated to room temperature and in retaining heat for this purpose after the ventilation is shut down.

Where the unit ventilator selected has a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct heating surface installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

A combined system employs the unit ventilator alone, its capacity being sufficient both for ventilation and for supplying the heat loss. Direct heating surface is omitted altogether. It becomes necessary then that the fan be running whenever the room is to be heated but this also gives assurance of ventilation, especially if automatic dampers are used in the air intake from out-of-doors and in the recirculating intake arranged so as to give a certain quantity of air from the outside (commensurate with weather conditions) whenever the unit is operating and after the room is heated. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating, but if the electric energy fails there will be practically no heating.

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall. It is difficult to obtain proper air distribution if the unit is erected either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom

The size and location of the vent[7] outlet is important. In many cases the sizes for public buildings are regulated by law, but the location of the vents generally is left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system.

The cross-sectional area of the vent flue itself may be figured on the basis of 15 sq in. of flue for each 100 cfm. Thus the vent flue area of a flue for a room equipped with one 1200 cfm unit ventilating machine would be 180 sq in. The area of vent flue opening from the room may be figured on the basis of 25 sq in. per 100 cfm.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, heating and ventilating them with air which otherwise would be passed to the outside

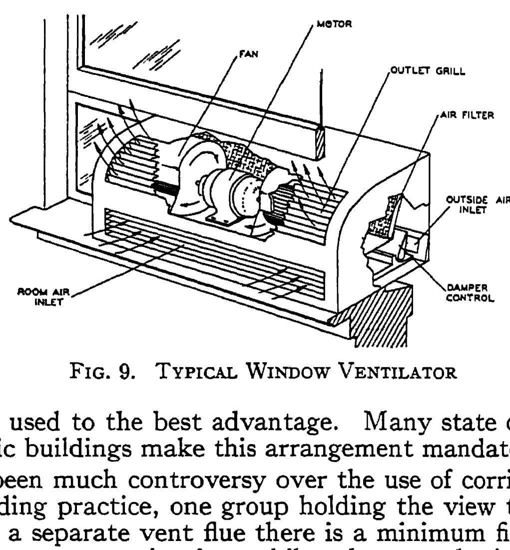


FIG. 9. TYPICAL WINDOW VENTILATOR

without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

There has been much controversy over the use of corridor ventilation in school building practice, one group holding the view that when each classroom has a separate vent flue there is a minimum fire risk and less likelihood of cross-contamination, while others emphasize the economy features of the corridor discharge and minimize the fire contamination, and other hazards.

WINDOW VENTILATORS

A window ventilator consists of filters and motor driven fans enclosed in a cabinet to be mounted on the window sill of homes or offices. These units accomplish ventilation, air cleaning, and air circulation. The direction of air discharge is manually adjustable for seasonal operation. Fig. 9 illustrates a unit of this type.

---

[7]A.S.H.V.E. RESEARCH REPORT No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson, and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 463).

A.S.H.V.E. RESEARCH REPORT No. 1017—Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet, and M. F. Lichtenfels (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 279).

UNIT HUMIDIFIERS

A unit humidifier consists essentially of some type of equipment for adding moisture to the air, usually a fan to draw the air through the humidifier, and in some cases tempering coils and filters, all encased in a single cabinet. These units are generally used in conjunction with heating systems which do not provide the necessary humidification during winter operation.

In any type of unit humidifier, the process of adding moisture to the air requires heat from a heating coil, water, or air itself.

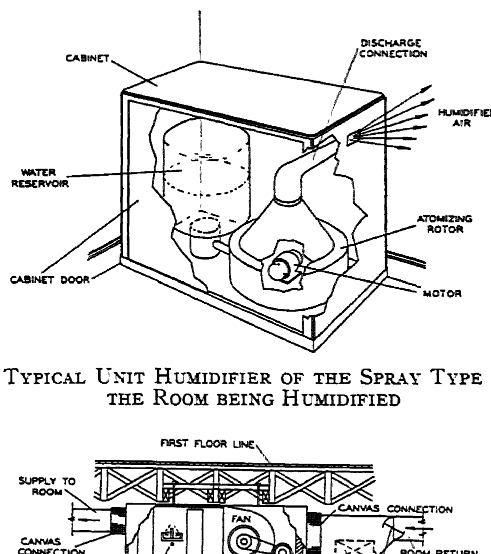


FIG. 10. TYPICAL UNIT HUMIDIFIER OF THE SPRAY TYPE FOR USE IN THE ROOM BEING HUMIDIFIED

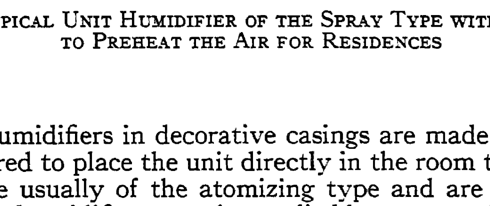


FIG. 11. TYPICAL UNIT HUMIDIFIER OF THE SPRAY TYPE WITH STEAM COIL TO PREHEAT THE AIR FOR RESIDENCES

Types of Units

Small unit humidifiers in decorative casings are made for applications where it is desired to place the unit directly in the room to be humidified. These units are usually of the atomizing type and are completely self-contained. The humidifier water is supplied by a reservoir which must be refilled at intervals. In most units the fine spray of water is mixed with some room air and the mixture is discharged directly into the room. The heat required for humidification in this method is obtained by transforming some of the sensible heat of the air to latent heat. A unit of this type is illustrated in Fig. 10.

Another type of small unit humidifier employs the principle of vaporizing the water by the direct application of heat. One method commonly used is to immerse an electric heating element in a reservoir of water

to heat it until some of the water is vaporized into the air stream. This type of unit is usually used in the same range of capacities as the spray type described previously.

A third type of unit humidifier used extensively is the larger spray type of unit to deliver enough humidifying capacity for a residence. In this type of unit either the water or air is heated. Fig. 11 illustrates a typical unit of this type. These units usually include air filters and in some cases provide ventilation air by means of an outside air duct connection to the unit. The units are available for either floor or ceiling mounting and are usually placed in a central location in the basement with short supply and return duct connections from the first floor. Room air is brought into the unit through the return duct connection and first passes over a tempering coil heated by steam or hot water, then is humidified by passing through some type of spray humidifier. Surplus moisture is removed by an eliminator and the humidified air is delivered to the room through a duct connection. Since a large percentage of the tempering coil capacity is transformed into latent heat during the humidifying process, the unit does not generally eliminate any existing steam radiation but does tend to improve comfort conditions by supplying heating during the off-period of furnace operation.

For a complete discussion of the principles of the various methods of humidification, refer to Chapter 26.

Chapter 22

UNIT AIR CONDITIONERS, COOLING UNITS, ATTIC FANS

Unit Air Conditioners, Functions, Types, Application, Cooling Units, Attic Fans

A UNIT is an assembly of the functional elements indicated by its name, such as air conditioning unit, room cooling unit, etc. A unit of this type may be complete in itself employing its own direct means of air distribution and source of refrigeration or heating, in which case it represents a complete self-contained unit. Or it may be coupled with separate means of refrigeration and air distribution, in which case it will still be considered a unit system in comparison with customary field fabricated central station systems.

The code, Standard Method of Rating and Testing Air Conditioning Equipment[1], defines the various types of unitary equipment:

1. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.
2. An *Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for maintaining temperature and humidity within prescribed limits.
3. A *Cooling Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for cooling and maintaining humidity within prescribed limits.
4. A *Self-Contained Air Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements. Self-contained air conditioning units are classified[2] according to the method of rejecting condenser heat (water cooled, air cooled, and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type).
5. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.
6. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

UNIT AIR CONDITIONERS

This equipment takes the form of an encased assembly including the apparatus necessary to perform either some or all of the functions of

---

[1]Prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association* and *Air Conditioning Manufacturers' Association*.

[2]Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association*.

cooling, dehumidifying, filtering, ventilation, air circulation, heating, and humidifying. Control of the air conditions is provided by manual switches, automatic devices or a combination of the two. The controls are usually mounted on the units.

The various elements required to produce the effects on the conditioned air are discussed herewith under separate headings.

Heating

Heating in the air conditioning unit is ordinarily accomplished by a heating coil of the non-ferrous finned tube type supplied with either steam or hot water. The steam or hot water is supplied from an external source.

In some cases the heating element may be an electric heater. Where electric power is low in cost, air conditioning units may provide heat from encased or open strip heaters (see Chapter 43). Radiant electric heaters are seldom used except as their radiant heat is absorbed by some receiving wall and then transmitted to the air in the form of convected heat.

Humidifying

Humidifying the air requires a source of heat which may be supplied by applying heat directly to the humidifier water, by applying heat to the air to be humidified, or by picking up heat directly from the air.

The oldest and best known method of humidifying air is by means of a spray. The simplest system is that in which the spray water is furnished from a constant water source, and the excess is permitted to run to waste. The spray effect may be accomplished either by a direct atomizing type which breaks the water down into fine particles by passing it through nozzles, or the water may be directed in a fine jet to a flat surface or target. In these methods some of the sensible heat of the air is transformed into latent heat. These methods are normally inefficient in the use of water. In some cases, the spray is impinged against a heated surface, thereby evaporating some of the water to increase the humidifying capacity. While this is practical in some instances, there is always the danger of scale formation where hard water is employed. A direct steam spray is used in industrial applications but is seldom used in air conditioning units for comfort work due to the resulting odors.

A popular method is to use a drip humidifier. In this type, the water flow is controlled by a solenoid valve and the water is poured into a pan. The pan contains a series of small holes through which the water passes and drips over a built-up section of galvanized screening. The air passes through the screens and picks up moisture.

Another method of humidifying is the evaporative pan type. This is used in comfort conditioning units and consists of a container offering as much water surface as possible and equipped with means of heating the water. The heat may be applied either electrically, by steam, hot water, or by circulation of the water from a heated space through the evaporating pan. Since the humidification is accomplished by surface evaporation only, if low pressure steam or hot water is used, it is essential that the air stream be directed across the surface and that the evaporating surface be large. The evaporative pan type of humidification limits the water wastage and is usually supplied with water through a float valve.

Due to the collection of salts in this evaporating pan such humidifying systems require occasional drainage and cleaning.

Other methods of humidification attempted in air conditioning units are through the use of wetted fabrics, porous earthenware plates, or other capillary surfaces. These methods rely upon the capillary absorption of the moisture from the liquid level into the portion exposed to the air. They have a tendency to lose their effectiveness due to the resulting deposit of mineral salts at the evaporating surfaces thereby clogging the pores and reducing the contact of the air with the water. Also they frequently become foul and often support bacterial growth.

Cooling and Dehumidifying

The cooling and dehumidifying effects on air are produced either simultaneously as in the case of a direct expansion cooling coil, or separately as in the case of an adsorption process and separate cooling coil.

In conditioning units, the use of surface cooling is probably the most common method of producing reduction in dry-bulb temperature of the air and dehumidification simultaneously. The type of surface employed may be cast or fabricated from tubes. In present day practices finned tubes or plate fins through which tubes are passed form the most generally used cooling surface. The detailed fabrication of this surface and the arrangement of the tubes will depend largely upon the type of refrigerant for which it is intended.

The simplest construction is that in which chilled water or brine is used as the refrigerating medium. With direct expansion refrigerant it is usually necessary to provide a special arrangement of headers so that proper distribution of refrigerant through all the surface is obtained. In some cases, ordinary brine coils can be used when operated as a flooded refrigerant system. In some units a combination of a direct spray and a refrigerant surface is used, the spray being directed against the surface. Such systems claim the advantage of air washing together with the maintenance of a clean and effective cooling coil.

When suface coolers are used, adequate protection in the form of filters or at least lint screens are necessary to prevent fouling of the surface from the air borne dirt. Surfaces not so protected frequently become completely matted with lint, grease, and similar dirt.

The sources of refrigeration used with these surface type conditioning units are discussed in Chapter 24. However, they may be divided into the following groups:

1. Direct expansion refrigerant in which the liquid refrigerant is evaporated within the coils of the unit. The vapor from these coils may be recompressed in centrifugal, rotary, or reciprocating type compressors, and the refrigerant again returned to the evaporator coil.

2. Indirect refrigeration by means of:
   - a. Cold well water.
   - b. Cold city water.
   - c. Artificial refrigerated water provided by direct expansion of refrigerant in a water cooler, direct steam jet refrigeration, or by the melting of ice.

Another direct means of cooling and dehumidification is through the use of ice. In such units the ice is brought into as intimate contact as

possible with the air handled. Provision is made for the removal of the moisture as rapidly as it is formed from the melting of the ice. Ice is also used to cool water which is circulated through the sprays.

Other methods of dehumidification accomplished by direct contact with the transfer medium are by means of the so-called *adsorption* and *absorption* systems. (See Chapter 23.) It must be recognized that these methods of dehumidification do not in themselves provide cooling. The substance removes the water vapor from the air thereby heating it. This highly dehumidified air may then be cooled either by partial rehumidification or by direct contact with a cooling medium of cold water or direct expansion refrigerant.

Filtering

The filtering or air cleaning function of an air conditioning unit is accomplished in a variety of ways depending upon the amount of filtering required. In unit systems where filtering alone is considered satisfactory,

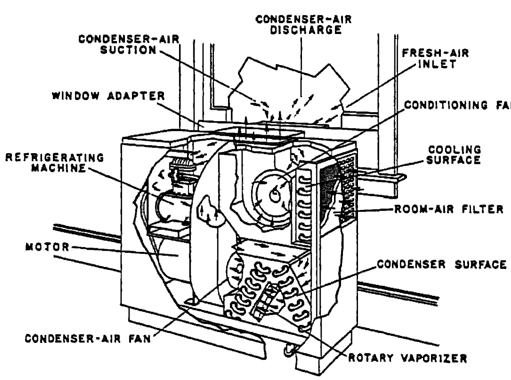


FIG. 1. SELF-CONTAINED ROOM TYPE AIR CONDITIONING UNIT FOR COOLING

the degree of filtering varies widely and in proportion to the actual needs. If the air is chiefly recirculated with but little outside air used for ventilation, filtering requirements are largely limited to keeping the coils in a clean and operable condition. Thus such units are frequently furnished with simple lint screens of low resistance and formed of moderately close meshed wire. Where outside air is used for ventilation, more complete filtering of dust particles is necessary and for this purpose, there are a large number of filters available on the market. Some of these filters are of the so-called *throw-away* type, constructed of inexpensive material so that when they become dirty or clogged they may be thrown away and replaced with new ones. All of these filtering methods are described in detail in Chapter 28.

Ventilating

The ventilating function or introduction of outdoor air is an important consideration in air conditioning units for comfort cooling. While a unit that recirculates all its air capacity is still considered an air conditioning

unit, the better type system provides for the introduction of a certain proportion of outdoor air. In some instances one of several units may operate entirely on outside air, while in other cases only a portion of the air handled by the unit is drawn from out-of-doors. In such cases a damper is provided either in the unit or in the duct connections for controlling the proportion of outdoor air.

Types of Units

Several types and designs of air conditioning units are available for selection. New designs are constantly appearing, with new improvements, greater capacities, wider range of application, and superior construction. Air conditioning units may be classified into the following types:

1. Self-contained air conditioning units.
   a. Room air conditioners for mounting either on the floor or window sill. The condensers are either air, water, or evaporatively cooled.
   b. Store air conditioners for mounting either inside the conditioned space and discharging air directly from the unit, or located outside the conditioned space with ducts connected to the unit. The condensers in this type of unit are water cooled.

2. Remote air conditioning units. These may be either the suspended type or floor type. Design of the floor type of unit varies depending upon the type of application. Units for multiple installation in office buildings or hotels, units for individual offices, and commercial refrigeration units are some of the varieties manufactured.

The self-contained room air conditioning units are finished in decorative cabinets to harmonize with the interiors of residences or offices. For the operation of this unit it is only necessary that it be located adjacent to a window or shaft to which air connections can be made and to plug in the motors to a convenient light socket. A unit of this type is illustrated in Fig. 1. In this particular unit, the conditioned air enters on the side, passing through a grille, filter, and cooling coil, and is delivered vertically to the room through a special motor and fan assembly. Refrigeration is furnished by a reciprocating compressor driven from a motor located in the base. This compressor utilizes an air cooled condenser. Air is drawn into the base by a fan mounted on the compressor motor, so arranged that the air passes through the refrigeration condenser and is again discharged out through the window connection. A novel feature of this design is that the condensate from the cooling coil is sprayed over the condenser surface and there vaporized, thus eliminating the need for drain connections. One advantage of this type of conditioning unit is that it may be removed from the occupied space during the winter season when cooling is not needed.

Other self-contained room air conditioners are available with water cooled or evaporatively cooled condensers. In the water cooled unit, a water connection and drain must be made to the unit, thus reducing the mobility of the unit. The evaporative cooled condenser model also requires a small water connection to supply the necessary water to evaporatively cool the condenser.

Self-contained room air conditioners have been made in sizes from ⅓ to 1½ hp. The ⅓ and ½ hp sizes are usually window type units with air cooled condensers.

Self-contained air conditioning units for stores and other commercial establishments have achieved prominence in the last few years. These units range in capacity from 1 to 15 hp. The equipment is enclosed in steel casings and in sizes up to 10 hp is finished to harmonize with the interior of commercial establishments. The units use water cooled condensers and are designed for floor mounting. Units in sizes up to and including 5 hp usually include air distributors to discharge the air directly into the conditioned space, thus eliminating the cost of duct work. The air distributors are adjustable to direct the air flow in such a manner to provide even air distribution in the space being conditioned. These units may use 100 per cent recirculated air or may supply quantities of ventilation air by means of a duct connected from outdoors to the inlet of the unit.

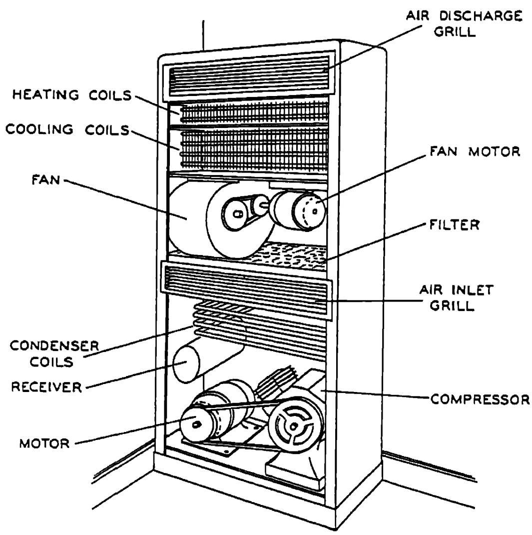


FIG. 2. SELF-CONTAINED STORE TYPE AIR CONDITIONING UNIT

Self-contained store units above 5 hp are usually located adjacent to the conditioned space and ducts connected from the unit to outlets in the conditioned space. This is necessary because it is not usually possible to evenly distribute the large volumes of air handled by these units from a single air distributor. The larger units are therefore not as decoratively finished since they are placed outside the conditioned space.

Another problem in the design of these units is to provide a means of removing the heat of compression and the heat of the motor from the unit. This may be done by a small water cooled cooling coil placed inside the condensing unit compartment. The same water is passed through the water cooled condenser. In the larger sizes, the enclosure around the condensing unit is perforated or screened so that the ambient air removes the condensing unit heat. This is possible since the units are placed outside the conditioned space and appearance of the unit is not a major factor. A third method used with units located in the

conditioned space is to pass some of the recirculated air through the condensing unit compartment then up into the air conditioner section. This method reduces the net cooling effect of the unit since some of the cooled air is used to remove heat from the condensing unit enclosure.

Store units usually provide for the inclusion of heating coils and humidifying equipment as optional equipment where winter ventilation and circulation are desired.

A typical self-contained store air conditioning unit is illustrated in Fig. 2. In this particular unit the air enters a grille located at the front of the unit, passes through a filter, and is discharged by a blower through a cooling and heating coil to an adjustable discharge distributor. The air is delivered in a manner to insure good distribution without being directed at the occupants. In some designs the air may be discharged

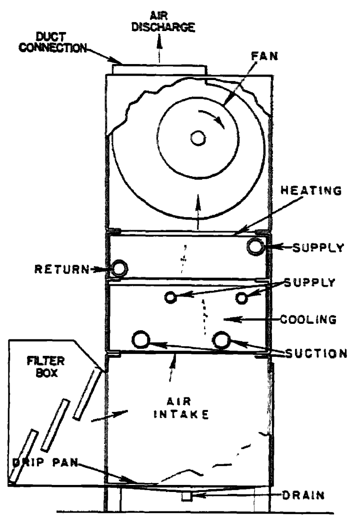


FIG. 3. VERTICAL REMOTE TYPE AIR CONDITIONING UNIT, YEAR 'ROUND

from the sides of the unit as well as from the front. The refrigerating effect is furnished by a reciprocating compressor belt connected to a motor, all mounted on a resilient base to reduce vibration. The heat dissipated by the condensing unit is generally removed by water cooling. Panels are removable for servicing and replacement of filters. The motor starter and other controls are mounted inside the enclosure and the unit may be operated either as a cooling unit or circulating unit by using the manual switches mounted on the side panel, or it may be automatically controlled by means of a thermostat.

Remotely located air conditioning units vary widely in details of construction and are different from the self-contained type of unit in that the sources of refrigeration or heating are not enclosed in the unit.

The vertical floor type remote unit shown in Fig. 3 consists of a fan section, housing one or more fans, mounted on a coil section in which are located a heating coil and a cooling coil, which may be built for

either direct expansion refrigerant, chilled water, or brine. These two sections are supported on a third or drip pan section. The distributing duct system is attached to the fan outlets and return and outside air connections are made to the drip pan. A filter box is illustrated attached to the drip pan section.

A horizontal remote type of air conditioning unit is illustrated in Fig. 4. This unit is similar in construction and operation to the vertical type explained previously. In the smaller sizes this type is installed in the

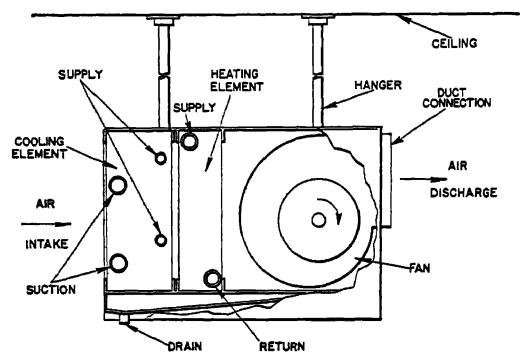


FIG. 4. HORIZONTAL REMOTE TYPE AIR CONDITIONING UNIT, YEAR 'ROUND

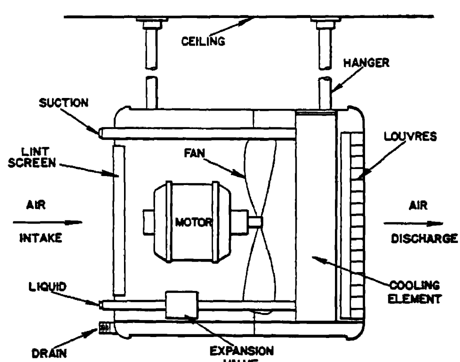


FIG. 5. SUSPENDED PROPELLER FAN TYPE COOLING AIR CONDITIONING UNIT

conditioned space discharging the air directly from an air distributor. A common type of suspended unit for exposed location utilizing a propeller type fan and suitable for summer conditioning only is illustrated in Fig. 5. Such units are equipped with either a direct expansion coil or one for chilled water or brine circulation. The outer cabinet is made of wood-grained steel or baked enamel and is insulated from the cool air chamber to prevent external condensation. The drip from the coil is collected in an insulated drip pan and carried to a drain. The inlet to the unit is provided with a lint screen to protect the cooling surface. Such units are normally used for recirculation only but may be connected for

ventilation through short full-size ducts. Similar units are available with twin housed fans of the same general construction, although usually such fans draw the air instead of blowing it through the coils.

A spray type remote conditioning unit is illustrated in Fig. 6. This spray type unit, which is similar to the arrangement given in Fig. 3, provides for the complete washing of the air and the cooling coil. For winter operation the spray provides means for humidification. The units may be obtained with by-pass dampers as shown in Fig. 6, to provide control of cooling in summer and humidification in winter. The spray type unit without the cooling coil may be used for humidification and heat control. This type of air conditioning unit is used in industrial process air conditioning as well as for comfort air conditioning.

All of these types of remote air conditioning units are usually located

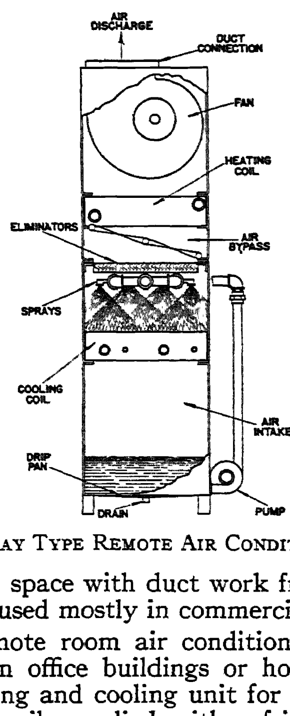


FIG. 6. SPRAY TYPE REMOTE AIR CONDITIONING UNIT

outside the conditioned space with duct work from the unit to the conditioned space and are used mostly in commercial application.

Another type of remote room air conditioner is the type used for multiple installations in office buildings or hotels. An all-year-round floor type remote heating and cooling unit for an exposed location and with direct expansion coil supplied with refrigerant from a remotely located compressor is shown in Fig. 7. A cooling coil for use with chilled water may be substituted for the direct expansion coil indicated. The fans below the separate cooling and heating elements deliver the air against deflectors thereby obtaining distribution across the face of the element and preventing condensate from dripping down into the fans. The plate upon which the fans are mounted serves as the drip pan from which the water is conducted to the drain. Separate elements are used for heating and cooling without manual control. When the unit is used for summer conditioning only, the heating coil may be omitted. The illustration indicates an evaporative type humidifier and drain pan.

Other units are available in which a target spray humidifier is substituted for the evaporative type thereby supplying humidification in winter for application in rooms with other existing heat sources. However, this spray will not provide a great deal of humidification unless the water or air passing through the unit is heated.

Still another remote type of unit is available in which the fans are mounted at the top of the unit delivering directly through a grille and drawing their air supply through the cooling and heating coils. Other variations in proportion and details of construction of this general arrangement are common. With this type of unit, ventilation is usually provided by means of a separate duct connected to the inlet of the unit.

An entirely different arrangement of the remote air conditioner for multiple installation is shown in Fig. 8. This places both the air inlet and the discharge at the top of the unit. The fan at one side discharges the air downward to the bottom where it turns and passes horizontally

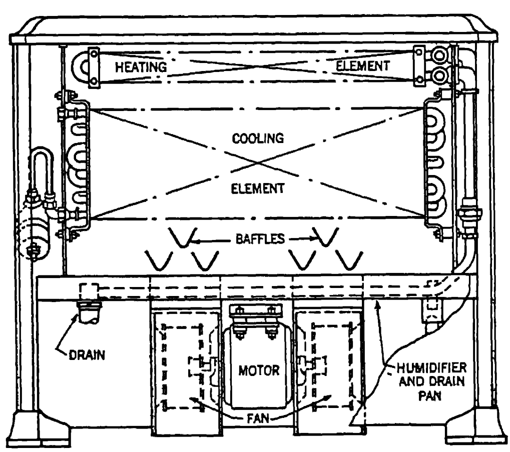


FIG. 7. FLOOR TYPE REMOTE ROOM AIR CONDITIONING UNIT FOR HEATING AND COOLING

through an atomizing spray air washer. The path then continues upward through eliminators, a cooling surface and a heating surface before it leaves the unit. With steam or hot water connected to the heating element, this unit gives controlled temperature, humidity, air cleaning, and air movement in both summer and winter. Excess water is run to waste. Acoustical treatment of the housing and outlet baffles permits installation where noise requirements are exacting.

Application

In the application of unit air conditioners it is important to consider the following points:

1. Location of the unit.
2. Air distribution.
3. Use of multiple units in lieu of a central plant system.
4. Self-contained units vs. remote air conditioners.
5. Water usage of self-contained units and methods of water conservation.

In locating air conditioning units, the characteristics of the conditioned space, the building construction, the type of system employed, the duct connections, the accessibility of the unit for servicing, as well as the sources of power, water, refrigeration, heating, and drain connections should be considered.

Locating units in the conditioned space demands serious attention to insure proper air distribution. If ventilation air is required, or if the condenser of a self-contained unit is of the air cooled type, the proximity to a source of outdoor air should be considered when locating the units. Self-contained units with water cooled condensers should be placed close to the water supply and drain, and care must be exercised that the ambient temperature is never below 32 F to prevent freezing the water in the condenser. It is, of course, important to locate the unit so that panels may be easily removed so that parts are easily accessible in case of trouble.

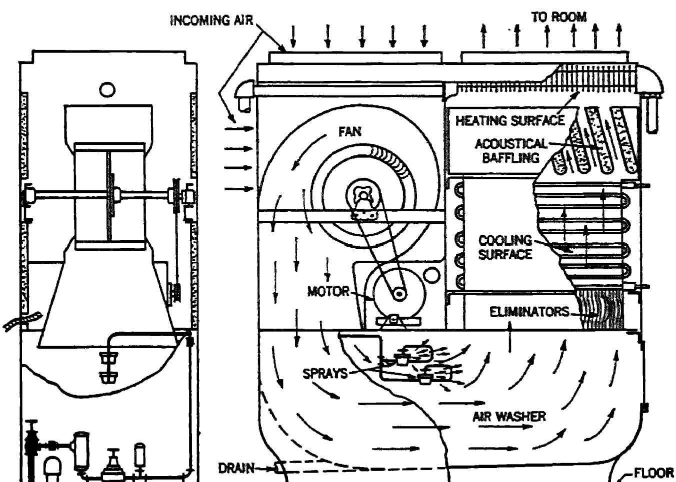


FIG. 8. REMOTE TYPE ROOM AIR CONDITIONING UNIT WITH TOP INLET AND OUTLET

Location of the remote type of unit also requires consideration of the source of refrigeration or heating. In the smaller sizes these units are placed in the conditioned space. The larger sizes are frequently located externally to the occupied and conditioned space and are connected thereto by means of delivery and return ducts. Such an arrangement permits the location of the conditioning unit convenient to either the source of refrigeration or outside air or both. It frequently permits the use of the basement or of space less valuable than that on the level or floor of the occupied zone. The design then approaches that of a *Central System*, (see Chapter 20). Oftentimes the same type of unit may find application in an exposed position for one job and in a concealed location for another. Frequently conditioning units are built into the structure or into the architectural design of a room so that they are entirely concealed except for the discharge and return grilles which are designed so as to correspond to the decorative scheme of the room.

Air distribution with self-contained room air conditioner or remote room air conditioner exposed in the conditioned space is usually through grilles or louvers built into the equipment. The grilles or louvers are adjustable to assist in directing the air properly. The discharge of the air from this type of unit is directed upward at some angle with respect to the horizontal so that the cool air is not directed at the occupants, but at the same time is carried to the most remote part of the room. In general, the air discharge should be designed to distribute cool air over the entire zone, dropping slowly and returning to the unit below the breathing line and along the floor. The location of doorways, air vents and heat-exposed walls should be carefully observed as they have a marked effect on the direction of air flow and on its uniformity of temperature.

Air distribution with the suspended remote air conditioner in the conditioned space requires attention to outlet air velocity. Sufficient air velocity should be provided to give adequate induction and mixing with room air thereby preventing the immediate dropping of the air stream and resulting objectionable cold drafts.

Air distribution with units, either self-contained or of the remote type, located outside the conditioned space is provided by ducts and outlet grilles. The location of these outlets is quite critical and is influenced both by the building construction, economies of connections, and by the distribution of load. There are a wide variety of outlet types used, and most of these have fixed delivery characteristics, thus requiring careful consideration in their location. Some types of outlets are now available with adjustable vanes thereby permitting some alteration in the delivery of the air stream after installation. This frequently eliminates objectionable down drafts resulting from the impingement of the air stream against posts, pillars, lighting fixtures, and beams. Refer to Chapter 30 for a complete discussion of this type of air distribution.

Multiple remote units are sometimes used in lieu of a central plant system in installations for large office buildings or hotels. These units, with a centrally located refrigeration or heating source, provide the advantages of individual room control by the occupants, use of less floor space since only piping lines need be run and no large ducts are required from floor to floor, and provide better fire protection since the absence of ducts prevents smoke or fire from being transmitted from one room or floor to another.

Self-contained units sometimes provide distinct advantages over remote units. Where the units are located in the conditioned space, the self-contained unit does not have any refrigerant or heating connections from an outside source and hence is more easily installed, makes a better appearance and is more easily removed if the owner wishes to relocate the equipment. When the units are located adjacent to the conditioned space, the self-contained units will use less total floor space than the remote type of system besides requiring less installation expense, due to the absence of refrigerant or steam piping.

The amount of water used in water cooled condenser types of self-contained units is sometimes an acute factor in application. Self-contained store units are almost universally water cooled. There are two types of water conserving equipment that can be used with a condensing unit; (1) a cooling tower, and (2) an evaporative condenser.

Cooling towers can easily be applied to the store conditioning unit in the same manner as they are applied to other condensing units. When the condensing unit enclosure is cooled by a water coil, the cooling tower must have enough capacity to supply both the water cooled condenser and water cooling coil. The condenser and water coil are placed in parallel to reduce the pressure drop.

Evaporative condensers can be applied to store units if the heat of the condensing unit is removed by recirculated air or a refrigerant coil, and if the water cooled condenser can be omitted and deducted from the price of the store cooler. In the case of a unit employing a water cooling coil to remove the heat from the condensing unit enclosure, the use of an evaporative condenser is uneconomical because a separate supply of water

TABLE 1. STANDARD RATING BASIS FOR SELF-CONTAINED AIR CONDITIONING UNITS

Functions	Types of Units	Rating Condition		
		Item	Description	Value
All	All	a	Barometric Pressure	29.92 in. Hg.
Cooling	Water-Cooled, Air-Cooled and Evaporatively Cooled Condensers	b	Unit Ambient and Air Entering Room—Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 80 F 67 F
		c	Ventilation Air	See Note
	Water-Cooled Condensers	d	Water Temperature Entering Unit	75 F
		e	Water Temperature Leaving Unit	95 F
	Air-Cooled and Evaporatively Cooled Condensers	f	Air Entering Outside Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 95 F 75 F
Heating	All Types Provided with Heating Function	g	Unit Ambient and Total Air Entering Unit	70 F
		h	Heating Medium, Pressure or Temperature (1) Dry Saturated Steam (2) Water In (3) Water Out	 16.7 lb per sq in. abs 180 F 160 F
Humidifying	All Types Provided with Humidifying Function	i	Unit Ambient	70 F
		j	Total Air Entering Unit (1) Dry-Bulb (2) Wet-Bulb	 70 F 53 F
Air Circulation	All	k	Filters	New and Clean

Note: Rating shall be based on both ventilation and recirculated room air entering at 80 F dry-bulb and 67 F wet-bulb temperature. (The *Note* as given in the code has been condensed in order to remove material not pertinent to this chapter.)

must be supplied to the condensing unit enclosure cooling coil, thus defeating the purpose of the evaporative condenser to conserve water.

Ratings

There are two codes governing the rating and testing of air conditioning units. The first code, Standard Method of Rating and Testing Air Conditioning Equipment[3], covers all types of air conditioning units except the self-contained type. The latter is covered by the Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling[4]. The two codes are necessary because of the basic difference caused by the heat given up by the self-contained units.

Self-contained air conditioning unit ratings are expressed in the code in terms of the effect produced on air such as:

1. The net total room cooling effect in Btu per hour. This is the actual heat removed from the room and is equal to the gross cooling effect less the heat given back to the room by the unit.
2. The net room dehumidifying effect in Btu per hour.
3. The net room sensible cooling effect in Btu per hour.
4. The sensible heating effect in Btu per hour.
5. The humidifying effect in pounds per hour.
6. The total air capacity in cubic feet per minute of standard air.

The standard rating basis as given in the code for self-contained units is tabulated in Table 1.

The standard rating basis for air conditioning units is similar to that given previously for self-contained units except the relative humidity of the entering air is specified as 50 per cent (66.7 F wet-bulb) instead of specifying the wet-bulb temperature as 67 F and the suction saturated refrigerant temperature is specified as 40 F for comfort cooling since an air conditioning unit does not include a condensing unit. The suction saturated refrigerant temperature of a self-contained air conditioning unit is not given in Table 1 as a basis of rating since this temperature is the temperature obtained when the self-contained unit operates as a system at the conditions given in Table 1. It is expected that the entering air conditions for the standard rating will agree when the code, Standard Method of Rating and Testing Air Conditioning Equipment, is revised.

COOLING UNITS

Cooling units may be used either in comfort cooling or commercial applications. As applied to industrial product conditioning and processing they are similar in construction to unit heaters described in Chapter 21 except that the heat transfer surface is supplied with refrigeration instead of heat. They are normally installed within the space to be served, or at least closely adjacent thereto.

Product cooling was originally accomplished by means of stationary pipe coils. This was later supplemented with the forced fan bunker systems in which air was passed over banks of coils. Today the coils

[3]Loc. Cit. Note 1.

[4]Loc. Cit. Note 2.

and fan are encased in an enclosure and controls are provided to maintain an average coil surface temperature. Thus, for any installation, the depth of coil, air flow, and face area determine the relation between dry-bulb temperature reduction and wet-bulb temperature reduction. Occasionally they are provided to receive outside air in which case this air is invariably filtered or washed to prevent any possible contamination of the product.

Features of Units

The principal field for cooling units is in cold storage plants, fur storage, fruit packing houses, provision stores, brewery fermentation and stock rooms, candy plants, and other industrial process work. In replacing bunker and wall coils in meat storage plants, cooling units give distinct advantages in compactness, lower first cost and maintenance expense, ease of defrosting, freedom from drip and the maintenance of sanitary conditions, as well as uniform temperature and humidity under variable load conditions. Cooling units by means of their positive air

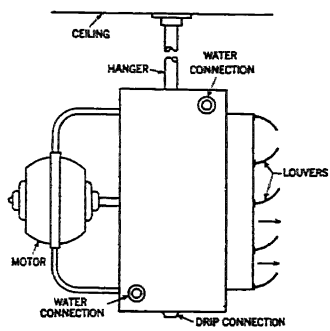


Fig. 9. Ceiling Type Cooling Unit

circulation prevent dead-air spots, frequently objectionable in this industry.

Types of Units

Cooling units are provided in two major types similar to unit heaters, either floor mounted with housed fan, or suspended with propeller type fans. Normally, air outlet velocities are lower than for heating, due largely to the effect of high velocities on the product. Cooling units are normally of the free delivery type although they occasionally are supplemented with duct work to provide more careful air distribution.

Typical cooling units are shown in Figs. 9, 10, and 11. Fig. 9 indicates a suspended type cooling unit which may be designed with or without a moisture eliminator. If high air velocities are maintained, an eliminator will be necessary to prevent the drops of moisture from being carried through with the air. The condensation that occurs is collected in a drip pan and removed from the system through a drain pipe. Fig. 10 indicates a typical floor-mounted unit of the housed fan type. The illustration shows a common form of distributing outlet designed to give low outlet velocities together with a controlled distribution. In process work, it is

often important that direct air distribution does not impinge on the product. Cooling units are normally constructed of galvanized steel or non-ferrous material in order to reduce the corrosive effect of their constant wetted condition.

Ratings

Since cooling units are mostly used in low temperature applications, the standard basis of rating is different than for air conditioning units. The Standard for Rating and Testing Air Conditioning Equipment states that the standard rating shall be based on air entering the cooling unit at 45 F dry-bulb and 85 per cent relative humidity, and the suction saturated refrigeration temperature shall be 30 F for commercial cooling.

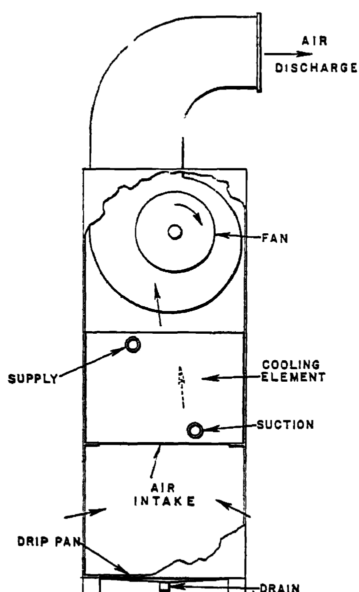


FIG. 10. SURFACE TYPE COOLING UNIT

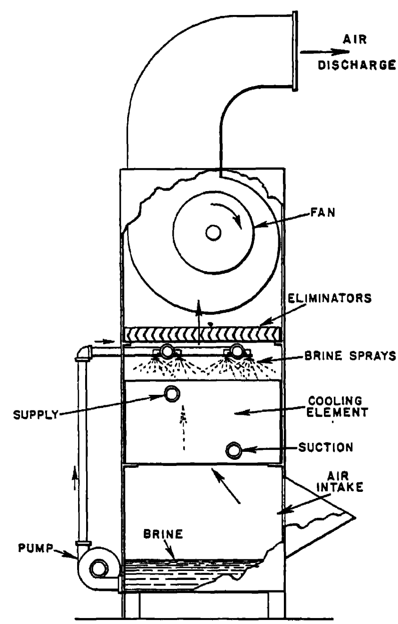


FIG. 11. BRINE SPRAY TYPE COOLING UNIT

Ratings of cooling units may be expressed in Btu per hour, or in tons of refrigeration. When ratings at other than standard conditions are given, the quantity, temperature, and relative humidity of the entering air should be specified, together with the refrigerant temperature within the coil. When chilled water or brine is used, the rate of circulation of the cooling media as well as its entering temperature should be given.

Defrosting

Cooling units are often called upon to operate in rooms where a temperature below freezing is maintained and low refrigerant temperatures are required. This results in the collection of frost on the heat transfer surface which in turn leads to a rapid loss in capacity and requires eventual defrosting. Such defrosting is accomplished by the following methods:

1. When the room is above freezing the source of refrigeration is cut off and the fan allowed to operate until the unit has defrosted.

2. A reversal of the refrigeration system may be provided and the so-called hot gas defrosting method used. This is accomplished by reversing the flow of the hot gas so that it is delivered directly from the compressor to the evaporator cooling unit. As soon as the ice and frost have been melted, the system is again returned to its normal cycle.

3. Where brine is used as a refrigerant, heated brine may be sent through the cooler to remove the ice.

4. When the room is at very low temperatures, warm air defrosting is sometimes used by providing for the admission and removal of warm air from outside the cooled space.

5. The surface may be sprayed with a strong brine solution.

In order to prevent the collection of frost in low temperature rooms where high latent heat loads are present, unit coolers equipped with a constant brine spray are frequently used. These are normally of the housed fan type similar to Fig. 10, but equipped with a pump for recirculating brine at intervals to maintain a non-freezing mixture as shown in Fig. 11.

COSTS

The following factors influence the cost of unit air conditioning installations:

1. Since the cost of the total job involves material cost plus installation labor and since through the use of unitary equipment, material costs can be kept to a minimum, every effort should be made to simplify installation.

2. Self-contained units in the small sizes now available, probably represent the lowest cost individual installations. They have, however, their limitations.

3. The floor type all-year-round air conditioning units for the occupied space with a remotely controlled compressor, heating sources being either the existing heat system or steam connections to the unit, probably afford the lowest cost all-year-round service for most individual rooms.

4. For multiple rooms or offices, the remotely located air conditioning unit with remote source of refrigeration probably represents the most economical installation. The larger self-contained air conditioners are particularly adaptable to stores, residences and small commercial installations.

Costs of operation vary widely depending upon the cost of power and water. Water costs in the larger installations are being materially reduced through the use of cooling towers and special types of condensers. It is difficult to make any comparison of operating costs of cooling as contrasted with heating equipment because the relative operating expense depends upon many factors including climatic conditions; *i.e.*, in the South the cost of operating cooling equipment greatly exceeds the operating cost of heating equipment, whereas in colder climates where cooling equipment is used about two months per year, the heating costs are probably higher than those for cooling.

ATTIC FANS

Attic fans, used during the warm months of the year to draw large volumes of outside air through a house, offer a means of using the comparative coolness of outside evening and night air to bring down the inside temperature.

Because the low static pressures involved are usually less than 1/8 in. of water, disc or propeller fans are generally used instead of the blower or housed types. The fans should have quiet operating characteristics, and they should be capable of giving about 20 to 30 air changes per hour in northern areas. In the South the usual specification requires one air change per minute which provides appreciable air movement in addition to the cooling effect.

Types

Open attic fans are units in which the fan is installed in a gable or dormer and one or more grilles are provided in the ceilings of the rooms below. Outdoor air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged out-of-doors by the fan. An attic stairway may be used in place of the grilles. It is essential that the roof and the attic walls be free from air leaks.

Boxed-in fans are units in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Outdoor air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings under the eaves.

Location

The locations of the fan, the outlet openings, and the grilles should be selected after consideration of the room and attic arrangement in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

Costs

The capacity of attic fans is from 3,000 to 30,000 cfm with the trend toward units in the range of capacity from 7,000 to 15,000 cfm.

Some typical data on an attic fan installation in an average six room house of frame construction containing 14,000 cu ft and located in the southern part of this country are:

Installed cost	\$100 to \$400, average \$200
Fan data	12,000 cfm average, 500 watts input
Operating period	April 15 to October 15, intermittently as weather conditions demand
Power consumption	500 kwhr per year for 8 months' operation

In northern climates the figures would be considerably reduced. A smaller capacity fan can be effectively used and the cost of an installation ranges from \$60.00 upwards.

Chapter 23

COOLING, DEHUMIDIFICATION AND DEHYDRATION

Definitions and Methods, Adsorbents, Absorbents, Nature of Processes, Temperature—Pressure—Concentration Relations, Dehydration Methods, Auxiliaries, Controls, Performance, Economics

THE addition or abstraction of heat to or from air, whether sensible or latent, requires (*a*) a medium held at the necessary temperature or vapor pressure to produce a flow of heat or moisture and (*b*) sufficient contact between the air and the medium to achieve the desired final condition. The medium may be solid or liquid. It may be used (*a*) directly, as in a water or brine spray, or (*b*) indirectly, as with a steam radiator or direct expansion cooling coil.

The contact is obtained through the use of exposed surface, to which the molecules of air are brought into direct physical proximity, thereby producing the heat interchange. These molecules then re-mix with uncontacted molecules in the air stream. The completeness of the interchange is a function of the number of such successive contacts, and is a measure of the efficiency of the surface[1]. The contacting surface may be that of the medium directly, such as a finely atomized spray or the bed of a solid dehydrating agent; or a chilled or warmed metal surface, as a coil; or a combination of medium and surface, such as a packed tower, where the medium produces the interchange and the surface provides the necessary contact area.

DEFINITIONS AND METHODS

There are several basic methods of producing the necessary difference in temperature or vapor pressure between air and the medium employed to achieve cooling or dehumidification, or both simultaneously:

Cooling of air involves its reduction in temperature due to the abstraction of sensible heat. It is always a result of contact with a medium held at a temperature lower than that of the air. Cooling may be accompanied by moisture addition (evaporation), by moisture extraction (dehumidification), or by no change of moisture content whatever. Such moisture change, if present, is considered as a secondary or by-product effect. As

[1]The Contact Mixture Analogy Applied to Heat Transfer with Mixtures of Air and Water Vapor, by W. H. Carrier (*A.S.M.E. Transactions*, Vol. 59, 1937).

previously stated, the medium may be directly in contact with the air (as water, brine, or ice), or indirectly through a barrier wall (as cooling surface). When the latter method is used, and the surface temperature is held above the air dew-point, only cooling occurs without moisture interchange.

Evaporative Cooling involves the adiabatic exchange of heat between air and a water spray or wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger. No heat is added or abstracted from the medium (water), which is continually recirculated. Cooling of the air occurs due to the temperature difference between entering air, and water at the wet-bulb temperature. Humidification occurs as a result of the vapor pressure exerted by the water which is higher than that corresponding to the entering air dew-point. Since this is an adiabatic exchange, the enthalpy of the air remains constant, while the dew-point rises and the dry-bulb falls, and the loss of sensible heat exactly equals the gain in latent heat (neglecting radiation losses). The maximum available temperature reduction is the total difference between entering dry- and wet-bulbs (wet-bulb depression). Equipment achieving the complete reduction is termed *completely* saturating or 100 per cent efficient, since the air leaves in a saturated state. Equipment utilizing only a portion of the wet-bulb depression is termed *partially* saturating.

Evaporative cooling is being used advantageously in many parts of the country. It is particularly applicable (1) in districts where the relative humidity is normally low during the cooling season, and (2) in applications where the cooling load is principally a sensible load.

Dehumidification of air, in its broadest connotation, means simply the removal of moisture. Usage in the art has restricted the application of the term, so that the former broad meaning is now properly covered by the complementary names dehumidification and dehydration. *Dehumidification* usually refers to the *condensation* of water vapor from air due to its contact with a chilled medium (see Cooling). This type of heat exchange invariably includes temperature reduction due to removal of sensible heat, which reduction may be considered a by-product effect.

Dehydration refers specifically to the removal of water vapor from air due to its contact with a dehydrating agent. The primary distinction between dehumidification and dehydration is the vapor pressure exerted at the surface of the contacting medium. In the case of dehumidification, this surface vapor pressure is always the same as that which would be exerted by a body of water (or ice) at that same surface temperature. In the case of a dehydrating agent, the surface vapor pressure is always *lower* than that exerted by water at the same temperature, and the effectiveness of the medium as a dessicant is largely a function of the amount by which this vapor pressure can be lowered at the working temperature involved.

Thus it is evident that the primary function of a dehydrating agent is to establish a vapor pressure difference between the air and the medium in order to secure thereby a removal of moisture (latent heat) from the air. In the simplest type of process, no heat is abstracted from the medium itself, and the process is essentially an adiabatic one in which the latent heat lost by the air is converted to sensible heat which raises the air temperature by an equivalent amount. This process is therefore an energy exchange, similar to, but the reverse of, adiabatic saturation.

Combination Methods. It is evident that two or more of the above processes—cooling, evaporative cooling, dehumidification and dehydration—may be combined by the proper application of interchangers in sequence. Such combinations are dictated by the availability of prime sources of energy and the economic justification of each.

This chapter discusses in detail the engineering and economic principles involved in the application of dehydration. For similar discussion of the other processes, refer to the following material: Cooling and dehumidification by the use of surface interchangers (cooling coils), see Chapter 25. Cooling, dehumidification and evaporative cooling with air washers, see Chapter 26. For sources of cooling involving city and well water and cooling towers, see Chapter 26, while for mechanical refrigeration and ice, refer to Chapter 24. For the thermodynamics of evaporative cooling, see Chapter 1.

DEHYDRATING AGENTS

Dehydrating agents may be divided into two general classifications:

1. *Adsorbent*—A material which has the ability to condense water vapor on its surface without itself being changed physically or chemically. Certain solid materials, such as silica gel, activated alumina and activated carbon have this property.

2. *Absorbent*—A material which has the ability to take up water vapor but which changes physically, chemically, or both, during the cycle. Calcium chloride is an example of a solid material while liquid materials include lithium chloride, calcium chloride, lithium bromide and ethylene glycol.

Adsorbents

These substances are characterized by a physical structure containing a great number of extremely small pores but still retaining sufficient mechanical strength to resist the wear and handling to which they are subjected. To be suitable for dehydration purposes such substances must fulfill the following requirements:

1. Possess suitable vapor pressure characteristics.
2. Be available at an economical cost.
3. Adsorb sufficient moisture per pound of material to avoid excessive bed dimensions.
4. Be chemically stable, resisting contamination from impurities.
5. Physically rugged to resist breakdown from handling, abrasion, etc.
6. Withstand breakdown from indefinitely repeated reactivation cycles.
7. Possess practical and efficient reactivation temperatures.

Aluminum Oxide (Alumina), in a porous, amorphous form is a solid adsorbent frequently called by the common name activated alumina. It contains small amounts of hydrated aluminum oxide, very small amounts of soda, and various metallic oxides. A good grade of activated alumina will show 92 per cent of Al_2O_3, and its soda content will be combined with silica and alumina into an insoluble compound. This substance also has the property of adsorbing certain gases and certain vapors other than water vapor—a property which is sometimes useful in air conditioning installations. It is available commercially in granules ranging from a fine powder to pieces approximately 1.5 in. in diameter. It has high adsorptive capacity per unit of weight and is non-toxic. It may be repeatedly re-activated after becoming saturated with adsorbed moisture without practical loss of its adsorptive ability. In the grade frequently used for air drying the re-activation may be accomplished at temperatures under 350 F. Specific gravity is 3.25 and the pores are reported to occupy 58 per cent of the volume of each particle. For most estimating purposes the volume-weight relation on a dry basis may be taken as 50 lb per cubic foot although in the smaller sizes the packed weight may be as much as 64 lb per cubic foot.

Silicon Dioxide (Silica), in a special form obtained by suitably mixing sulphuric acid with sodium silicate, is another solid adsorbent and is commonly called silica gel. Its capillary structure is exceedingly small, so small that its exact structure has to be deduced rather than observed. The gel is available commercially in a wide variety of sizes of granules ranging from 4 to 300 mesh. It has high adsorptive capacity per unit of weight, it is non-toxic, and may be repeatedly re-activated without

practical deterioration. Re-activation may be accomplished at temperatures of air up to 600 F although it is frequently accomplished with air or other gases at temperatures not over 300 F. Volume of the capillary pores is reported to be from 50 to 70 per cent of the total solid volume. For most estimating purposes the volume-weight relation can be assumed as from 38 to 40 lb per cubic foot on a dry basis. It also has the property of absorbing certain gas and vapors other than water vapor.

Other substances having properties which make them available as solid adsorbents include lamisilate and charcoal but details of their physical properties are not available.

Nature of Adsorption Process

The adsorbent does not go into solution but water vapor is extracted from the air-vapor stream passing through the bed of adsorbent material and is caught and retained in the capillary pores. The exact nature of the process which goes on during adsorption is not known but it is stated that the action is brought about by surface condensation, and also by a difference between the vapor pressure of the water condensing inside the pores and the partial pressure of the water vapor in the air-vapor mixture. The adsorbing process in the bed can continue until the vapor pressures come into equilibrium. The amount of vapor adsorbed will depend on the adsorbent substances being used, but for any single substance the amount depends on the temperature of the bed as well as on the partial pressure of the air-vapor mixture being passed over it.

As the bed of material adsorbs moisture, its vapor pressure approaches that of the contacting air and the rate of adsorption gradually slows down so that equilibrium may not be reached for 24 to 48 hours. Because of this diminishing rate of adsorption, commercially designed systems do not permit the state of equilibrium to be reached but generally operate on a 10 to 30 min contact time—the period of most rapid adsorption.

As the process of adsorption goes on heat is liberated in the bed. The heat so liberated is the latent heat of the water vapor condensed together with the so-called heat of wetting. For a pound of water vapor at 60 F the latent heat released by condensation is approximately 1057 Btu. The heat of wetting for silica gel, for example, is about 200 Btu, making a total heat of adsorption of approximately 1257 Btu per pound of water adsorbed from the air-vapor mixture passing through the silica gel bed. The heat of wetting varies with the substance being used as the adsorbent while the latent heat of condensation depends only on the temperature and pressure of the water vapor.

Temperature—Pressure—Concentration Relations

Since the adsorptive ability of an adsorbent depends on the temperature of the bed and on the partial pressure difference between the pores and the air-vapor mixture, it is important to know the pressures and temperatures at which pressure equilibrium is reached.

Evidently the equilibrium conditions represent the limits beyond which adsorption of vapor cannot continue. The relationship can be shown graphically and Fig. 1 is such a chart for silica gel. Charts of like nature can be plotted for other adsorbent materials.

The equilibrium conditions for a gel bed maintained at constant temperature while the water vapor adsorption is allowed to continue until pressure equilibrium is reached is shown in Fig. 1. Each curve on the chart shows a certain dew-point temperature, and therefore a certain pressure of the saturated water vapor.

As an example in the interpretation of the chart consider the case when moist air at a temperature of 80 F and a partial vapor pressure of 0.5 in. of mercury flows through a bed of silica gel which is at a temperature of 80 F. The chart indicates that the equilibrium of pressure between the

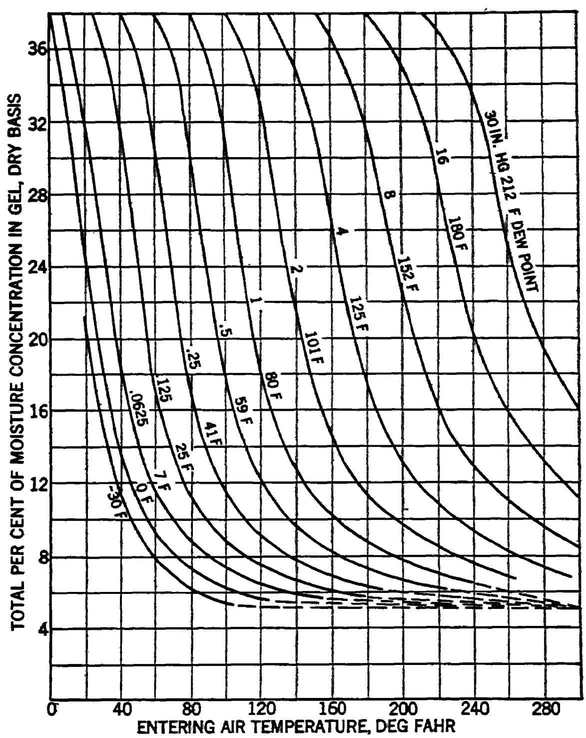


FIG. 1. TEMPERATURE—VAPOR PRESSURE—CONCENTRATION RELATION FOR A SILICA GEL BED AT CONSTANT TEMPERATURE

air-vapor mixture and the bed is reached when the dry bed has adsorbed moisture to the extent of 30 per cent of the weight when dry. When this happens the bed can adsorb no more moisture unless its temperature is changed.

While charts of this kind can show the limiting properties of the substances they are seldom directly applicable to the solution of air conditioning problems unless considerable additional information is available. This takes the form of performance data covering the characteristics of the equipment in which the adsorbent bed is placed. Such performance data are presented later in this discussion.

Absorbents

Any absorbent substance may be used as an air drying agent if it has a vapor pressure lower than the vapor pressure in the air-vapor mixture from which the moisture is to be removed.

Solid Absorbents. The substances used are in general the solid forms of the liquid absorbents, more commonly calcium chloride due to its low cost. At present they are used principally in small dessicating chambers, and in small dryers of the cartridge type, through which air is forced under pressure.

Liquid Absorbents. These are characteristically water solutions of materials in which the vapor pressure is reduced to a suitable level by governing the concentration of the solution. In addition to having suitable vapor pressure characteristics a practical absorbent must also be widely available at economical cost, be non-corrosive, odorless, non-toxic, non-inflammable, chemically inert against any impurities in the air stream, stable over the range of use and especially it must not precipitate out at the lowest temperature to which the apparatus is exposed. It must have low viscosity and be capable of being economically regenerated or concentrated after having been diluted by absorbing moisture.

Water solutions, or brines, of the chlorides or bromides of various inorganic elements such as lithium chloride and calcium chloride are the absorbents most frequently used in connection with air conditioning applications and detailed attention is confined to these two in this chapter.

Nature of Absorption Process

The application consists of bringing the air-vapor stream into intimate contact with the absorbent, permissibly by passing the air stream through a finely divided spray of the brine but more generally by passing the air over a contacting pack where the liquid absorbent presents a large surface to the air stream. The difference in vapor pressure causes some of the vapor in the air-vapor mixture to migrate into the brine. Here it condenses into liquid water and decreases the concentration of the absorbent.

As the water vapor is added to the absorbent and condenses, it gives up its latent heat of condensation which tends to raise the temperature of both the absorbent and the moist air stream. For every pound of water absorbed and condensed the heat added to the air stream and the brine combined is obtainable from steam tables. For instance, at 60 F the amount of this heat is about 1057 Btu. In addition to this heat there is involved also the so-called heat of mixing which is frequently considerable.

A more complicated cycle involves heat removal from the contacting medium, either within or external to the interchanger. Thus the temperature of the medium may be higher than, equal to, or lower than that of the air, depending on the agent used and the function to be performed. In such a cycle, the dehydration process may be accompanied by cooling or heating, or neither, and such effect, if present, may be either a necessary by-product of the process, or for the specific purpose of obtaining both latent and sensible heat removal simultaneously. The heat thus produced in the bed is to a large extent transferred to the air being dried, and in the average air conditioning installation must be removed by passing the air through an aftercooler.

Temperature—Pressure—Concentration Relations

Since the absorption process can continue only as long as there is a difference in vapor pressure between the absorbent and the air-vapor mixture and since at a given temperature of the absorbent the vapor pressure depends on the concentration of the solution, evidently there must be a relation between these quantities which if known would state the limits of the process. The relationship would also depend on the absorbent being used, and would have to be determined for each substance used as an absorbent. This relationship is shown graphically in

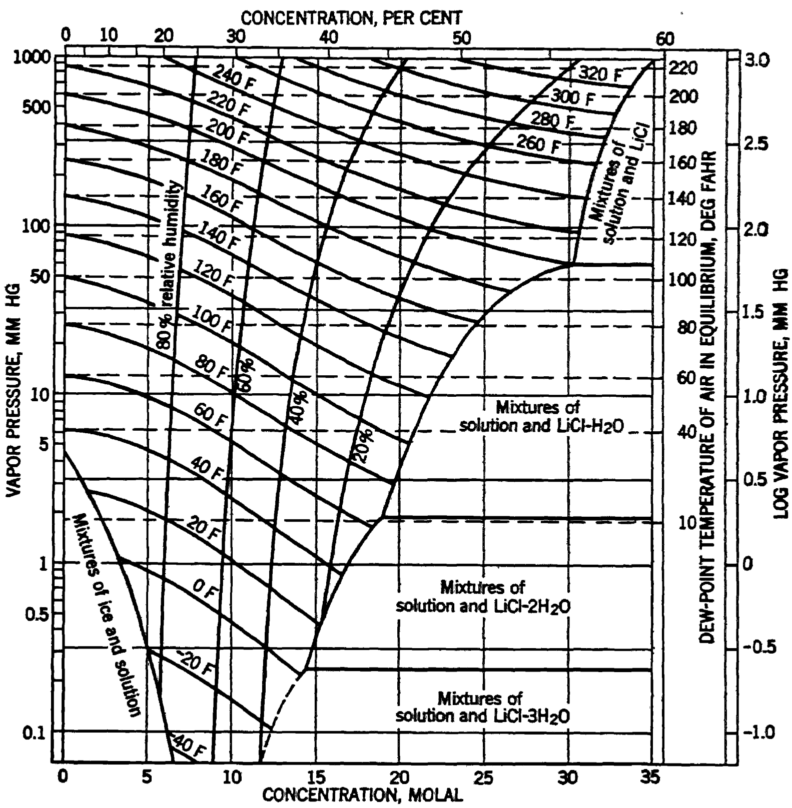


Fig. 2. Temperature—Pressure—Concentrations for Lithium Chloride

Fig. 2 for lithium chloride, and Fig. 3 presents similar data for calcium chloride. These charts are essentially similar to Fig. 1, and their direct usefulness is limited by much the same considerations. Other physical properties of lithium chloride are shown in Tables 1, 2 and 3.

In Fig. 2 and Table 1 the unit of concentration is the *mol.* A M molal solution is definied as a solution containing $M \times 42.37$ grains of anhydrous lithium chloride per 1000 grains of water. The formula connecting concentration in mols with weight in per cent is equivalent to: $(100 \times M \times 42.37) \div [1000 + (M \times 42.37)]$.

TABLE 1. PROPERTIES OF LITHIUM CHLORIDE SOLUTIONS

Concentration Pound Mols (42.4 Lb) *LiCl* per 1000 Lb Water	Concentration Per Cent by Weight	Specific Gravity at 100 F	Viscosity (Millipoise)		Partial Heat of Mixing at 0 F, Btu per Lb	Temp. Coeff. of Partial Heat of Mixing Btu per Lb per F	Specific Heat at 70 F	Boiling Point F at (760 mm Hg)	Freezing Point, F
			80 F	180 F					
0	0.0	1.000	8.61	3.48	0.00	0.000	0.998	212.0	32.0
2	7.8	1.037	11.19	4.56	2.04	−0.014	0.901	215.8	16.3
4	14.5	1.076	14.42	6.01	7.24	−0.036	0.831	221.5	− 5.8
6	20.2	1.111	18.62	7.78	16.70	−0.069	0.778	228.9	−34.2
8	25.3	1.143	24.32	10.00	31.90	−0.109	0.739	238.1	−69.0
10	29.7	1.172	32.28	12.91	51.10	−0.143	0.710	248.4	−90.0
12	33.7	1.199	43.45	16.56	75.70	−0.160	0.687	258.8	−40.0
14	37.4	1.225	60.26	21.28	90.80	−0.167	0.666	268.9	1.0
16	40.4	1.248	82.04	27.10	124.80	−0.176	0.647	277.9	36.5
18	43.3	1.270	113.80	35.48	145.00	−0.186	0.631	285.8	58.1
20	46.0	1.291		46.45	162.00	−0.194		293.2	86.4
22	48.4			60.67	171.00	−0.200		300.2	133.0
24	50.3			84.33	177.00	−0.200		307.0	156.0
26	52.4				182.00	−0.210		313.0	180.0
28	54.3				191.00	−0.210		318.0	190.0
30	56.1				194.00	−0.210		323.0	195.0
32	57.5				198.00	−0.220		328.0	280.0

TABLE 2. DEW-POINT OF AIR IN EQUILIBRIUM WITH LITHIUM CHLORIDE SOLUTIONS
CONCENTRATION IN POUND MOLS (42.4 LB) LITHIUM CHLORIDE PER 1000 LB WATER

Dew-Point at Zero Conc.	Concentration of Lithium Chloride														
	2.0	4.0	6.0	8.0	10.0	12.0	14.0	16.0	18.0	20.0	22.0	24.0	26.0	28.0	30.0
320	315.2	308.7	299.9	290.2	279.7	269.4	259.6	251.5	244.1	236.5	230.0	223.8	218.6	214.5	210.3
300	295.4	289.1	280.5	270.9	260.6	250.5	240.8	232.6	225.4	218.0	211.8	205.8	200.8	196.9	192.8
280	275.6	269.5	261.1	251.7	241.5	231.6	222.2	214.0	206.7	199.7	193.5	187.8	183.2	179.3	175.2
260	255.8	250.0	241.9	232.6	222.7	212.8	203.5	195.5	188.4	181.7	175.4	170.0	165.6	162.0	158.4
240	236.0	230.4	222.5	213.5	203.8	194.2	185.0	177.1	170.0	163.6	157.5	152.2	148.3	144.6	140.5
220	216.2	210.8	203.2	194.4	184.9	175.5	166.4	158.6	151.6	145.3	139.6	134.6	130.7	127.3	124.2
200	196.4	191.2	183.9	175.4	166.1	156.7	148.0	140.3	133.5	127.3	121.9	117.0	113.3	110.1	
180	176.6	171.6	164.7	156.4	147.3	138.1	129.6	122.1	115.5	109.4	104.2	99.6	96.0		
160	156.8	152.1	145.4	137.4	128.6	119.7	111.3	103.9	97.4	91.6	86.6	82.2			
140	137.0	132.6	126.1	118.4	109.9	101.3	93.1	85.9	79.5	73.8	69.0				
120	117.2	113.0	106.8	99.4	91.1	82.7	74.7	67.8	61.5	56.0					
110	107.3	103.2	97.2	89.9	81.9	73.5	65.6	58.8	52.6	47.1					
100	97.4	93.4	87.5	80.5	72.7	64.4	56.6	49.8	43.7	38.2					
90	87.5	83.6	77.9	71.0	63.3	55.2	47.6	40.8	34.8	29.3					
80	77.6	73.8	68.4	61.6	54.0	46.1	38.5	31.8	25.9	20.6					
70	67.7	64.0	58.7	52.2	44.8	37.0	29.5	22.9	17.2	12.0					
60	57.8	54.3	49.1	42.7	35.5	27.9	20.5	14.0	8.3						
40	38.0	34.7	29.9	23.9	16.9	9.6	2.4	−3.9							
20		15.1	10.7	5.0	−1.7	−8.7	−15.4								
0		−4.5	−8.6	−13.9	−20.2	−27.0	−33.3								

Example 1. Determine the dew-point, wet-bulb, relative humidity and absolute humidity of air in equilibrium at 100 F with pure lithium chloride solution of density 1.270.

Solution. From Table 1 the concentration of a solution of density 1.270 at 100 F is 18.0 *M*. From Fig. 2 the dew-point of 18 *M* lithium chloride at 100 F is 43.7 F. From Table 6, Chapter 1, the partial pressure of water over the solution is 0.2858 in. of Hg, and the absolute humidity is 42.00 grains per pound dry air. From the psychrometric chart, the wet-bulb is 66.3 F, and the relative humidity is 15 per cent.

Example 2. Determine the boiling point, and freezing point of 18 *M* lithium chloride solutions.

Solution. From Table 1, boiling point (standard) is 285.8 F, freezing point is 58.1 F.

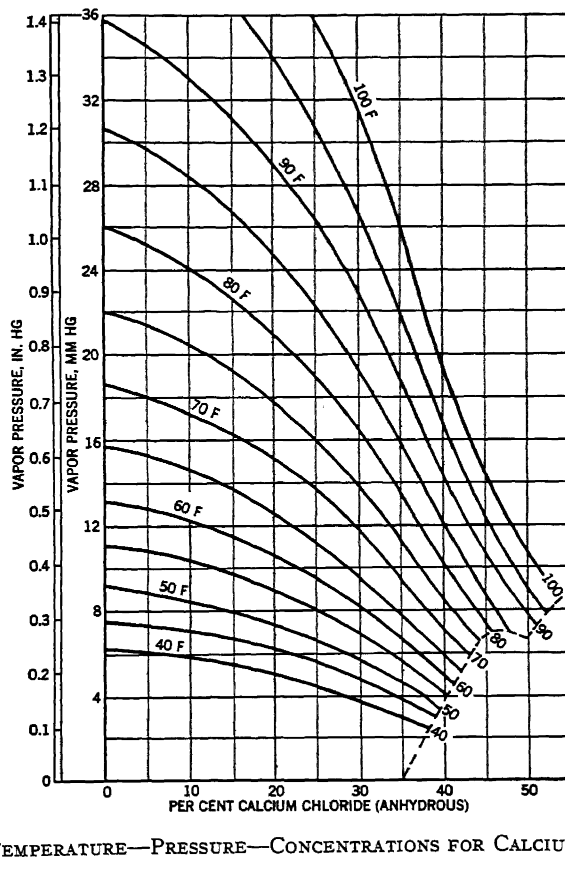


Fig. 3. Temperature—Pressure—Concentrations for Calcium Chloride

Example 3. Calculate the heat of vaporization of 1 lb of water from a large amount of 18 *M* lithium chloride solution at the boiling point.

Solution. The heat of boiling is equal to the heat of mixing plus the heat of boiling pure water at the same temperature. The heat of mixing from Table 1 at 18 *M* and 285.8 F is $145 - (0.186 \times 285.8) = 92$ Btu per pound. The heat of vaporization of water from steam tables at 285.8 F is 920 Btu per pound. Therefore the heat of vaporization of water from the solution is $920 + 92 = 1012$ Btu per pound.

Example 4. One thousand pounds of air per minute at 100 F dry-bulb with a dew-point of 70 F and a relative humidity of 39 per cent is passed over 18 *M* lithium chloride solution. The rate of flow of the solution is 200 gpm and the entering temperature is 80 F. The air leaves the absorber at 85 F dry-bulb and dew-point of 35 F. Calculate

(*a*) the heat to be removed from the lithium chloride solution to maintain these conditions, and (*b*) the temperature rise of the solution in passing through the absorber.

Solution. (*a*) The enthalpy of the entering air at 100 F dry-bulb and 39 per cent relative humidity = $h_a + \mu h_{as}$ = 24.00 + (0.39 × 47.40) = 42.49 Btu per pound (Table 6, Chapter 1).

The relative humidity of the air leaving at 85 F dry-bulb and 35 F dew-point is 16.7 per cent (psychrometric chart). Enthalpy of leaving air = 20.39 + (0.167 × 28.85) = 25.21 Btu per pound (Table 6, Chapter 1).

Heat to be extracted from air = 1000 (42.49 − 25.21) = 17,280 Btu per minute. Heat of mixing = 145 − (0.186 × 80) = 130 Btu per pound of moisture removed. From Table 6, Chapter 1, the moisture removal per pound of air = 0.01574 − 0.00426 = 0.01148 lb. Heat of mixing for 1000 lb of air = 1000 × 0.01148 × 130 = 1492 Btu. Total heat extraction = 17,280 + 1492 = 18,772 Btu per minute.

(*b*) The weight of solution circulated is 200 × 1.27 (Table 1) × 8.33 = 2116 lb per minute. Its heat capacity = 2116 × 0.631 (Table 1) = 1335 Btu per minute per degree Fahrenheit. The temperature rise = 18,772 ÷ 1335 = 14.1 F.

Table 3. Density of Lithium Chloride Solutions

Concentration Pound Mols (42.4 lb) $LiCl$ per 1000 lb Water	Temperature Deg F						
	0	50	100	150	200	250	300
0							
2		1.045	1.037	1.026	1.012		
4	1.090	1.085	1.076	1.064	1.052		
6	1.124	1.119	1.111	1.100	1.087		
8	1.156	1.150	1.143	1.132	1.122		
10	1.188	1.181	1.172	1.162	1.152	1.142	
12	1.217	1.209	1.199	1.188	1.178	1.168	
14	1.242	1.235	1.225	1.214	1.203	1.192	
16		1.257	1.248	1.236	1.226	1.215	
18		1.279	1.270	1.259	1.248	1.237	
20			1.291	1.280	1.279	1.568	
22				1.310	1.289	1.278	1.267
24				1.317	1.307	1.296	1.286
26					1.313	1.312	1.302
28					1.338	1.327	1.318
30						1.34	1.33
32							1.35

SOLID DEHYDRATION METHODS

One type of equipment suitable for producing dehydration with solid drying agents utilizes an apparatus with continuous operating rotating beds or dampers as illustrated in Fig. 4. The apparatus consists essentially of a cylinder or drum filled with dehydrating material. Air flow through the drum is directed by baffles which permit three independent air streams to flow through the adsorbing material. One air stream consists of the wet air which is to be dehydrated. The second is heated activation air used for drying that part of the dehydrating material which has become saturated. The third air stream purges away the products of combustion left in the bed from the activation cycle (when direct fired) and cools the bed to a temperature low enough to permit pickup of moisture when that part of the bed returns to the dehydration cycle.

In the rotating bed apparatus, the baffle sheets are stationary and the screened bed rotates at a definite speed to permit the proper time of

contact in the activation, purge and dehydration cycles. In the rotating damper apparatus, the bed remains stationary and a sectionalized damper rotates. This rotating damper produces the same effect as if the stationary baffles previously mentioned rotated.

Clean heated air for activation is supplied at temperatures normally ranging from 300 to 450 F. Any source of heat such as high pressure steam coils, electric heaters or oil or coal-fired air heat interchangers can, therefore, be used for heating the air. Direct-fired heaters are limited to gas since there must be no combustion products to contaminate the adsorbent.

Another type of solid dehydrating equipment uses two complete sets of stationary adsorbing beds, arranged so that one set is dehydrating the air while the other set is being activated. With the dampers in the position shown in Fig. 5, air to be dried flows through one set of beds and is dehydrated, while activation air is heated and circulated through the

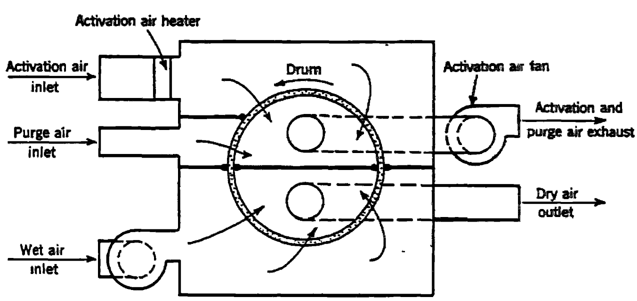


FIG. 4. SOLID ADSORBENT DEHYDRATOR—ROTATING BED TYPE

other set. After activation is complete, the beds are purged by shutting off the activation air heaters and allowing unheated air to circulate through them.

After the beds which are dehydrating have adsorbed moisture to a degree which begins to impair performance, a timer-controller causes the dampers to rotate to the opposite side. Thus the beds which on the previous cycle were adsorbing have activation air circulated through them, and vice versa. Activation air is heated in the same manner as with continuous equipment.

LIQUID DEHYDRATING METHODS

One type of system utilizing liquid dehydrating agents includes an external interchanger having essential parts consisting of a dehydration contactor, a solution concentrator, a solution heater and a solution cooler, all as shown in Fig. 6. The dehydrator contactor is located in the wet air stream. The air to be conditioned is brought into contact with an aqueous brine solution having a vapor pressure below that of the entering air, resulting in a transfer of moisture (latent heat). As previously described, this results in a conversion of latent to sensible heat, raising both air and solution temperatures. This temperature rise is kept down

by precooling the brine in the solution cooler to a predetermined temperature, which is usually below that of the air, by city, well or chilled water.

The excess water of condensation, which dilutes the brine, is removed in the solution concentrator. This is a low pressure steam heat exchanger which over-concentrates a portion of the weak liquor and returns it to the main brine reservoir for re-pumping. The concentrator operates in the manner of an evaporative condenser, whereby moisture is evaporated from the brine by the heating coils into a stream of regeneration air, taken

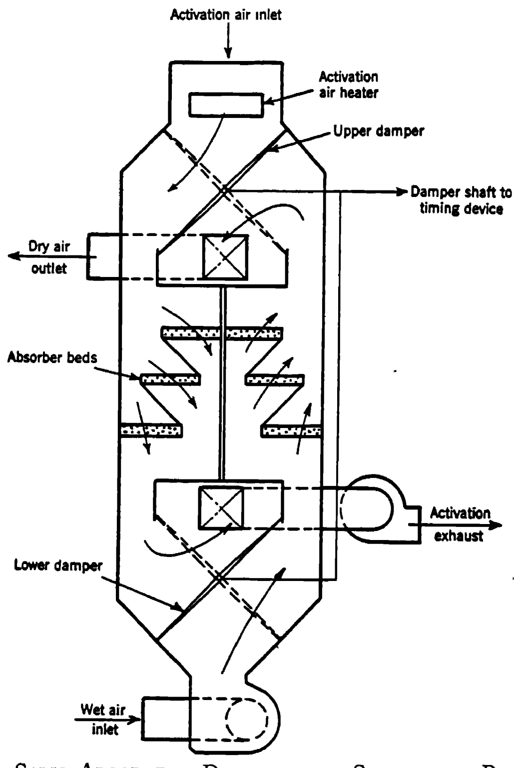


FIG. 5. SOLID ADSORBENT DEHYDRATOR—STATIONARY BED TYPE

from and rejected to the outside atmosphere. Low pressure steam is normally used for heating the brine. When it is desirable or necessary to use gas or electricity, an auxiliary low pressure steam boiler is usually added to the equipment. Concentrators operating on a simple boiler principle have not as yet been commercially practical.

It should be noted that the solution concentration phase is the reverse of the dehydration process. During concentration the aqueous vapor pressure of the solution is greater than that of the surrounding air, while during dehydration, the reverse is the case. Utilization of this principle permits winter humidification, by heating (instead of cooling) the solution pumped to the contactor. Water is thereby evaporated into, instead of

condensed out of, the conditioned air stream. This requires dilution of the brine externally to the contactor, rather than concentration.

Another type of liquid dehydrator utilizes an integral interchanger which employs the same type of solution concentrator as described for the system with the external interchanger. However, the dehydration contactor and solution cooler are combined by placing a cooling coil directly in the wet air stream. This coil provides the contacting surface between air and the warm concentrated solution which is sprayed over the cooling surface. By circulating a cooling medium through this coil, control of solution temperature (hence its vapor pressure) is accomplished directly in the air stream.

ESTIMATING OF LOADS

Where equipment is used which removes sensible and latent heat simultaneously such as a chilled water or direct expansion dehumidifier,

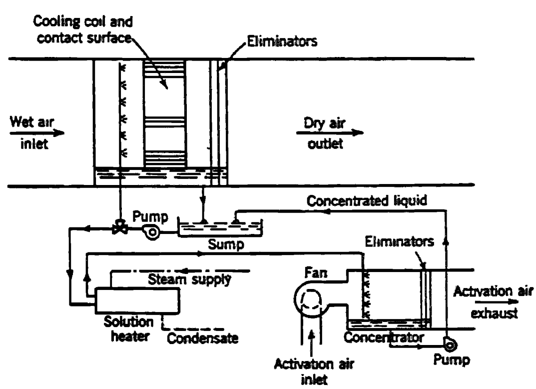


FIG. 6. LIQUID ABSORBENT DEHYDRATOR IN WHICH SOLUTION COOLER AND CONTACTOR ARE COMBINED

the basis of selection is usually the *maximum total heat load.* The operating characteristics of such equipment normally produce satisfactory dew-points with adequate capacity at other loads, including the maximum latent load which occurs at a less-than-maximum total load. With dehydration equipment in which moisture removal is achieved independently of sensible cooling, it is necessary that equipment be chosen for the maximum load of each functional element. The sensible cooler should be selected for the maximum sensible load; the dehydrator for the maximum latent load. These loads need not occur simultaneously, and in fact, rarely do.

In estimating the maximum latent heat load for comfort applications, it is considered good practice to select an outside design dew-point for the locality which is exceeded on not more than 5 per cent of the days during the season. A smaller percentage, or even the maximum dew-point recorded, may be advisable for rigorous industrial applications.

Due consideration should also be given to moisture seepage through building materials and vapor infiltration through openings. These items

become important at dew-points below 50 F and have an extreme effect upon load and equipment selection at dew-points below 35 F.

LOCATION OF EQUIPMENT

All types of dehydration equipment are, in general, applicable in one of several possible locations in the system air flow diagram. The choice of the type of equipment and its location is dependent upon the work to be performed, the capacity of the dehydration equipment to remove moisture and the type of energy available for activation or regeneration.

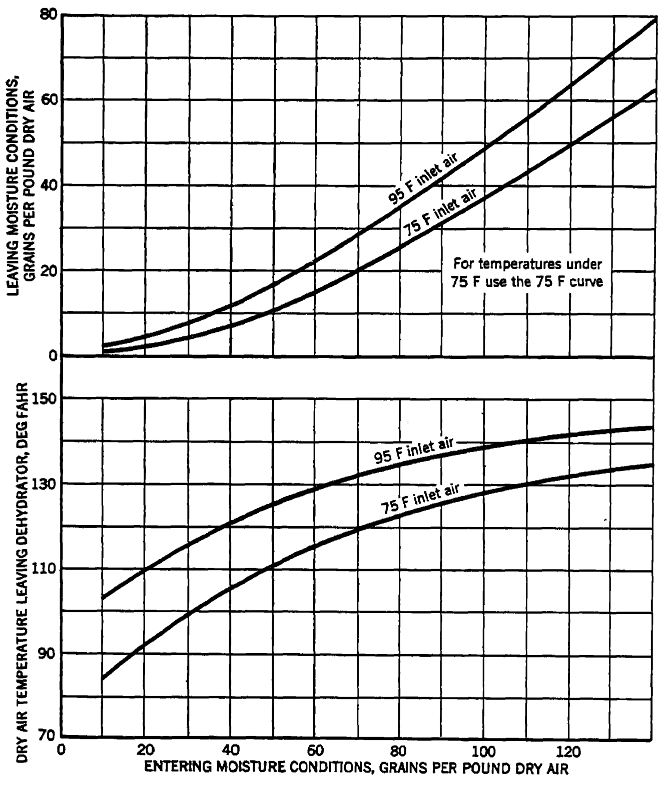


FIG. 7. SILICA GEL DEHYDRATOR PERFORMANCE DATA

Dehydration apparatus may be located: (*a*) to treat outside air only, (*b*) to treat return air only, or (*c*) to treat a mixture of outside and return air.

EQUIPMENT AUXILIARIES

Precoolers. When cold water is available, it is generally economical to use this water in a precooling coil in the outside air stream. The dehumidifying accomplished by this coil reduces the load on the dehydrator; and moreover, lowering the temperature of the inlet air to the dehydrator results in a higher dehydrator moisture removal efficiency.

Dry Air Coolers. Particularly with the solid adsorbent process, and to a lesser extent with liquid absorbents, a dry air cooler is employed to remove sensible heat from the dehydrated air whenever it leaves the dehydrator at an elevated temperature. A cooling coil using city water is usual practice, and is considered economical whenever the difference between effluent air and entering water temperatures is greater than 15 F.

Sensible Heat Coolers. Since the normal conditioning system requires sensible heat removal, auxiliary equipment may be needed for this function. This is almost always in the form of cooling surface using water, brine or direct expansion refrigerant. It is located on the leaving side of the dehydrator, but frequently treats in addition a large volume of room air which is not circulated through the dehydrator for moisture reduction.

CONTROLS

The use of dehydration equipment makes possible the use of a relatively simple control system with a humidistat or, alternatively, a wet-bulb controller, to regulate the operation of the dehydrator, and a thermostat

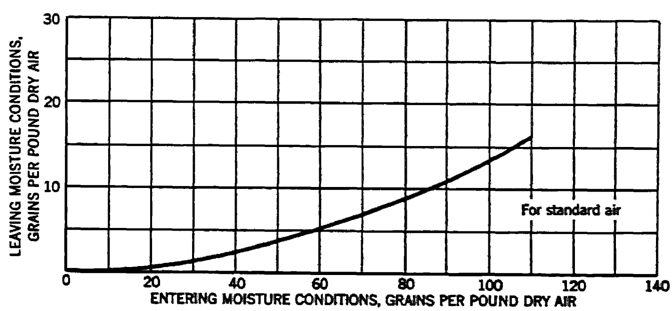


Fig. 8. Activated Alumina Dehydrator Performance Data

to control the sensible cooling apparatus. Functionally, the relative humidity control may consist of one of the following:

1. *Stop—Start*—Where the humidistat starts the dehydrator on rising humidity and stops it on falling humidity.

2. *Bypass*—Where the humidistat modulates face and bypass dampers located at the wet air inlet of the dehydrator. Thus the quantity of air passing through the dehydrator is proportioned in accordance with the change in latent heat load.

3. *Vapor Pressure Control* (used with liquid absorbents)—Where the humidistat directly controls the temperature or concentration of the contacting solution, thereby matching the latent heat removal to the load requirement.

EQUIPMENT PERFORMANCE

It is recognized that, whereas the curves relating temperature and vapor pressure of the several dehydration agents (Figs. 1, 2 and 3) accurately define the equilibrium limits for these materials, these curves cannot be used for predicting performance of available equipment. This is because (*a*) the materials themselves can only be utilized efficiently within certain ranges of moisture concentration, and (*b*) the degree to

which the vapor pressure of the air being treated approaches that at the surface of the material depends upon the completeness of the contact. For this reason, actual moisture removing capacity is determined from performance curves of the several materials under practical conditions of temperature, concentration and contacting efficiency as shown in Figs. 7, 8 and 9. While these curves by no means explore all the performance possibilities, they may be considered to be representative of sound design

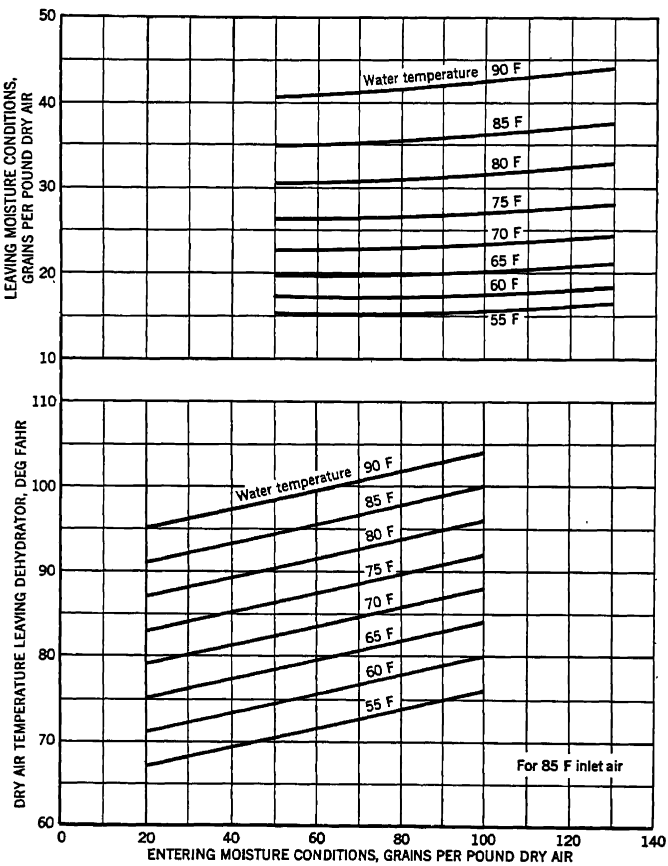


Fig. 9. Lithium Chloride Contactor Performance Data

and application practice. Representative data on heat input and water consumption are given in Table 4.

ECONOMICS OF DEHYDRATION

Almost all summer comfort air conditioning, as well as much industrial and commercial air conditioning, requires both of the functions of sensible and latent heat removal from air. Each of the methods of cooling and dehumidification has as its objective either the removal of sensible or latent heat, or both simultaneously. Choice of method and

medium therefore depends solely on whether (*a*) method and medium are physically able to accomplish the desired result with practical equipment, and (*b*) method and medium are justifiable economically.

Referring particularly to the problem of moisture removal, it may be stated that either dehumidification using chilled water, brine or direct expansion refrigerant, or dehydration using solid or liquid agents, is *equally practical* from the viewpoint of engineering performance for the vast majority of comfort and for a great many industrial applications;

TABLE 4. PERFORMANCE COEFFICIENTS AND WATER REQUIREMENTS OF SOLID AND LIQUID DEHYDRATION PROCESSES

Moisture Grains per Lb		Temperature Deg F		Thermal Performance Ratio[a]	Cooling Water Consumption[b]	
In	Out	In	Out		Temp. Deg F	Gpm per 1000 Btu per Hr of Latent Heat
Solid Adsorbent[c]						
130	65	85	138	0.39		
130	45	85	154	0.35		
65	30	85	118	0.28		
45	12	85	115	0.23		
Liquid Absorbent[d]						
130	65	85	99	0.45	85	0.23
130	45	85	101	0.45	85	0.44
130	45	85	87	0.41	60	0.26
65	30	85	83	0.41	60	0.64

[a]Thermal performance ratio is defined as latent heat removed from entering air divided by the heat input into the regenerator. Latent heat may be actually abstracted from system or converted to sensible heat, depending on process. No credit is given in the performance ratio for abstraction of sensible heat; where it occurs, it is considered a by-product effect. Values are approximate and, while they can be construed as typical, may vary considerably with design and economic application.

[b]Cooling water shown is that required solely to produce the latent heat removal. Additional water is required by both processes for reducing effluent temperatures below those listed. Gallons per minute shown are necessarily an economic approximation, weighing amount of surface against water consumption.

[c]Solid adsorbent based on the use of gas for activation.

[d]Liquid absorbent based on the use of steam at 12 lb per square inch gage for regeneration.

that is, their fields of application overlap. It cannot properly be stated that one method or material is superior *functionally* to another, since all have the same objective, and each is capable of attaining that objective.

For this reason, the choice of method and agent can normally be considered strictly on its economic merits. The choice of dehydration, mechanical refrigeration, natural cold water or ice, must be justified by the initial investment of available equipment, the availability and cost of prime energy sources, and the charges properly allocable to space occupied, labor of operation, maintenance, etc.

ECONOMIC COMPARISONS

It is evident that it is not possible to set forth definite rules governing the choice of dehydration equipment in preference to other methods of

dehumidification. It is only possible to state certain general conditions which tend to make dehydration favorable or unfavorable.

Dehydration tends to be favorable where:

1. Steam or gas is available at a cost substantially lower than electricity.
2. Required dry-bulb temperature is high or unimportant in comparison to maintenance of proper relative humidity.
3. Sensible cooling can be supplied by low cost city, well, or river water available at the proper temperature. For comfort conditioning, this temperature cannot normally be higher than 65 F.
4. An abnormally high room latent heat load or a large outside air latent load is encountered (such as in a dance hall, theater, restaurant, etc.).
5. Abnormally low room dew-points are required (such as 40 F or lower for some manufacturing operations).
6. Low temperature water is available but high in cost or limited in quantity.
7. In low temperature driers a complementary heat exchange can be utilized. In such cases, the sensible heat of the dry air from the dehydrator is reduced by the evaporation of moisture within the drier.

Of the factors just enumerated Item 1 is the most important influence, and if favorable, it indicates the desirability of considering dehydration. The other items are of lesser importance as criteria, but each has a direct influence in the economic considerations.

Dehydration tends to be unfavorable where:

1. Electricity is low in cost.
2. Normal comfort dew-points are required with a preponderantly sensible heat load.
3. Mechanical refrigeration is required for sensible heat removal.
4. Water temperature is too high for sensible heat removal. For comfort conditioning, this usually means water above 65 F.
5. Water is available in adequate quantity and at such temperature that it can be used directly for both sensible and latent removal, or can be further chilled more cheaply by mechanical refrigeration. For comfort conditioning, this normally means water below 55 F.

No single item just mentioned will necessarily disqualify dehydration, but will tend to require several favorable factors to make it a possibility for selection.

The previously outlined criteria are general and inclusive. When analyzed with respect to the possible fields of application, it is evident that dehydration equipment can be used, within its legitimate economic limits, for: air conditioning for human comfort, commercial cooling for food products requiring low humidities, industrial air conditioning for processes, and industrial drying. Attention is called to those particularly favorable industrial conditioning and drying applications in which the dried air can be used at effluent temperature without further treatment.

REFERENCES

Direct Evaporative Cooling for Homes in the Southwest, by A. J. Rummel (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, October, 1940, p. 621).

Chemical Dehumidification Agents, by F. R. Bichowsky (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, October, 1940, p. 627).

Chapter 24

REFRIGERATION

Mechanical Refrigeration, Characteristics of Compression System, Absorption Systems, Expansion Valves, Condensers, Evaporators and Coolers, Refrigerant Pipe Sizes, Ice Systems, Equipment Selection, Reverse Cycle

COOLING and dehumidification in air conditioning work usually requires refrigeration equipment. The localities where cold water from a natural source is at a sufficiently low temperature for comfort air conditioning are rare, and evaporative cooling is generally restricted to sections of the country where humidities are naturally low.

The important difference between the refrigeration equipment used for comfort air conditioning and that used for commercial refrigeration is the use of a relatively higher evaporator temperature. This temperature is usually above freezing in air conditioning refrigeration equipment. The higher evaporator temperature (that is high suction pressure) affects the design of the system used, and makes possible the use of systems that are not always practical for commercial refrigeration.

MECHANICAL REFRIGERATION

The fundamentals of mechanical refrigeration systems are similar, although they differ in the methods used for compression of the refrigerant vapor.

Refrigerant vapor, usually saturated or slightly superheated, is drawn into the compressor as diagrammed in Fig. 1. It is then compressed and discharged at a higher pressure to a condenser. The vapor is condensed as it contacts a heat transfer surface over which is flowing a cooling medium such as water, air or a combination of the two. The liquid refrigerant flows to the evaporator through an expansion valve which reduces its pressure and regulates its flow. In the evaporator, the refrigerant absorbs heat from the medium which is to be cooled. When this medium is water or brine, the evaporator is known as a water or brine cooler and the refrigeration system, if used for air cooling, is known as an indirect system. When the medium cooled is air, the evaporator is known as a direct expansion cooler and the system is known as a direct expansion system.

Fundamentally, the function of the system is to absorb heat at one temperature and pump it to a higher temperature, where it may be

Table 1. Properties of Ammonia

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	30.42	0.02419	9.116	42.9	611.8	0.0975	1.3352	666.8	1.4439	720.3	1.5317
2	31.92	0.02424	8.714	45.1	612.4	0.1022	1.3312	667.6	1.4400	721.2	1.5277
4	33.47	0.02430	8.333	47.2	613.0	0.1069	1.3273	668.4	1.4360	722.2	1.5236
5	34.27	0.02432	8.150	48.3	613.3	0.1092	1.3253	668.8	1.4340	722.6	1.5216
6	35.09	0.02435	7.971	49.4	613.6	0.1115	1.3234	669.3	1.4321	723.1	1.5196
8	36.77	0.02440	7.629	51.6	614.3	0.1162	1.3195	670.1	1.4281	724.1	1.5155
10	38.51	0.02446	7.304	53.8	614.9	0.1208	1.3157	670.9	1.4242	725.0	1.5115
12	40.31	0.02451	6.996	56.0	615.5	0.1254	1.3118	671.7	1.4205	725.9	1.5077
14	42.18	0.02457	6.703	58.2	616.1	0.1300	1.3081	672.5	1.4168	726.8	1.5039
16	44.12	0.02462	6.425	60.3	616.6	0.1346	1.3043	673.4	1.4130	727.8	1.5001
18	46.13	0.02468	6.161	62.5	617.2	0.1392	1.3006	674.2	1.4093	728.7	1.4963
20	48.21	0.02474	5.910	64.7	617.8	0.1437	1.2969	675.0	1.4056	729.6	1.4925
22	50.36	0.02479	5.671	66.9	618.3	0.1483	1.2933	675.8	1.4021	730.5	1.4889
24	52.59	0.02485	5.443	69.1	618.9	0.1528	1.2897	676.6	1.3985	731.4	1.4853
26	54.90	0.02491	5.227	71.3	619.4	0.1573	1.2861	677.3	1.3950	732.4	1.4816
28	57.28	0.02497	5.021	73.5	619.9	0.1618	1.2825	678.1	1.3914	733.3	1.4780
30	59.74	0.02503	4.825	75.7	620.5	0.1663	1.2790	678.9	1.3879	734.2	1.4744
32	62.29	0.02508	4.637	77.9	621.0	0.1708	1.2755	679.7	1.3846	735.1	1.4710
34	64.91	0.02514	4.459	80.1	621.5	0.1753	1.2721	680.4	1.3812	736.0	1.4676
36	67.63	0.02521	4.289	82.3	622.0	0.1797	1.2686	681.2	1.3779	736.8	1.4643
38	70.43	0.02527	4.126	84.6	622.5	0.1841	1.2652	681.9	1.3745	737.7	1.4609
39	71.87	0.02530	4.048	85.7	622.7	0.1863	1.2635	682.3	1.3729	738.2	1.4592
40	73.32	0.02533	3.971	86.8	623.0	0.1885	1.2618	682.7	1.3712	738.6	1.4575
41	74.80	0.02536	3.897	87.9	623.2	0.1908	1.2602	683.1	1.3696	739.0	1.4559
42	76.31	0.02539	3.823	89.0	623.4	0.1930	1.2585	683.4	1.3680	739.5	1.4542
44	79.38	0.02545	3.682	91.2	623.9	0.1974	1.2552	684.2	1.3648	740.4	1.4510
46	82.55	0.02551	3.547	93.5	624.4	0.2018	1.2519	684.9	1.3616	741.3	1.4477
48	85.82	0.02558	3.418	95.7	624.8	0.2062	1.2486	685.6	1.3584	742.2	1.4445
50	89.19	0.02564	3.294	97.9	625.2	0.2105	1.2453	686.4	1.3552	743.1	1.4412
52	92.66	0.02571	3.176	100.2	625.7	0.2149	1.2421	687.1	1.3521	744.0	1.4382
54	96.23	0.02577	3.063	102.4	626.1	0.2192	1.2389	687.8	1.3491	744.8	1.4351
56	99.91	0.02584	2.954	104.7	626.5	0.2236	1.2357	688.5	1.3460	745.7	1.4321
58	103.7	0.02590	2.851	106.9	626.9	0.2279	1.2325	689.2	1.3430	746.5	1.4290
60	107.6	0.02597	2.751	109.2	627.3	0.2322	1.2294	689.9	1.3399	747.4	1.4260
62	111.6	0.02604	2.656	111.5	627.7	0.2365	1.2262	690.6	1.3370	748.2	1.4231
64	115.7	0.02611	2.565	113.7	628.0	0.2408	1.2231	691.3	1.3341	749.1	1.4202
66	120.0	0.02618	2.477	116.0	628.4	0.2451	1.2201	691.9	1.3312	749.9	1.4172
68	124.3	0.02625	2.393	118.3	628.8	0.2494	1.2170	692.6	1.3283	750.8	1.4143
70	128.8	0.02632	2.312	120.5	629.1	0.2537	1.2140	693.3	1.3254	751.6	1.4114
72	133.4	0.02639	2.235	122.8	629.4	0.2579	1.2110	694.0	1.3226	752.4	1.4086
74	138.1	0.02646	2.161	125.1	629.8	0.2622	1.2080	694.6	1.3199	753.3	1.4059
76	143.0	0.02653	2.089	127.4	630.1	0.2664	1.2050	695.3	1.3171	754.1	1.4031
78	147.9	0.02661	2.021	129.7	630.4	0.2706	1.2020	695.9	1.3144	755.0	1.4004
80	153.0	0.02668	1.955	132.0	630.7	0.2749	1.1991	696.6	1.3116	755.8	1.3976
82	158.3	0.02675	1.892	134.3	631.0	0.2791	1.1962	697.2	1.3089	756.6	1.3949
84	163.7	0.02684	1.831	136.6	631.3	0.2833	1.1933	697.8	1.3063	757.4	1.3923
86	169.2	0.02691	1.772	138.9	631.5	0.2875	1.1904	698.5	1.3040	758.3	1.3896
88	174.8	0.02699	1.716	141.2	631 8	0.2917	1.1875	699.1	1.3010	759.1	1.3870
90	180.6	0.02707	1.661	143.5	632.0	0.2958	1.1846	699.7	1.2983	759.9	1.3843
92	186.6	0.02715	1.609	145.8	632.2	0.3000	1.1818	700.3	1.2957	760.7	1.3818
94	192.7	0.02723	1.559	148.2	632.5	0.3041	1.1789	700.9	1.2932	761.5	1.3793
96	198.9	0.02731	1.510	150.5	632.6	0.3083	1.1761	701.5	1.2906	762.2	1.3768
98	205.3	0.02739	1.464	152.9	632.9	0.3125	1.1733	702.1	1.2881	763.0	1.3743
100	211.9	0.02747	1.419	155.2	633.0	0.3166	1.1705	702.7	1.2855	763.8	1.3718
102	218.6	0.02756	1.375	157.6	633.2	0.3207	1.1677	703.3	1.2830	764.6	1.3693
104	225.4	0.02764	1.334	159.9	633.4	0.3248	1.1649	703.8	1.2805	765.3	1.3668
106	232.5	0.02773	1.293	162.3	633.5	0.3289	1.1621	704.3	1.2780	766.1	1.3643
108	239.7	0.02782	1.254	164.6	633.6	0.3330	1.1593	705.0	1.2755	766.9	1.3619
110	247.0	0.02790	1.217	167.0	633.7	0.3372	1.1566	705.5	1.2731	767.6	1.3596
112	254.5	0.02799	1.180	169.4	633.8	0.3413	1.1538	706.1	1.2708	768.3	1.3573
114	262.2	0.02808	1.145	171.8	633.9	0.3453	1.1510	706.6	1.2684	769.1	1.3550
116	270.1	0.02817	1.112	174.2	634.0	0.3495	1.1483	707.2	1.2661	769.8	1.3527
118	278.2	0.02827	1.079	176.6	634.0	0.3535	1.1455	707.7	1.2636	770.5	1.3503
120	286.4	0.02836	1.047	179.0	634.0	0.3576	1.1427	708.2	1.2612	771.3	1.3479
122	294.8	0.02846	1.017	181.4	634.0	0.3618	1.1400	708.6	1.2587	772.0	1.3455
124	303.4	0.02855	0.987	183.9	634.0	0.3659	1.1372	709.1	1.2563	772.8	1.3431
126	312.2	0.02865	0.958	186.3	633.9	0.3700	1.1344	709.6	1.2538	773.5	1.3407
128	321.2	0.02875	0.931	188.8	633.9	0.3741	1.1316	710.0	1.2513	774.2	1.3383

removed by an available cooling medium. In order to conserve refrigerant, virtually all refrigeration systems are completely closed and the same refrigerant is recirculated.

The fundamental heat equations (disregarding losses) which should be kept in mind are: (1) the heat absorbed in the evaporator plus the heat added to the refrigerant during compression equals the heat rejected by the condenser; (2) the heat added to the refrigerant during compression is equal to the input to the compressor shaft less the heat dissipated from the compressor to the surroundings.

In the case where the compressor is driven by an electric motor, the heat due to compression is equal to the motor input less the electrical motor losses, less the power transmission losses and less the heat dissipated from the compressor to the surroundings.

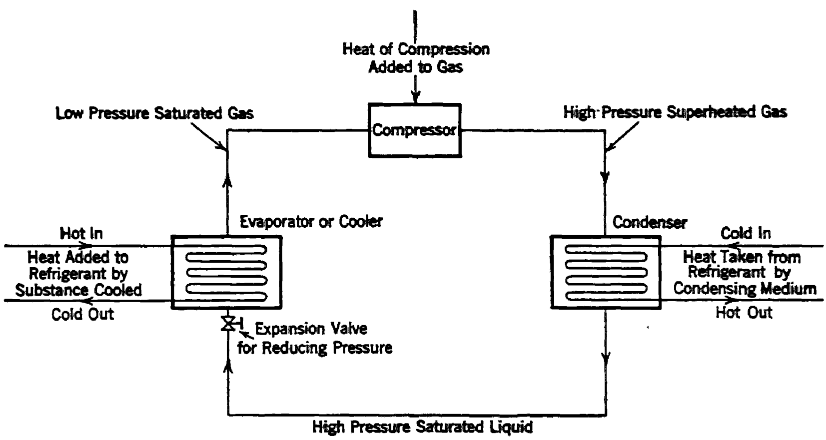


FIG. 1. MECHANICAL REFRIGERATION SYSTEM

Refrigerants

There are many substances which might be used as refrigerants in mechanical refrigeration systems, but in practice the choice is limited by a wide variety of considerations including availability, cost, safety, chemical stability and adaptability to the type of refrigerating system to be used.

In this chapter detailed consideration is limited to six substances, viz: ammonia, dichlorodifluoromethane (F_{12}), methyl chloride, carbon dioxide, monofluorotrichloromethane (F_{11}), and water, properties for each of which are given in Tables 1, 2, 3, 4, 5 and 6. Each table gives the principal physical properties of the saturated substance, and all are arranged in uniform fashion. In all except the water table, columns are included which give the heat content and entropy of the superheated vapor at two selected points. The first four refrigerants named are used in reciprocating and rotary compressors. The last two are used in centrifugal compressors. Water is also used in steam jet equipment.

Table 2. Properties of Dichlorodifluoromethane(F_{12})

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	23.87	0.0110	1.637	8.25	78.21	0.01869	0.17091	81.71	0.17829	85.26	0.18547
2	24.89	0.0110	1.574	8.67	78.44	0.01961	0.17075	81.94	0.17812	85.51	0.18529
4	25.96	0.0111	1.514	9.10	78.67	0.02052	0.17060	82.17	0 17795	85.76	0.18511
5	26.51	0.0111	1.485	9.32	78.79	0.02097	0.17052	82.29	0.17786	85.89	0.18502
6	27.05	0.0111	1.457	9.53	78.90	0.02143	0.17045	82.41	0.17778	86.01	0.18494
8	28.18	0.0111	1.403	9.96	79.13	0.02235	0.17030	82.66	0.17763	86 26	0.18477
10	29.35	0.0112	1.351	10.39	79.36	0.02328	0.17015	82.90	0.17747	86.51	0.18460
12	30.56	0.0112	1.301	10.82	79.59	0.02419	0.17001	83.14	0.17733	86.76	0.18444
14	31.80	0.0112	1.253	11.26	79.82	0.02510	0.16987	83.38	0.17720	87.01	0.18429
16	33.08	0.0112	1.207	11.70	80.05	0.02601	0.16974	83.61	0.17706	87.26	0.18413
18	34.40	0.0113	1.163	12.12	80.27	0.02692	0.16961	83.85	0.17693	87.51	0.18397
20	35.75	0.0113	1.121	12.55	80.49	0.02783	0.16949	84.09	0.17679	87.76	0.18382
22	37.15	0.0113	1.081	13.00	80.72	0.02873	0.16938	84.32	0.17666	88.00	0.18369
24	38.58	0.0113	1.043	13.44	80.95	0.02963	0.16926	84.55	0.17652	88.24	0.18355
26	40.07	0.0114	1.007	13.88	81.17	0.03053	0.16913	84.79	0.17639	88.49	0.18342
28	41.59	0.0114	0.973	14.32	81.39	0.03143	0.16900	85.02	0.17625	88.73	0.18328
30	43.16	0.0115	0.939	14.76	81.61	0.03233	0.16887	85.25	0.17612	88.97	0.18315
32	44.77	0.0115	0.908	15.21	81.83	0.03323	0.16876	85.48	0.17600	89.21	0.18303
34	46.42	0.0115	0.877	15.65	82.05	0.03413	0.16865	85.71	0.17589	89.45	0.18291
36	48.13	0.0116	0.848	16.10	82.27	0 03502	0.16854	85.95	0.17577	89.68	0.18280
38	49.88	0.0116	0.819	16.55	82.49	0.03591	0.16843	86.18	0.17566	89.92	0.18268
39	50.78	0.0116	0.806	16.77	82.60	0.03635	0.16838	86.29	0.17560	90.04	0.18262
40	51.68	0.0116	0.792	17.00	82.71	0.03680	0.16833	86.41	0.17554	90.16	0.18256
41	52.70	0.0116	0.779	17.23	82.82	0.03725	0.16828	86.52	0.17549	90.28	0.18251
42	53.51	0.0116	0.767	17.46	82.93	0.03770	0.16823	86.64	0.17544	90.40	0.18245
44	55.40	0.0117	0.742	17.91	83.15	0.03859	0.16813	86.86	0.17534	90.65	0.18235
46	57.35	0.0117	0.718	18.36	83.36	0.03948	0.16803	87.09	0.17525	90.89	0.18224
48	59.35	0.0117	0.695	18.82	83.57	0.04037	0.16794	87.31	0.17515	91.14	0.18214
50	61.39	0.0118	0.673	19.27	83.78	0.04126	0.16785	87.54	0.17505	91.38	0.18203
52	63.49	0.0118	0.652	19.72	83.99	0.04215	0.16776	87.76	0.17496	91.61	0.18193
54	65.63	0.0118	0.632	20.18	84.20	0.04304	0.16767	87.98	0.17486	91.83	0.18184
56	67.84	0.0119	0.612	20.64	84.41	0.04392	0.16758	88.20	0.17477	92.06	0.18174
58	70.10	0 0119	0.593	21.11	84.62	0.04480	0.16749	88.42	0.17467	92.28	0.18165
60	72.41	0.0119	0.575	21.57	84.82	0.04568	0.16741	88.64	0.17458	92.51	0.18155
62	74.77	0.0120	0.557	22.03	85.02	0.04657	0.16733	88.86	0.17450	92.74	0.18147
64	77.20	0.0120	0.540	22.49	85.22	0.04745	0.16725	89.07	0.17442	92.97	0.18139
66	79.67	0.0120	0.524	22.95	85.42	0.04833	0.16717	89.29	0.17433	93.20	0.18130
68	82.24	0.0121	0.508	23.42	85.62	0.04921	0.16709	89.50	0.17425	93.43	0.18122
70	84.82	0.0121	0.493	23.90	85.82	0.05009	0.16701	89.72	0.17417	93.66	0.18114
72	87.50	0.0121	0.479	24.37	86.02	0.05097	0.16693	89.93	0.17409	93.99	0.18106
74	90.20	0.0122	0.464	24.84	86.22	0.05185	0.16685	90.14	0.17402	94.12	0.18098
76	93.00	0.0122	0.451	25.32	86.42	0.05272	0.16677	90.36	0.17394	94.34	0.18091
78	95.85	0.0123	0.438	25.80	86.61	0.05359	0.16669	90.57	0.17387	94.57	0.18083
80	98.76	0.0123	0.425	26.28	86.80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
82	101.70	0.0123	0.413	26.76	86.99	0.05534	0.16655	90.98	0.17372	95.01	0.18068
84	104.8	0.0124	0.401	27.24	87.18	0.05621	0.16648	91.18	0.17365	95.22	0.18061
86	107.9	0.0124	0.389	27.72	87.37	0.05708	0.16640	91.37	0.17358	95.44	0.18054
88	111.1	0.0124	0.378	28.21	87.56	0.05795	0.16632	91.57	0.17351	95.65	0.18047
90	114.3	0.0125	0.368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
92	117.7	0.0125	0.357	29.19	87.92	0.05969	0.16616	91.97	0.17337	96.07	0.18033
94	121.0	0.0126	0.347	29.68	88.10	0.06056	0.16608	92.16	0.17330	96.28	0.18026
96	124.5	0.0126	0.338	30.18	88.28	0.06143	0.16600	92.36	0.17322	96.50	0.18018
98	128.0	0.0126	0.328	30.67	88.45	0.06230	0.16592	92.55	0.17315	96.71	0.18011
100	131.6	0.0127	0.319	31.16	88.62	0.06316	0.16584	92.75	0.17308	96.92	0.18004
102	135.3	0.0127	0.310	31.65	88.79	0.06403	0.16576	92.93	0.17301	97.12	0.17998
104	139.0	0.0128	0.302	32.15	88.95	0.06490	0.16568	93.11	0.17294	97.32	0.17993
106	142.8	0.0128	0.293	32.65	89.11	0.06577	0.16560	93.30	0.17288	97.53	0.17987
108	146.8	0.0129	0.285	33.15	89.27	0.06663	0.16551	93.48	0.17281	97.73	0.17982
110	150.7	0.0129	0.277	33.65	89.43	0.06749	0.16542	93.66	0.17274	97.93	0.17976
112	154.8	0.0130	0.269	34.15	89.58	0.06836	0.16533	93.82	0.17266	98.11	0.17969
114	158.9	0.0130	0.262	34.65	89.73	0.06922	0.16524	93.98	0.17258	98.29	0.17961
116	163.1	0.0131	0.254	35.15	89.87	0.07008	0.16515	94.15	0.17249	98.48	0.17954
118	167.4	0.0131	0.247	35.65	90.01	0.07094	0.16505	94.31	0.17241	98.66	0.17946
120	171.8	0.0132	0.240	36.16	90.15	0.07180	0.16495	94.47	0.17233	98.84	0.17939
122	176.2	0.0132	0.233	36.66	90.28	0.07266	0.16484	94.63	0.17224	99.01	0.17931
124	180.8	0.0133	0.227	37.16	90.40	0.07352	0.16473	94.78	0.17215	99.18	0.17922
126	185.4	0.0133	0.220	37.67	90.52	0.07437	0.16462	94.94	0.17206	99.35	0.17914
128	190.1	0.0134	0.214	38.18	90.64	0.07522	0.16450	95.09	0.17196	99.53	0.17906
130	194.9	0.0134	0.208	38.69	90.76	0.07607	0.16438	95.25	0.17186	99.70	0.17897
132	199.8	0.0135	0.202	39.19	90.86	0.07691	0.16425	95.41	0.17176	99.87	0.17889
134	204 8	0.0135	0.196	39.70	90.96	0.07775	0.16411	95.56	0.17166	100.04	0.17881
136	209.9	0.0136	0.191	40.21	91.06	0.07858	0.16396	95.72	0.17156	100.22	0.17873
138	215.0	0.0137	0.185	40.72	91.15	0.07941	0.16380	95.87	0.17145	100.39	0.17864
140	220.2	0.0138	0.180	41.24	91.24	0.08024	0.16363	96.03	0.17134	100.56	0.17856

Table 3. Properties of Methyl Chloride

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From —40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	18.73	0.0162	5.052	14.4	192.4	0.0328	0.4197	215.6	0.467	237.2	0.507
2	19.60	0.0162	4.856	15.1	193.1	0.0344	0.4196	216.2	0.466	237.7	0.505
4	20.47	0.0163	4.661	15.8	193.8	0.0360	0.4195	216.7	0.465	238.2	0.504
5	20.91	0.0163	4.563	16.2	194.1	0.0368	0.4195	217.0	0.464	238.5	0.503
6	21.39	0.0163	4.476	16.6	194.4	0.0376	0.4194	217.3	0.464	238.8	0.502
8	22.34	0.0164	4.303	17.3	195.1	0.0391	0.4193	217.9	0.463	239.4	0.501
10	23.30	0.0164	4.129	18.1	195.8	0.0407	0.4192	218.5	0.463	240.0	0.500
12	24.38	0.0164	3.984	18.8	196.3	0.0423	0.4184	219.0	0.462	240.5	0.499
14	25.46	0.0164	3.839	19.6	196.7	0.0439	0.4176	219.5	0.462	241.0	0.498
16	26.55	0.0165	3.693	20.3	197.2	0.0454	0.4168	220.0	0.461	241.5	0.498
18	27.63	0.0165	3.548	21.1	197.6	0.0472	0.4160	220.5	0.461	242.0	0.497
20	28.71	0.0166	3.403	21.8	198.1	0.0486	0.4152	221.0	0.460	242.5	0.496
22	29.98	0.0166	3.288	22.5	198.5	0.0501	0.4148	221.5	0.459	243.0	0.495
24	31.25	0.0166	3.172	23.3	198.9	0.0516	0.4143	222.0	0.459	243.6	0.495
26	32.53	0.0167	3.057	24.0	199.3	0.0532	0.4139	222.4	0.458	244.1	0.494
28	33.80	0.0167	2.941	24.8	199.7	0.0547	0.4134	222.9	0.458	244.7	0.494
30	35.07	0.0168	2.826	25.5	200.1	0.0562	0.4130	223.4	0.457	245.2	0.493
32	36.55	0.0168	2.734	26.2	200.5	0.0577	0.4124	223.9	0.456	245.7	0.492
34	38.03	0.0169	2.642	27.0	200.9	0.0592	0.4118	224.3	0.455	246.2	0.492
36	39.51	0.0169	2.549	27.7	201.4	0.0607	0.4111	224.8	0.455	246.7	0.491
38	40.99	0.0169	2.457	28.5	201.8	0.0622	0.4105	225.2	0.454	247.2	0.491
39	41.73	0.0170	2.411	28.8	202.0	0.0629	0.4102	225.5	0.453	247.4	0.490
40	42.47	0.0170	2.365	29.2	202.2	0.0637	0.4099	225.7	0.453	247.7	0.490
41	43.33	0.0170	2.328	29.6	202.4	0.0644	0.4096	225.9	0.453	248.0	0.490
42	44.18	0.0171	2.290	29.9	202.6	0.0651	0.4093	226.1	0.452	248.3	0.489
44	45.89	0.0171	2.216	30.7	203.0	0.0666	0.4087	226.6	0.451	248.8	0.489
46	47.61	0.0171	2.141	31.4	203.3	0.0680	0.4081	227.0	0.451	249.4	0.488
48	49.32	0.0172	2.067	32.2	203.7	0.0695	0.4075	227.5	0.450	249.9	0.488
50	51.03	0.0172	1.992	32.9	204.1	0.0709	0.4069	227.9	0.449	250.5	0.487
52	53.00	0.0172	1.931	33.7	204.4	0.0724	0.4063	228.2	0.448	251.0	0.486
54	54.97	0.0173	1.870	34.4	204.7	0.0739	0.4056	228.6	0.448	251.5	0.486
56	56.94	0.0173	1.810	35.2	205.1	0.0754	0.4050	228.9	0.447	252.0	0.485
58	58.91	0.0173	1.749	35.9	205.4	0.0769	0.4043	229.3	0.447	252.5	0.485
60	60.88	0.0174	1.688	36.7	205.7	0.0784	0.4037	229.6	0.446	253.0	0.484
62	63.13	0.0174	1.638	37.4	206.0	0.0798	0.4030	229.9	0.445	253.5	0.483
64	65.37	0.0174	1.588	38.2	206.3	0.0812	0.4024	230.3	0.444	254.0	0.483
66	67.62	0.0175	1.539	38.9	206.6	0.0827	0.4017	230.6	0.443	254.5	0.482
68	69.86	0.0175	1.489	39.7	206.9	0.0841	0.4011	231.0	0.442	255.0	0.482
70	72.11	0.0176	1.439	40.4	207.2	0.0855	0.4004	231.3	0.441	255.5	0.481
72	74.66	0.0176	1.398	41.1	207.5	0.0869	0.3998	231.6	0.440	256.0	0.480
74	77.21	0.0177	1.357	41.9	207.7	0.0883	0.3992	232.0	0.439	256.5	0.480
76	79.76	0.0177	1.315	42.6	208.0	0.0898	0.3985	232.3	0.439	256.9	0.479
78	82.31	0.0178	1.274	43.4	208.2	0.0912	0.3979	232.7	0.438	257.4	0.479
80	84.86	0.0178	1.233	44.1	208.5	0.0926	0.3973	233.0	0.437	257.9	0.478
82	87.74	0.0178	1.199	44.8	208.7	0.0940	0.3967	233.3	0.436	258.4	0.478
84	90.62	0.0179	1.165	45.6	209.0	0.0953	0.3960	233 6	0.435	258.9	0.477
86	93.50	0.0179	1.130	46.3	209.2	0.0967	0.3954	233.9	0.435	259.4	0.477
88	96.38	0.0180	1.096	47.1	209.5	0.0980	0.3947	234.2	0.434	259.9	0 476
90	99.26	0.0180	1.062	47.8	209.7	0.0994	0.3941	234.5	0 433	260.4	0.476
92	102.49	0.0180	1.033	48.6	209 9	0.1008	0.3935	234.8	0.433	260.8	0.476
94	105.72	0.0181	1.005	49.3	210.2	0.1022	0.3929	235.1	0.432	261.2	0.475
96	108.94	0.0181	0.9764	50.1	210.4	0.1035	0.3922	235.4	0.432	261.6	0.475
98	112.17	0.0182	0.9478	50.8	210.7	0.1049	0.3916	235.7	0.431	262.0	0.474
100	115.40	0.0182	0.9193	51.6	210.9	0.1063	0.3910	236.0	0.431	262.4	0.474
102	119.00	0.0183	0.8952	52.3	211.1	0.1076	0.3903	236.4	0.430	262.8	0.474
104	122.60	0.0183	0.8712	53.1	211.3	0.1090	0.3897	236.8	0.430	263.2	0.473
106	126.20	0.0184	0.8471	53.8	211.4	0.1103	0.3890	237.1	0.429	263.5	0.473
108	129.80	0.0184	0.8231	54.6	211.6	0.1117	0.3884	237.5	0.429	263.9	0.472
110	133.40	0.0185	0.7990	55.3	211.8	0.1130	0.3877	237.9	0.428	264.3	0.472
112	137.42	0.0185	0.7786	56.1	212.0	0.1144	0.3871	238.1	0.427	264.6	0.471
114	141.44	0.0185	0.7583	56.8	212.2	0.1157	0.3864	238.3	0.427	264.8	0.470
116	145.46	0.0186	0.7379	57.6	212.4	0.1171	0.3858	238.6	0.426	265.1	0.470
118	149.48	0.0186	0.7176	58.3	212.6	0.1184	0.3851	238.8	0.426	265.3	0.469
120	153.50	0.0187	0.6972	59.1	212.8	0.1198	0.3845	239.0	0.425	265.6	0.468

TABLE 4. PROPERTIES OF CARBON DIOXIDE

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	305.5	0.01570	0.29040	18.8	138.9	0.0418	0.3024	153.7	0.3342	167.5	0.3612
2	315.9	0.01579	0.28030	19.8	138.8	0.0440	0.3014	153.7	0.3330	167.6	0.3600
4	326.5	0.01588	0.27070	20.8	138.8	0.0461	0.3005	153.7	0.3318	167.7	0.3588
5	332.0	0 01592	0.26610	21.3	138.8	0.0472	0.3000	153.7	0.3312	167.7	0.3582
6	337.4	0.01596	0.26140	21.8	138.7	0.0483	0.2994	153.7	0.3306	167.8	0.3576
8	348.7	0.01605	0.25260	22.9	138.7	0.0504	0.2982	153.7	0.3293	167.9	0.3563
10	360.2	0.01614	0.24370	24.0	138.7	0.0526	0.2970	153.7	0.3281	168.0	0.3550
12	371.9	0.01623	0.23540	25.0	138.6	0.0548	0.2958	153.7	0.3270	168.1	0.3538
14	383.9	0.01632	0.22740	26.1	138.6	0.0571	0.2946	153.7	0.3259	168.2	0.3526
16	396.2	0.01642	0.21970	27.2	138.5	0.0593	0.2933	153.7	0.3249	168.3	0.3513
18	408.9	0.01652	0.21210	28.3	138.4	0.0616	0.2921	153.7	0.3238	168.5	0.3501
20	421.8	0.01663	0.20490	29.4	138.3	0.0638	0.2909	153.7	0.3227	168.6	0.3489
22	434 0	0.01673	0.19790	30.5	138.2	0.0662	0.2897	153.7	0.3214	168.7	0.3479
24	448.4	0.01684	0.19120	31.7	138.1	0.0686	0.2885	153.7	0.3202	168.8	0.3470
26	462.2	0.01695	0.18460	32.9	138.0	0.0710	0.2873	153.7	0.3189	168.9	0.3460
28	476.3	0.01707	0.17830	34.1	137.9	0.0734	0.2861	153.7	0.3177	169.0	0.3451
30	490.8	0 01719	0.17220	35.4	137.8	0.0758	0.2849	153.7	0.3164	169.1	0.3441
32	505.5	0.01731	0.16630	36.7	137.7	0.0781	0.2834	153.7	0.3158	169.2	0.3431
34	522.6	0.01744	0.16030	37.9	137.4	0.0804	0.2820	153.7	0.3151	169.3	0.3421
36	536.0	0.01759	0.15500	39.1	137.2	0.0828	0.2805	153.7	0.3145	169.4	0.3411
38	551.7	0.01773	0.14960	40.4	136.9	0.0851	0.2791	153.7	0.3138	169.5	0.3401
39	559.7	0.01780	0.14700	41.0	136.8	0.0862	0.2783	153.7	0.3135	169.5	0.3396
40	567.8	0.01787	0.14440	41.7	136.7	0.0874	0.2776	153.7	0.3132	169.6	0.3391
41	576.0	0.01794	0.14185	42.3	136.5	0.0887	0.2768	153.7	0.3127	169.6	0.3386
42	584.3	0.01801	0.13930	42.9	136.3	0.0899	0.2761	153.7	0.3122	169.7	0.3381
44	601.1	0.01817	0.13440	44.3	136.1	0.0924	0.2745	153.7	0.3112	169.8	0.3371
46	618.2	0.01834	0.12970	45.6	135.7	0.0950	0.2730	153.7	0.3101	169.9	0.3362
48	635.7	0.01851	0.12500	47.0	135.4	0.0975	0.2714	153.7	0.3091	170.0	0.3352
50	653.6	0.01868	0.12050	48.4	135.0	0.1000	0.2699	153.7	0.3081	170.1	0.3342
52	671.9	0.01887	0.11610	49.8	134.5	0.1027	0.2681	153.7	0.3069	170.2	0.3333
54	690.6	0.01906	0.11170	51.2	133.9	0.1054	0.2663	153.7	0.3057	170.3	0.3324
56	709.5	0.01927	0.10750	52.6	133.4	0.1081	0.2644	153.7	0.3046	170.5	0.3315
58	728.8	0.01948	0.10340	54.0	132.7	0.1108	0.2626	153.7	0.3034	170.6	0.3306
60	748.6	0.01970	0.09940	55.5	132.1	0.1135	0.2608	153.7	0.3022	170.7	0.3297
62	769.0	0.01995	0.09545	57.0	131.3	0.1164	0.2584	153.7	0.3012	170.8	0.3289
64	789.4	0.02020	0.09180	58.6	130.6	0.1194	0.2560	153.7	0.3002	170.9	0.3281
66	810.3	0.02048	0.08800	60.2	129.7	0.1223	0.2535	153.7	0.2991	171.0	0.3273
68	831.6	0.02079	0.08422	61.9	128.7	0.1253	0.2511	153.7	0.2981	171.1	0.3265
70	853.4	0.02112	0.08040	63.7	127.5	0.1282	0.2487	153.7	0.2971	171.2	0.3257
72	875.8	0.02152	0.07654	65.5	126.0	0.1321	0.2450	153.7	0.2962	171.3	0.3250
74	898.2	0.02192	0.07269	67.3	124.5	0.1360	0.2414	153.7	0.2953	171.4	0 3242
76	921.3	0.02242	0.06875	69.4	122.8	0.1398	0.2377	153.7	0.2945	171.5	0.3235
78	944.8	0.02300	0.06473	71.6	120.9	0.1437	0.2341	153.7	0.2936	171.6	0.3227
80	968.7	0.02370	0.06064	73.9	118.7	0.1476	0.2304	153.7	0.2927	171.7	0.3220
82	993.0	0.02456	0.05648	76.4	116.6	0.1578	0.2195	153.7	0.2920	173.8	0.3215
84	1017.7	0.02553	0.05223	79.4	113.9	0.1679	0.2087	153.7	0.2914	176.0	0.3209
86	1043.0	0.02686	0.04789	83.3	110.4	0.1781	0.1978	153.7	0.2907	178.2	0.3204
87.8	1069.9	0.03454	0.03454	97.0	97.0	0.1880	0.1880	153.7	0.2901	180.1	0.3199

Types of Compressors

There are many different types of compressors, using various refrigerants. Each type has its advantages for its particular application, and those generally used for air conditioning are of the following types:

1. Reciprocating compressors (commonly referred to as piston type).
2. Centrifugal compressors.
3. Steam jet.

Reciprocating compressors are available in a wide range of sizes and types. Any of a number of refrigerants, including dichlorodifluoromethane

(F_{12}), methyl chloride, ammonia, carbon dioxide, and sulphur dioxide may be used in reciprocating machines. The first of these is used extensively in direct expansion systems of comfort air conditioning.

Compressors may be classified into two general types, (*a*) open type, (*b*) enclosed type. If the driving mechanism is external to the compressor, then the shaft must be brought out through the crankcase and a shaft seal or stuffing box must be used to prevent escape of the refrigerant. This type of compressor is known as an open-type compressor. When the driving mechanism is located within the crankcase of the compressor in such a way as to avoid the necessity of a shaft seal, the compressor is known as the completely enclosed or hermetically sealed type.

Open type compressors may be further classified as belt driven and directly connected. A great number of direct-driven units are now being used which generally operate at higher rotational speeds than the belt-driven type.

The present tendency is toward forced lubrication of the bearings of compressors by means of an oil pump driven from the crankshaft, although there are many splash lubricated compressors on the market. The chief advantages of the forced lubricated compressor are that the lubrication system requires less energy for its operation than the splash type, the oil can be easily filtered before it enters the bearings, and less oil is usually required.

The compressor capacity must be selected for and matched to the maximum load for the installation on which it is to be used. Air-conditioning loads, however, vary over a wide range, and a wide fluctuation in air conditions may result during periods of light load if on-and-off control of full compressor capacity is used. To prevent such undesirable fluctuation, several methods are employed to vary the capacity of reciprocating compressors, such as:

1. By-passing one or more cylinders, of a multicylinder compressor, from discharge to suction.
2. Rendering the suction valves of one or more cylinders of a multicylinder compressor inoperative. This is usually accomplished by depressing the suction valves.
3. Varying the speed of the compressor, usually by using variable speed or two-speed electric motors.
4. Using clearance pockets to control the quantity of refrigerant pumped.
5. Restricting the suction inlet to one or more of the cylinders of a multicylinder compressor either by an automatic modulating valve or by an *on* and *off* valve.

All of these methods, with the exception of variable speed, result in a slightly lowered overall compressor efficiency when in use, since the mechanical losses remain constant whereas the quantity of refrigerant pumped is lowered.

Centrifugal compressors are used with very low pressure refrigerants; usually both evaporator and condenser work below atmospheric pressure. Water and monofluorotrichloromethane (F_{11}) are the refrigerants commonly used in centrifugal machines.

Compression of the refrigerant is accomplished by means of centrifugal force; therefore, this type of compressor is inherently suitable for large volumes of refrigerant at low pressure differentials. Two or more stages are usually required and high speeds are necessary to obtain good efficiency.

Table 5. Properties of Monofluorotrichloromethane (F_{11})

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht Ct.	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	98.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.460	16.00	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.020	19.10	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24.50	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0.01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2.090	26.70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0.01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

Table 6. Properties of Water

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From +32 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
32	0.0887	0.01602	3296.0	0.00	1073.0	0.0000	2.1826	1096.9	2.2277	1120.8	2.2688
35	0.1000	0.01602	2941.0	3.02	1074.4	0.0062	2.1724	1098.3	2.2172	1122.2	2.2581
40	0.1217	0.01602	2441.0	8.05	1076.8	0.0163	2.1555	1100.6	2.2000	1124.5	2.2406
45	0.1475	0.01602	2034.0	13.07	1079.2	0.0262	2.1390	1102.9	2.1832	1126.7	2.2234
50	0.1780	0.01602	1702.0	18.08	1081.5	0.0361	2.1230	1105.2	2.1667	1129.0	2.2066
55	0.2140	0.01603	1430.0	23.08	1083.9	0.0459	2.1073	1107.5	2.1506	1131.3	2.1902
60	0.2561	0.01603	1206.0	28.08	1086.2	0.0556	2.0920	1109.8	2.1349	1133.5	2.1742
65	0.3054	0.01604	1021.0	33.08	1088.6	0.0652	2.0771	1112.2	2.1196	1135.8	2.1585
70	0.3628	0.01605	868.0	38.07	1090.9	0.0746	2.0625	1114.5	2.1046	1138.1	2.1432
75	0.4295	0.01606	740.0	43.06	1093.2	0.0840	2.0483	1116.7	2.0900	1140.3	2.1283
80	0.507	0.01607	632.9	48.05	1095.5	0.0933	2.0344	1119.0	2.0758	1142.5	2.1138
85	0.596	0.01609	543.3	53.04	1097.8	0.1025	2.0208	1121.2	2.0619	1144.7	2.0996
90	0.698	0.01610	467.9	58.03	1100.0	0.1116	2.0075	1123.4	2.0483	1146.8	2.0857
95	0.815	0.01612	404.2	63.01	1102.3	0.1206	1.9946	1125.6	2.0350	1148.9	2.0721
100	0.949	0.01613	350.3	68.00	1104.6	0.1296	1.9819	1127.9	2.0220	1151.1	2.0588
105	1.101	0.01615	304.4	72.98	1106.8	0.1384	1.9695	1130.2	2.0093	1153.2	2.0458

For properties of steam at high temperatures, see Table 8, Chapter 1.

The evaporator is usually constructed as an integral part of the centrifugal type condensing unit, to chill water which is then circulated to the air conditioning system. This is done because it would not be economical to pipe these large volumes of refrigerant any distance.

Centrifugal compressors like reciprocating compressors can be divided into two general types, open and enclosed. In general, the open type compressor is geared to the driving mechanism, and operates at higher speed than the driving motor or turbine. A modern completely enclosed direct-driven, centrifugal compressor is illustrated in Fig. 2.

The compressor capacity can be varied by controlling the condensing pressure. This is accomplished by regulating the quantity and temperature of the condenser cooling water. The capacity falls off with increasing condensing pressure. Centrifugal compressors are seldom

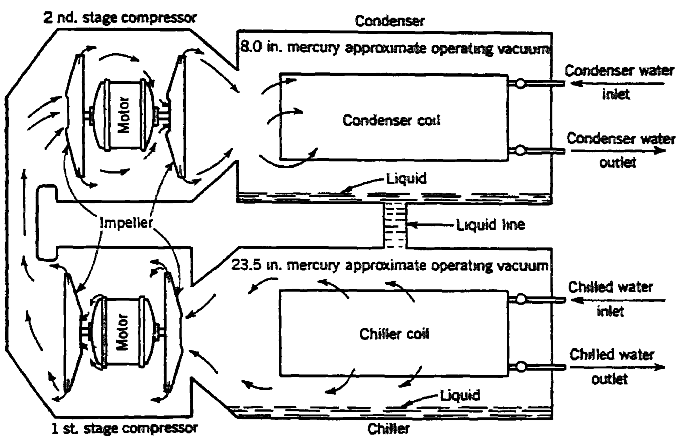


FIG. 2. ENCLOSED TYPE CENTRIFUGAL CONDENSING UNIT

built for less than 50 tons capacity, since it is not practical to make impellers which pump much less than the volume of refrigerant required for this tonnage.

The steam jet type of compressor, under certain circumstances, is desirable for use in air conditioning. Steam supplies directly the power used for compressing the refrigerant, thus eliminating the losses connected with other methods of supplying energy. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiency of the equipment is somewhat lower than that of the positive mechanical type compressor. The condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures. Steam jet boosters or compressors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

A diagrammatic representation of a typical steam ejector water cooling system is shown in Fig. 3. The figures correspond to an average representative system. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam. As this requires heat, and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the

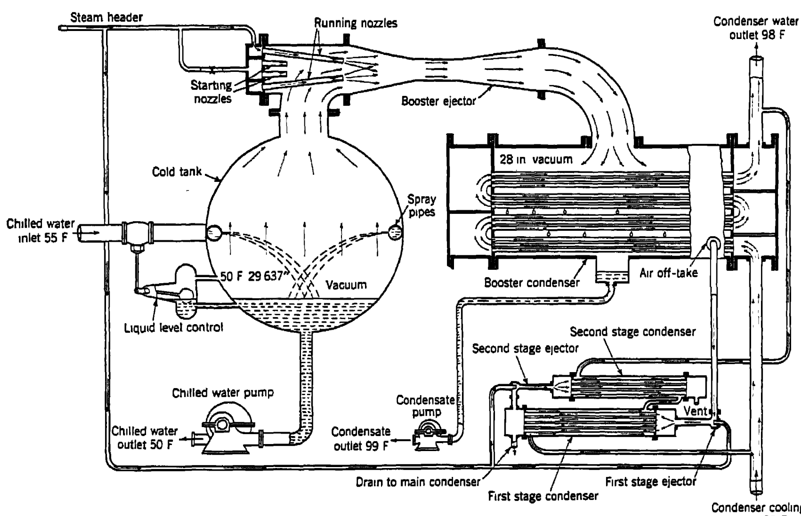


FIG. 3. DIAGRAMMATIC ARRANGEMENT OF STEAM JET VACUUM COOLING UNIT

desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed in the evaporator, plus any entrained air taken from the circulated water, to a somewhat higher absolute pressure and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture then passes from the ejector into the condenser.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A secondary condenser is necessary to condense the steam in the secondary jet.

Although steam jet vacuum cooling units have been built for as small as 5 to 6 tons capacity, a single booster of smaller than 15 tons capacity is difficult to build. They can readily be built for steam pressures of from

5 to 200 lb per square inch and condenser water temperatures as high as 90 F. The steam consumption in pounds per hour per ton of refrigeration increases rapidly as the booster steam pressure is lowered. For example, the lowering of the booster steam pressure from 200 to 90 lb per square inch results in an increase in steam consumption of approximately 5 per cent whereas a further decrease in booster steam pressure to 10 lb per square inch increases the steam consumption by approximately 72 per cent over that required at 200 lb per square inch.

The capacity of a steam jet system is usually controlled by controlling the number of boosters in use since the unit usually has several boosters operating on the same evaporator. Usually one booster is automatically

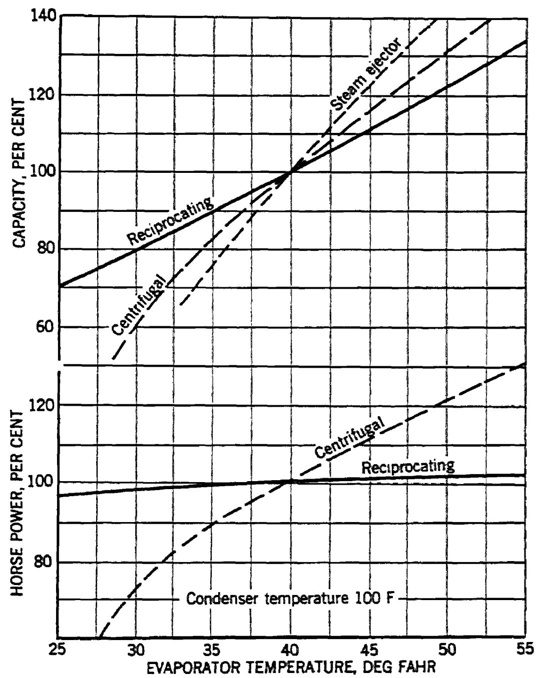


FIG. 4. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

controlled whereas the others are manually operated. The capacity is dependent, as for all compressors, upon the evaporator temperature, or in other words, the suction pressure. For example, the capacity is lowered approximately 17 per cent if the evaporator or chilled water temperature is lowered from 50 to 45 F. The capacity therefore can be controlled to some extent by regulating the evaporator temperature.

CHARACTERISTICS OF COMPRESSION SYSTEMS

The various types of compression systems have quite different characteristics of capacity and power with varying evaporator and condenser temperatures, as may be noted from curves in Figs. 4 and 5.

From Fig. 5 it may be observed that power requirements for the centrifugal compressor increase much more rapidly than for the reciprocating compressor with increase in evaporator temperature. Similarly, the capacities of the steam ejector and centrifugal compressors increase more rapidly than those of the reciprocating compressor with increase in evaporator temperature. Thus, both the steam jet and centrifugal machines tend to be more self-regulating than the reciprocating. It is also evident from Fig. 5 that the steam jet equipment is best suited for operation at high evaporator temperatures.

The effect of condenser temperature upon the power and capacity of the different types of compressors is shown in Fig. 6. It may be noted that the power required by the reciprocating compressor increases rapidly

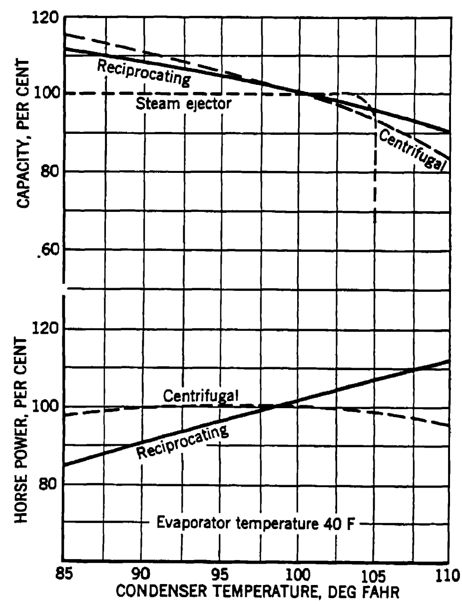


FIG. 5. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

with increase in condenser temperature, while the power curve for the centrifugal compressor is relatively flat. It is also evident that the capacity of the steam jet compressor is independent of condenser temperature until a certain point is reached where it drops to zero. As previously stated, steam jet equipment requires more condensing water than other types of compression systems. Consequently, steam jet systems are well suited to those applications where condensing water is cheap, or where condensing water is rather high in temperature.

ABSORPTION SYSTEMS

The fundamental rule governing the absorption (in a closed system) of a gas by a liquid is Raoult's Law, which states that at any given tem-

perature the ratio of the partial pressure of a volatile component in a solution to the vapor pressure of the pure component at the same temperature is equal to its mol fraction in the solution. The mol fraction in turn is equal to the number of mols of substance divided by the total number of mols present. The number of mols in a given weight of a compound is equal to the weight divided by the molecular weight.

This law applies strictly, only to what is known as an ideal solution, that is, one in which the inter-molecular forces between the substances present in the solution are equal. Actually, no such solutions exist, so that deviations from Raoult's Law are always found in practice. The deviation is called positive when the observed pressure is greater than that calculated from Raoult's Law, while the term negative deviation refers to the opposite case. Negative deviations are found wherever chemical attraction exists between the solvent and the solute. Positive

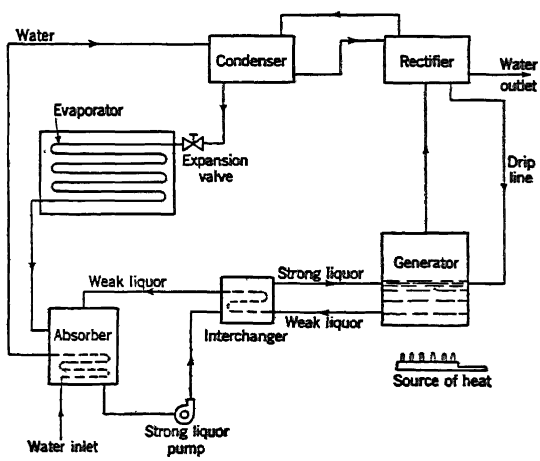


Fig. 6. Closed Absorption System

deviation occurs when there is a difference in the internal pressure of the components, chemical attraction between them being absent.

In order to make an effective absorption machine, large negative deviations from Raoult's Law must be shown by solutions of the refrigerant in the liquid absorbent, because the larger the negative deviation, the greater is the amount of refrigerant that can be cycled, using a given weight of absorbent. Cycling a large amount of refrigerant for a given weight of absorbent is important because of the heat required to raise the temperature of the mixture and disassociate the refrigerant and the absorbent. Only the latent heat of the refrigerant can be recovered for useful work.

Many refrigerant-absorbent combinations have been proposed and quite a number have been tested. A diagrammatic representation of a typical closed absorption system is outlined in Fig. 6. In this system a mixture of refrigerant and absorbent is evaporated in the generator, passes to an analyzer and rectifier where it is purified, and then to a condenser where the refrigerant and remaining absorbent is condensed. It

then passes through an expansion valve to an evaporator, where heat is absorbed from a cooling load. From the evaporator the vapor and residual absorbent passes to an absorber where it meets absorbent which is initially low (weak) in refrigerant concentration. The absorbent absorbs the vapor and the strong absorbent liquor is transferred to the generator through an interchanger with the weak liquor returning from the generator.

A cooling medium, ordinarily water, is used in the absorber to remove the heat of absorption and maintain the absorptive power of the absorber at a maximum.

Like the steam jet system, the absorption system compares most favorably when a cheap source of cooling water and steam or other heat is available. Unlike the steam jet system, the comparative performance is usually best with a wide range of temperature between the evaporator and absorber, since with a good refrigerant-absorbent combination, the amount of heat and water required for a given refrigerating effect increases slowly with an increase of evaporator-condenser temperature range.

At the present time the most used refrigerant-absorbent combinations are: (1) water and ammonia, and (2) dichloromonofluoromethane and dimethyl ether of tetraethylene glycol. With the latter combination the boiling points of the refrigerant and absorbent are sufficiently wide apart that almost pure refrigerant is obtained without the use of a rectifier.

EXPANSION VALVES

The thermostatic expansion valve is a device to regulate the flow of liquid refrigerant so that the evaporator will always be used to best advantage. The evaporator coil must be kept as full as possible without any chance of liquid refrigerant entering the suction line. The expansion valve accomplishes this by regulating the supply of refrigerant, so that the temperature of the gas leaving the evaporator is always slightly higher than the temperature of the boiling refrigerant inside of it. This difference in temperature between the outgoing (suction) gas and the liquid refrigerant in the evaporator is called the superheat of the gas.

The operation of the thermostatic expansion valve can best be explained by means of a diagram, Fig. 7. A small refrigerant charge in the control bulb exerts a pressure through the tube to the upper side of the diaphragm, which tends to open the valve.

The magnitude of this pressure is determined by the temperature of the suction gas leaving the evaporator, as the control bulb is attached to the suction line at this point and is at approximately the same temperature. The suction pressure in the evaporator is transmitted through the equalizer tap and exerts an opposing force on the other side of the diaphragm in the direction to close the valve. This pressure corresponds to the temperature of the boiling refrigerant. The resulting force on the diaphragm is determined by the differential between the temperature of the suction gas and the boiling point of the refrigerant, which is the amount of superheat in the gas. If this temperature differential becomes greater (superheat increases), the resultant force on the diaphragm opens the valve and admits more refrigerant. The reverse is true if the superheat decreases,

and the valve partly closes, thus admitting less refrigerant. The spring keeps the valve closed until the resultant force on the diaphragm corresponds to the desired superheat. The adjustment of the spring will change the amount of superheat to be maintained in the suction gas.

The selection of the expansion valve is, of course, determined by the capacity of the valve. The capacity of a valve with a given orifice is determined by the refrigerant used, the differential of pressure across the valve and the amount the liquid is sub-cooled as it enters the valve. The expansion valves are usually rated at zero sub-cooling of the liquid, or 100 per cent liquid. Oftentimes special devices are used to properly distribute the refrigerant among the parallel paths of the evaporator. These distributing devices usually have considerable pressure drop. Where they are used, the pressure drop across the expansion valve is not the difference between suction and discharge pressures, as allowance must be made for the pressure drop across the distributing device. An equal-

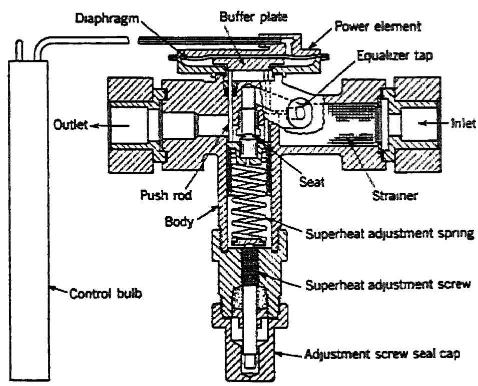


FIG. 7. TYPICAL THERMOSTATIC EXPANSION VALVE

izer connection from the evaporator suction line must be made to the underside of the diaphragm (see Fig. 7) whenever the valve outlet is not at the evaporator pressure so as to insure suction pressure at this point. When distributing devices are used, this equalizer connection is essential for proper operation of the valve. Another pressure drop allowance must be made for the liquid line, particularly when the liquid line has an appreciable vertical rise.

CONDENSERS

Condensers used for liquifying the refrigerant are of three general designs: (1) air, (2) water, and (3) evaporative (combination air and water).

Air Cooled

Air cooled condensers are seldom used for capacities above 3 tons of refrigeration, unless an adequate water supply is extremely difficult to

obtain, as, for instance, in railway air conditioning. Even on fractional tonnage installations, air is used as the condensing medium only where water is expensive or where simplicity of installation warrants the higher condensing pressure, and consequent higher power costs than would be obtained using water as the condensing medium.

The conventional air cooled condenser consists of an extended surface coil across which air is blown by a fan. The hot discharge gas enters the coil at the top and, as it is condensed, flows to a receiver located below the condenser. Air cooled condensers should always be located in a well ventilated space so that the heated air may escape and be replaced by cooled air.

The principal disadvantages of air cooled condensers are the power required to move the air and the reduction of capacity on hot days. This loss of capacity due to high condensing pressures on hot days requires that equipment of increased capacity be selected to meet the peak load. Thus at normal loads the equipment is oversized.

Water Cooled

Water cooled condensers are of the double pipe type, the shell and tube type, or the shell and coil type. Double pipe condensers are arranged so that water passes through the inner of two concentric pipes and refrigerant circulates through the annular space between the pipes. Where possible, there should be counter-flow of the refrigerant and the condensing water to obtain maximum temperature differences. This type is usually used only with small condensing units.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is therefore necessary to determine a balance so that the quantity of water insures economical compressor operation.

Because there is a decided tendency to conserve the water in city mains and because most large cities are restricting the use of water for air conditioning and refrigeration equipment, it is often necessary to install cooling towers or evaporative condensers. Cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains. Economies are possible when a cooling tower is used, which cannot be achieved by the use of condensing water from city mains. In certain localities, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of the outdoor temperature. With a cooling tower, the temperature of the condensing water may rise to 80 or 85 F under maximum conditions, but under less than maximum conditions the temperature of the water leaving the cooling tower drops

considerably. It has been established that in these localities during 50 per cent of the time, the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60 F, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is small as the only water required is that used to make up the loss by evaporation in the cooling tower itself. Refer to the section on cooling towers in Chapter 26.

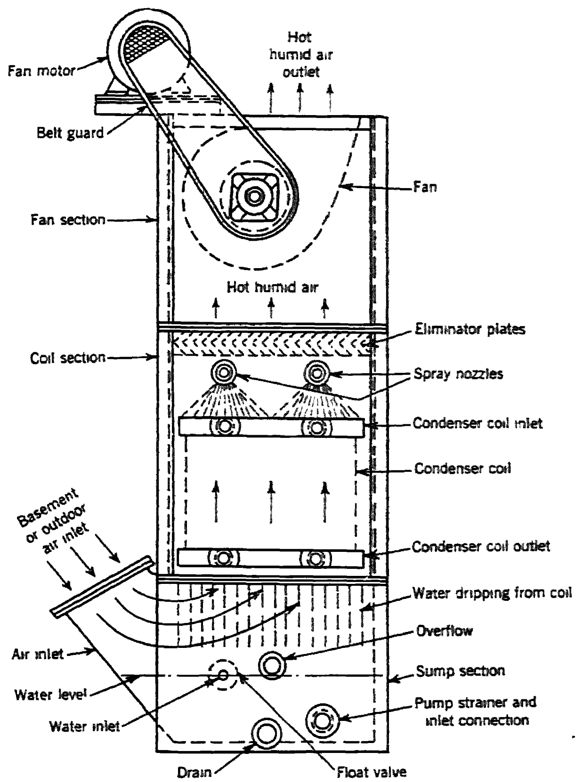


FIG. 8. SCHEMATIC VIEW OF AN EVAPORATIVE CONDENSER

Shell and coil condensers are in general use for medium sized condensing units, and consist of a coil of tubing mounted inside a shell. The cooling water passes through the coil.

Evaporative Condensers

Due to the high cost of city water for condenser purposes, and due to ordinances in some localities prohibiting the discharge of large quantities of such water into the sewage systems, there has been developed a condenser which uses a minimum amount of water on a finned surface, cooling

it to approximately the wet-bulb temperature of the surrounding atmosphere.

The end view of a typical evaporative condenser is shown in Fig. 8. The fan draws the air over a finned tube condenser which is kept wet by a water spray. The discharge refrigerant gas from the compressor enters the top of the condenser coil and the liquid refrigerant is drained from the bottom of the coil into a liquid receiver and then circulates through the remaining portion of the system in the usual way.

The water is circulated through the spray nozzles and the level is maintained in the sump by means of a float valve. The eliminator plates are placed in the path of the water-air mixture so as to remove the entrained water. The air leaving the unit is almost completely saturated, so that care must be taken in locating discharge ducts to prevent condensation.

Evaporative condensers are available in sizes up to 100 tons or more. These units use only a small portion of the water required for a water cooled condenser. The water is vaporized by the heat of the refrigerant so that each pound of water used extracts approximately 1000 Btu from the refrigerant, whereas, under standard rating conditions where the water temperature rise is 20 F, each pound of water extracts only 20 Btu from the refrigerant. Including the water lost by entrainment in the discharge air, by overflow and stand-by evaporation, the water used is about 3 to 5 per cent of the amount that would be required for a water cooled condenser.

The evaporative condenser requires more maintenance, occupies greater space (must be located where air is available), and has a higher first cost than the water cooled condenser, but where the use of water is restricted or expensive, the evaporative condenser has become widely accepted. Compared with a water cooled condenser and cooling tower, which combination uses about the same quantity of water, the evaporative condenser has the advantage of lower cost and smaller space requirements.

EVAPORATORS AND COOLERS

The types of coolers used in connection with air conditioning work fall into three general groups. The *first*, is the direct cooling of water; the *second*, direct cooling of air; and the *third*, cooling of brine for circulation in a closed system, which can cool either water or air. One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor the suction temperature or the

temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio. There are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used because of fire or other risks, especially in densely populated areas, the brine can be cooled in an isolated room or building and can then be circulated through the air conditioning equipment. This arrangement eliminates any possibility of direct contact between the air and refrigerant.

REFRIGERANT PIPE SIZES

The selection of proper pipe sizes and frictional pressure losses varies with the installation and the capacity of the system. Generally the suction piping should be selected so that the pressure loss is between 2 and 3 lb per square inch. The pressure drop in liquid lines should be maintained so as to permit no vaporization in the pipes with limiting pressure drops not to exceed 5 lb per square inch. Hot or discharge gas lines should be limited to approximately 4 lb per square inch pressure drop. All pressure drops mentioned are total system losses and include not only the piping losses, but also the pressure losses in the valves, fittings and coils.

Pressure drops for discharge or hot gas lines may be determined from Table 7. Pressure losses in liquid refrigerant lines of various sizes and capacities are given in Table 8. Pressure drops of suction refrigerant pipe lines at varying capacities and refrigerant temperatures are given in Table 9. Oil circulating with the refrigerant appreciably increases the pressure losses in both suction and discharge lines from that given in these tables. All tables are for 100 ft of pipe, including an average number of fittings, and for other lengths the losses are proportionate. Losses through

Table 7. Pressure Losses in Dichlorodifluoromethane Discharge or Hot Gas Lines[a]

Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[b]									
	Line Sizes, Inches									
	⅝	¾	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝
10,000	2.3	1.0	0.6							
15,000	4.9	2.0	1.0							
20,000	8.5	3.4	1.7	0.6						
25,000		5.3	2.6	0.9						
30,000		7.5	3.6	1.2	0.5					
40,000			6.4	2.1	0.7					
50,000			9.8	3.1	1.0	0.5				
60,000				4.4	1.3	0.7				
70,000				6.0	1.9	0.9				
80,000				8.0	2.5	1.1				
90,000				10.2	3.1	1.4				
100,000					3.8	1.7	0.5			
125,000					6.0	2.6	0.7			
150,000					8.5	3.8	1.0			
175,000					11.6	5.1	1.3			
200,000						6.7	1.7	0.6		
250,000						10.4	2.6	0.9		
300,000							3.7	1.2	0.5	
400,000							6.7	2.2	0.9	
500,000							10.5	3.5	1.5	0.7
600,000								5.0	2.1	1.0
800,000								9.0	3.8	1.8
1,000,000									5.8	2.9
1,250,000									9.5	4.4
1,500,000										6.4
2,000,000										11.3

[a]Soft annealed copper tubing up to and including ¾ in. outside diameter. Hard copper pipe ⅞ in. outside diameter and larger.

[b]Length of tubing includes the average number of fittings.

control and regulating valves must be added to the other pipe losses to determine the total drop. All copper pipe referred to in these tables are of type L wall thickness and are designated by outside diameter.

The effect of the sizes of refrigerant lines on the system may be studied by referring to the preceding discussion on Characteristics of Compression Systems. It will be noted that any lowering of the suction pressure at the compressor lowers the capacity. Therefore, excessive pressure drop through the suction piping should be avoided. On the other hand, the suction line must not be made too large when using refrigerants which are soluble in oil, because under such circumstances the velocity of the returning refrigerant may become too low to carry back the entrained oil. Pressure drop in the discharge line also lowers the capacity of the system

TABLE 8. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE LIQUID REFRIGERANT LINES

Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]			
	Pipe Sizes, Inches			
	7/8	1 1/8	1 3/8	1 5/8
100,000	0.6			
125,000	0.9			
150,000	1.3			
175,000	1.8			
200,000	2.3	0.6		
225,000	2.9	0.8		
250,000	3.6	1.0		
275,000	4.3	1.2		
300,000	5.1	1.4		
325,000	5.9	1.6		
350,000	6.9	1.8		
375,000	7.9	2.1		
400,000	9.0	2.3	0.8	
450,000		2.9	1.0	
500,000		3.5	1.3	
550,000		4.3	1.5	0.7
600,000		5.0	1.8	0.8
700,000		6.7	2.4	1.1
800,000		8.7	3.1	1.4
900,000			3.9	1.7
1,000,000			4.7	2.1
1,200,000			6.7	3.0
1,400,000			9.0	4.0
1,600,000				5.1
1,800,000				6.3
2,000,000				7.9
2,200,000				9.2

[a]Length of tubing includes the average number of fittings.

but not to the same extent as does the pressure drop in the suction line. The velocities of the refrigerant in either suction or discharge lines must not be excessive or noise will result. Velocities of 1000 to 2000 fpm are common in suction lines, and from 2000 to 3500 fpm are used in discharge lines. Velocities in the discharge lines as high as 5000 fpm can only be used where the fittings and bends are all stream-lined as noise will otherwise result.

The pressure drop in the liquid line affects the capacity of the expansion valve as the pressure drop across the valve is reduced by the amount of the pipe line drop. If the liquid line drop is sufficient to cause flashing (*i.e.* vaporizing) of some of the liquid refrigerant, a hissing noise in the lines and valves usually develops.

Table 9. Pressure Losses in Dichlorodifluoromethane Suction Refrigerant Lines

Copper Pipe Actual O.D. Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
¾	2,000	0.3	0.3	0.2	0.2	0.2	0.1	0.1
	4,000	1.3	1.0	0.8	0.7	0.6	0.5	0.4
	6,000	2.8	2.2	1.8	1.5	1.2	1.0	0.9
	8,000	4.8	3.8	3.1	2.6	2.1	1.8	1.5
	10,000	7.4	5.8	4.8	3.9	3.3	2.8	2.3
	12,000	10.5	8.4	6.8	5.6	4.7	4.0	3.3
	14,000	14.0	11.0	9.1	7.6	6.4	5.4	4.5
	16,000		14.5	12.0	9.8	8.3	7.0	5.8
	18,000			15.0	12.3	10.4	8.7	7.2
	20,000				15.0	12.7	10.7	8.9
1⅛	7,000	0.4	0.3	0.3	0.2	0.2	0.2	0.1
	10,000	1.0	0.7	0.5	0.5	0.4	0.3	0.3
	15,000	1.9	1.5	1.2	1.0	0.8	0.7	0.6
	20,000	3.3	2.6	2.1	1.7	1.4	1.2	1.0
	25,000	5.0	4.0	3.2	2.7	2.2	1.9	1.6
	35,000	9.7	7.7	6.2	5.1	4.3	3.6	3.0
	45,000	15.8	12.6	10.0	8.4	7.0	5.9	4.9
	60,000				14.8	12.2	10.2	8.6
	70,000						14.0	11.7
1⅜	10,000	0.3	0.2	0.2	0.2	0.1	0.1	0.1
	15,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	20,000	1.2	0.9	0.7	0.6	0.5	0.4	0.4
	30,000	2.6	2.1	1.6	1.3	1.1	0.9	0.8
	40,000	4.6	3.6	2.8	2.3	1.9	1.6	1.4
	50,000	7.0	5.5	4.4	3.5	2.9	2.5	2.1
	60,000	10.0	7.8	6.2	5.0	4.2	3.5	3.0
	80,000		14.0	11.0	8.7	7.3	6.2	5.2
	100,000				13.5	11.3	9.5	8.2
1⅝	30,000	1.6	1.3	1.0	0.8	0.7	0.6	0.5
	40,000	2.7	2.1	1.7	1.4	1.1	0.9	0.8
	50,000	4.2	3.2	2.5	2.1	1.7	1.4	1.2
	60,000	6.1	4.5	3.6	2.9	2.4	2.0	1.7
	70,000	8.7	6.3	4.8	3.8	3.1	2.6	2.2
	80,000		8.4	6.3	4.9	4.0	3.3	2.8
	90,000			8.0	6.2	4.9	4.1	3.5
	100,000			10.0	7.6	6.1	5.0	4.2
	120,000					8.6	7.0	5.9
	140,000						9.5	7.9

[a]Length of tubing includes the average number of fittings.

Table 9. Pressure Losses in Dichlorodifluoromethane Suction Refrigerant Lines (Continued)

Copper Pipe Actual O.D. Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a] — Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
2⅛	50,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	100,000	2.6	1.8	1.4	1.1	0.9	0.8	0.7
	150,000	5.6	3.9	3.0	2.4	2.0	1.6	1.4
	200,000	9.8	6.7	5.2	4.1	3.4	2.8	2.4
	250,000	14.8	10.3	8.0	6.3	5.1	4.2	3.6
	300,000		14.5	11.3	9.0	7.2	6.0	5.0
	350,000		19.5	15.3	12.0	9.7	7.8	6.7
	400,000			19.6	15.3	12.5	10.0	8.5
2⅝	50,000	0.2	0.2	0.1	0.1	0.1	0.1	0.1
	100,000	0.7	0.6	0.5	0.4	0.3	0.2	0.2
	150,000	1.6	1.2	1.0	0.8	0.6	0.5	0.4
	200,000	2.8	2.1	1.7	1.4	1.1	0.9	0.7
	250,000	4.3	3.4	2.6	2.1	1.7	1.3	1.1
	300,000	6.1	4.5	3.7	3.0	2.4	1.9	1.5
	350,000	8.2	6.0	5.0	4.0	3.2	2.5	2.0
	400,000		7.8	6.5	5.1	4.2	3.3	2.7
	450,000			7.7	6.4	5.3	4.0	3.5
	500,000				7.8	6.4	5.0	4.2
	550,000					7.7	6.2	5.1
	600,000						7.4	6.2
3⅛	200,000	1.2	1.0	0.8	0.6	0.5	0.4	0.4
	300,000	2.6	2.0	1.6	1.3	1.0	0.8	0.7
	400,000	4.5	3.4	2.6	2.1	1.7	1.4	1.3
	500,000	7.3	5.4	4.1	3.3	2.7	2.2	1.9
	600,000		8.1	6.0	4.7	3.8	3.1	2.7
	700,000			8.4	6.5	5.2	4.2	3.5
	800,000				8.6	6.8	5.5	4.6
	900,000					8.7	7.0	5.9
	1,000,000						8.9	7.3
3⅝	300,000	1.2	0.9	0.7	0.6	0.5	0.4	0.3
	400,000	2.0	1.6	1.3	1.0	0.8	0.7	0.6
	500,000	3.2	2.5	1.9	1.6	1.3	1.0	0.9
	600,000	4.6	3.6	2.8	2.2	1.8	1.5	1.3
	700,000	6.4	4.9	3.8	3.0	2.5	2.0	1.7
	800,000	8.7	6.4	4.9	3.9	3.2	2.5	2.2
	900,000		8.2	6.2	4.9	3.9	3.2	2.7
	1,000,000			7.7	6.1	4.9	4.0	3.3
	1,100,000			9.4	7.3	5.8	4.8	4.0
	1,200,000				8.7	6.9	5.6	4.8
	1,300,000					8.0	6.6	5.6
	1,400,000					9.3	7.6	6.4

[a]Length of tubing includes the average number of fittings.

Table 9. Pressure Losses in Dichlorodifluoromethane Suction Refrigerant Lines (Concluded)

Copper Pipe Actual O.D Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
4 1/8	400,000	1.0	0.8	0.6	0.4	0.4	0.3	0.3
	600,000	2.4	1.8	1.4	1.1	0.9	0.7	0.6
	800,000	4.1	3.1	2.4	2.0	1.6	1.3	1.1
	1,000,000	6.6	4.8	3.7	3.0	2.5	2.0	1.6
	1,200,000	10.0	7.1	5.4	4.4	3.5	2.9	2.4
	1,400,000		10.0	7.5	5.9	4.8	3.9	3.3
	1,600,000			10.0	7.7	6.2	5.1	4.2
	1,800,000				10.0	7.9	6.4	5.3
	2,000,000					9.7	7.9	6.6
	2,200,000						9.5	7.9

[a]Length of tubing includes the average number of fittings.

ICE SYSTEMS

Cold water systems using ice as the cooling agent have been installed in many theaters, restaurants, funeral homes, churches and other places where short hours of operation and high peaks of cooling demand make this type of system desirable. A comparatively small quantity of ice in the water cooling tank of such a system can release refrigeration at a relatively rapid rate. For instance, neighborhood theaters having a peak demand of 1,200,000 Btu per hour (100 tons refrigeration) have found 8 ton capacity ice bunkers satisfactory.

In operation, the water in the air conditioning system is circulated over ice placed in an insulated box and is cooled to the 38 or 40 F range or higher if desired. This cold water is pumped from the ice bunker to air cooling coils or spray type air washers. The blowers, coils, air washer or air handling sections are the same as those parts in any system employing cold water as a refrigerant.

The ice water cooler or ice bunker is usually built on the job in a location where it can easily be iced. It can be built of any desired material such as concrete, steel, or wood with a 4 in. thickness of standard insulation to save the ice from one period of use to the next. The basic requirement is that the tank be durable and water tight. A typical bunker with connections to a coil type air conditioning system is shown in Fig. 9. About 60 cu ft of gross bunker volume is allowed per ton of ice capacity.

The shape of the bunker usually conforms to the available space. The one illustrated has overhead sprays, but if head-room is lacking the ice is placed on the floor of the bunker with the water returned around the lower part of the blocks from a perforated distribution pipe run along one side of the bunker. To secure good circulation the supply water is extracted from a similar perforated pipe on the opposite side of the bunker.

The temperature of the water is controlled at a predetermined point by a thermostat in the supply line. If the temperature drops too low, a part of the return water is by-passed directly to the sump and is not cooled over the ice. In the larger systems it is customary to install an overflow control which, as the ice melts, discards the excess water through an economizer coil. The surface of the economizer is large in relation to the flow so that the water is warmed to 60 F or more as it is discharged from the system.

EQUIPMENT SELECTION

The selection of proper refrigeration equipment for any air conditioning job is of utmost importance for satisfactory results. The most important factors in the selection of the equipment are:

1. Loads (as determined by the conditions of the space to be cooled).
2. Economics (both initial and operating costs).

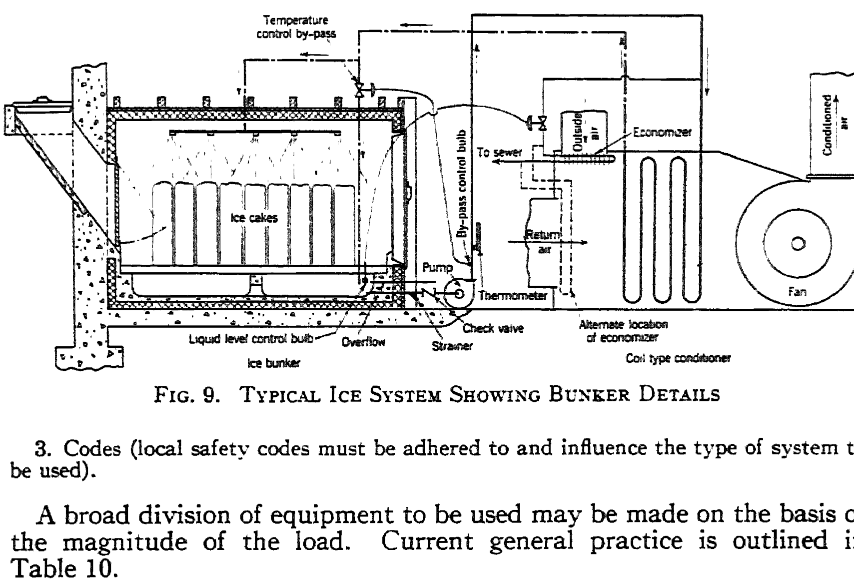


FIG. 9. TYPICAL ICE SYSTEM SHOWING BUNKER DETAILS

3. Codes (local safety codes must be adhered to and influence the type of system to be used).

A broad division of equipment to be used may be made on the basis of the magnitude of the load. Current general practice is outlined in Table 10.

Unit or *packaged* systems, consisting of a reciprocating compressor, condenser, evaporator and fans, are generally used in the smaller sized jobs where electric power is available, as they are manufactured complete ready to install and are the most economical.

The reciprocating compressor in the built up central system covers the widest range of application since it is applicable to either the direct expansion or indirect systems and can be driven by steam or gas engines, or by electric motors. The quantity of condensing cooling medium required is also less than for any other system with the exception of the centrifugal compressor, which uses the same amount.

Centrifugal compressors are used for large installations, and usually where the indirect system is required. The driving mechanism can be

TABLE 10. BASIS OF EQUIPMENT SELECTION

Capacity Tons	Majority Used	Some Used	Few Used
0 to 5	Unit systems in conditioned space.	Unit central systems using duct distribution.	Built up central systems.
5 to 25	Built up central systems using reciprocating compressors.	Unit central systems using duct distribution.	Unit systems in conditioned space. Built up systems using absorption and adsorption systems.
25 to 50	Built up central systems using reciprocating compressors.	Built up central systems using centrifugal compressors.	Central systems using adsorption systems.
50 to 400	Built up central systems using reciprocating compressors.	Built up central systems using steam jet and centrifugal compressors.	
400 and Over	Built up central systems using centrifugal compressors.	Built up central systems using steam jet.	

steam turbine or electric motor. The steam jet system is used where steam is available and cooling water can be had in large quantities.

It will be noted by referring to Fig. 4 that all systems using compressors have a common characteristic and that is, that the capacity varies with the evaporating temperature. Not only can the equipment be selected to produce a given result but the performance can be predicted under varying load conditions by the simple expedient of using the variable of evaporating temperature as the abscissa and the load or capacity as the ordinate in a series of curves.

Manufacturers of compressors and cooling coils furnish performance data for apparatus that can be plotted in the form of curves similar to those shown in Fig. 10. The performance of a compressor is plotted as a series of curves, each curve being drawn for a given condensing pressure. The performance of a direct expansion coil at two different air velocities is plotted on the same graph. The operating point will be, of course, where the two curves cross.

TABLE 11. TYPICAL OPERATING CONDITIONS FOR TWO TYPES OF LOAD

Type of Enclosure	Load, Btu per Hour			Ratio Sensible to Total	Air Entering Coil		Operating Balance Point		
	Sensible	Latent	Total		Deg F	Per Cent R.H.	Evaporator Temp Deg F	Condenser Pressure Lb per Sq In.	Per Cent Sensible Heat
Restaurant	103,000	45,000	148,000	0.695	82	45	34.4	123	69.9
Office	121,000	27,000	148,000	0.820	82	45	42.2	100	82.1

Data given in Table 11 illustrate two types of conditioned enclosures having the same total load of 148,000 Btu per hour, but with two different ratios of sensible to total heat. In the case of the office with a ratio of 82 per cent sensible to total heat, the operating point *A* in Fig. 10 is found to be 42.2 F evaporating temperature with a face velocity of 500 fpm. In the case of the restaurant, with a ratio of 69.5 per cent sensible to total heat, the air velocity is lowered to 300 fpm and the evaporating temperature is lowered to 34.4 F as shown in point *B* of Fig. 10. In order to obtain the same capacity, a larger condensing unit is used. This illustration assumes zero pressure drop through the suction line. The pressure drop can be taken into account by shifting the compressor performance curves by the amount of pressure drop expressed in degrees Fahrenheit.

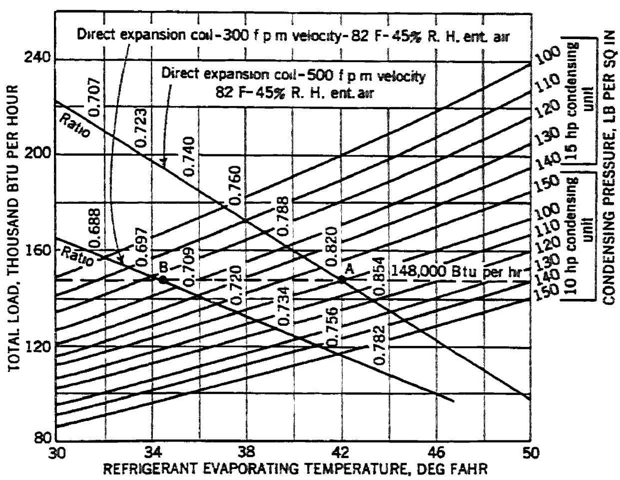


FIG. 10. COMPRESSOR AND COIL PERFORMANCE

THE REVERSE CYCLE

In heating by the reverse refrigeration cycle energy is absorbed in an evaporator from some available source of heat, pumped to a higher temperature and delivered to a condenser. The heat from the condenser is used for heating purposes. The compressor acts as a heat pump whose fundamental function is to raise the potential of the heat. The theoretical ratio of the heat delivered to the work of compression is given in Equation 1.

$$\frac{T_2}{T_2 - T_1} \tag{1}$$

where

T_1 = absolute temperature of evaporator.
T_2 = absolute temperature of condenser.

Thus, with a small spread of temperature between the evaporator and the condenser, 6 or 8 times as much heat may be obtained theoretically,

and 3 to 5 times practically, as the work introduced[1]. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

1. Well water is the most desirable since its temperature is higher than other sources even in the winter, and thus a large amount of heat may be removed in relation to the weight of water handled.

2. Air may be used but its specific heat is low and its temperature uncertain. When the most heat is needed, the temperature of the air is lowest, thus resulting in the least favorable temperature combination.

3. It has been proposed to obtain heat by freezing water but this is still in the experimental stage.

Some of the other factors which act as limitations are: the large temperature spread when using air as a source of heat and when attempting to cool with even moderately low outside temperatures, the frequent disparity between the size of the cooling load and heating load requiring

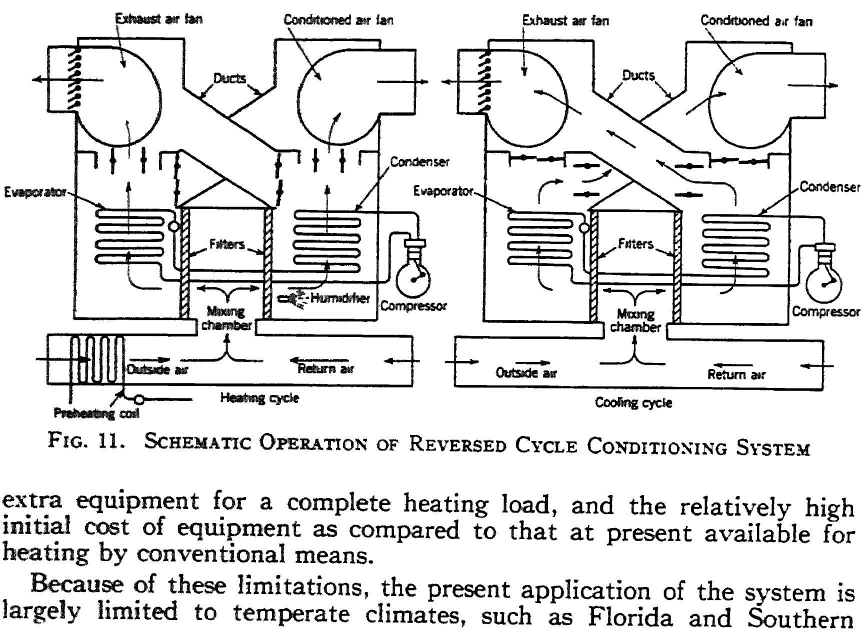


FIG. 11. SCHEMATIC OPERATION OF REVERSED CYCLE CONDITIONING SYSTEM

extra equipment for a complete heating load, and the relatively high initial cost of equipment as compared to that at present available for heating by conventional means.

Because of these limitations, the present application of the system is largely limited to temperate climates, such as Florida and Southern

[1]Cooling Homes, A Field for Refrigeration, by A. R. Stevenson, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926.

The Heat Pump, An Economical Method of Producing Low-grade Heat from Electricity, by T. G. N. Haldane (*Electric Review*, Vol. 105, p. 1161-1162, December 27, 1929, and *I. E. E. Journal*, Vol. 68, p. 666-675, June, 1930).

Edison Building Heated and Cooled by Electricity, by H. L. Doolittle (*Power*, Vol. 74, p. 384, September 8, 1931).

House Heating by Pump with 5 to 1 Pick-up Ratio, by Gilbert Wilkes and R. E. Marbury (*Electrical World*, Vol. 100, p. 828, December 17, 1932).

An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 307).

Using the Reversed Cycle Refrigerating Principle for a Self-Contained Heating and Cooling Unit, by Henry L. Galson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1935, p. 497).

Heating by Reversed Refrigeration, by A. J. Lawless (*Heating, Piping and Air Conditioning*, August p. 473, September, p. 519, 1940).

California, or to heating only for intermediate seasons, or to other localities which have peculiar advantages as, for instance, the ready availability of well water. In these locations it is frequently possible to do all of the heating necessary with the refrigeration equipment so that the extra cost is only that of reversing the functions of the condenser and evaporator.

There are a number of reversed systems now in operation, particularly among utility companies, using well water as the source of heat. These systems range in size up to 320 hp. In the case of the largest system in operation at present, the cost of the electrical energy would have to be approximately 0.7 cents per kilowatthour in order to compete with oil at 6 cents per gallon.

A typical arrangement of a reversed cycle conditioning system where air is used as a source of heat is shown in Fig. 11. If the air seldom drops below freezing, heat is often required in the morning and cooling during the afternoon in order to maintain comfortable conditions in such a system. The arrangement as shown lends itself to automatically changing over as required.

Example 1. Electrically driven dichlorodifluoromethane condensing units are to be used in an air conditioning system, requiring 20 tons refrigerating capacity for conditions of maximum load. An overall analysis of the seasons operating conditions shows an average load factor of 62.5 per cent, and allowing for variable time intervals of operation of refrigeration units installed, three-quarters of the operating season, or 750 hr, would require operation of the equipment at one-half load, and one-quarter of the operating season or 250 hr full load capacity of the refrigeration equipment would be required.

The increased first cost of 2-10 hp, 10 ton condensing units over 1-20 hp, 20 ton condensing unit is, $830.00 installed price, to the customer.

The increased first cost of a 2-speed compressor motor of 20 hp size over a constant speed of 20 hp size motor including increased starter cost is $210.00. The efficiency of the 2-speed motor above is 83 per cent at full load speed, and 79 per cent for full load at ½ speed. At ½ speed, full load is ½ total bhp of full load speed.

Discuss the consideration involved in making a decision as to whether a single unit with a 20 hp motor of the 2 speed type would be used in preference to 2-10 hp constant speed units.

Solution. The cost of 2-10 hp 10 ton units in excess of 1-20 hp, 20 ton unit with 2-speed motor, is $830.00 − $210.00 or $620.00, increased first cost. At 15 per cent fixed charges, this represents an increased annual cost of $93.00 for 2 compressors over one compressor. The advantage of 2 compressors instead of one compressor on an installation of this type, is in the breakdown service provided in the event one compressor is shut down for repairs the system could be operated at one-half capacity utilizing the duplicate machine. The motor efficiency of the constant speed unit would be higher at full load than would be the efficiency of the 2-speed motor at low speed. Offsetting this latter advantage however, is the fact that the condenser on the condensing unit would provide a lower refrigerant condensing temperature for ½ load operation with the same final condensing water temperature than would be the case with duplicate units each furnished with its own compressor and condenser. Operation at a lower condensing temperature would provide for a power saving compensating for the lower efficiency of the 2-speed motor when operated at slow speeds. It is, in a case of this kind, purely a question as to whether or not the purchaser would deem an investment of $620.00 more and an increased fixed charge of $93.00 a year, advisable to get breakdown service through the installation of duplicate units. In most cases, this increased first cost would not be warranted because of the fact that satisfactory indoor conditions could not be obtained at full load if only one-half the refrigeration capacity were available.

Example 2. For condensing purposes, an air conditioning system uses city water which has an average 70 F supply temperature. The following table lists the number

of hours per year during which definite wet-bulb temperatures and corresponding refrigeration rates pertain.

Wet-Bulb Temperature F	No. of Hours per Year	Refrigeration Required Tons
80	6	284
79 – 75	100	233
74 – 70	277	183
69 – 65	330	157
64 – 60	277	144
59 – 55	158	79
54 – 50	52	37
	Total 1200 hours	

If the power requirements of a dichlorodifluoromethane refrigeration system are in accordance with the following data on partial load operation, determine the seasonal power cost at 2 cents per kwhr:

Tons of Refrigeration	284	233	183	157	144	79	37
Kw per ton	0.89	0.89	0.87	0.86	0.86	0.93	0.97

Solution. Seasonal power cost:

Wet Bulb Temperature F	Ton-Hours	Kwhr
80	6 × 284 = 1,704	1,704 × 0.89 = 1,517
79 – 75	100 × 233 = 23,300	23,300 × 0.89 = 20,750
74 – 70	277 × 183 = 50,700	50,700 × 0.87 = 44,100
69 – 65	330 × 157 = 51,800	51,800 × 0.86 = 44,500
64 – 60	277 × 144 = 39,900	39,900 × 0.86 = 34,300
59 – 55	158 × 79 = 12,500	12,500 × 0.93 = 11,600
54 – 50	52 × 37 = 1,920	1,920 × 0.97 = 1,860
Totals	181,824 ton-hours	158,627 kwhr

The 158,627 kwhr at 2 cents per kwhr will cost $3,173.

The average consumption will be $\frac{158{,}627 \text{ kwhr}}{181{,}824 \text{ ton-hours}} = 0.873$ kw per ton.

Example 3. Using the data from Example 2, if city water costs 20 cents per thousand gallons, and if 1.25 gal are used per minute per ton, estimate the annual water cost.

Solution.

$60 \times 1.25 = 75$ gal per ton-hour.

181,824 ton-hours $\times$ 75 = 13,620,000 gal per year.

$$\frac{13{,}620{,}000 \times \$0.20}{1000} = \$2{,}724 \text{ the yearly cooling water cost.}$$

Chapter 25

HEAT TRANSFER SURFACE COILS

Coil Applications, Construction and Arrangement, Steam Coils, Water Coils, Direct-Expansion Coils, Flow Arrangements, Applications, Calculation of Heat Transfer, Air Flow Resistance, Coil Performance, Selection

THE coils described in this chapter are used in air conditioning systems for heating or cooling an air stream under forced convection. The surface coil equipment may be made up of a number of banks assembled in the field, or the entire assembly may be factory constructed. The applications of each type of coil are limited to the field within which it is rated. Other limitations are imposed by code regulations, by proper choice of materials for the refrigerants used and the condition of the air handled, or by an economic analysis of the possible alternates on each installation.

For heating service, these coils are used as preheaters, reheaters or booster heaters, (see Chapters 20 and 21). The function of the coils is air heating only, but the apparatus assembly may include means for humidification and air cleaning. Steam or hot water are the usual heating media, although others are used in special cases, such as reheating by means of discharge gas from a refrigerating system.

Coils are used for air cooling with or without accompanying dehumidification. Examples of cooling applications without dehumidification are precooling coils using well water or other relatively high temperature water to reduce the load on the refrigerating machinery, or water cooled coils to remove sensible heat in connection with chemical moisture-absorption apparatus. By proper coil selection it is possible to handle both sensible cooling and dehumidification together as further explained later. The apparatus assembly usually includes an air cleaning means to protect the coil from accumulation of dirt and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are the usual functions, there are cases of cooling coils purposely wetted as an aid to air cleaning and odor absorption.

The usual cooling media used in surface coils are cold water and volatile refrigerants such as dichlorodifluoromethane and methyl chloride, but others are used in special cases. Brines are seldom required for the range of applications covered by this chapter, although there are cases where low entering air temperatures with large latent heat loads require a refrigerant temperature so low that water becomes impractical. Some-

times, also, brine from an industrial system already installed is the only convenient source of refrigeration.

For combined cooling and dehumidifying, surface coils present an alternate to spray dehumidifiers. For many applications it is possible, by proper selection of apparatus, choice of air velocities, refrigerant temperatures, etc., to perform the same duty with either. In a few cases both sprays and coils are used. The coils may then be installed within the spray chamber, either in series with the sprays or below them. In making the selection between spray and surface dehumidifiers, certain advantages of each should be considered. The fact that a spray dehumidifier is usually designed to deliver nearly saturated air tends to simplify the control problem. In this case the dry-bulb temperature is also the dew-point, and hence a dew-point control can be arranged by using a simple duct thermostat. Spray dehumidifiers have the advantage over unwetted coils of a certain degree of air cleaning and odor absorption. On the other hand, coils make possible a closed and balanced cooling water circuit, obviating the unbalanced pumping head, the complication of water level control, and danger from possible floods incidental to multiple-spray dehumidifiers, especially if located on different levels. The use of coils often makes it possible for the same surface to serve for summer cooling and winter heating by circulating cold water in the one season and hot water in the other, with consequent saving in apparatus and piping. Surface-coil dehumidifiers seldom deliver saturated air, and wet-bulb depression of 0.5 to 4 F (or more) is usual. Another advantage is that where the surface coil system can be used with direct expansion of refrigerant, it is comparatively low in initial and operating costs. Of course the safety of the occupant must be kept in mind in comfort conditioning applications. Some localities have refrigeration codes which restrict the use of direct-expansion coils in the air stream, and hence local codes should be consulted by the engineer before a system employing direct expansion methods is designed. The choice between spray dehumidifiers and coils depends upon the necessities and the economic aspects of each case and no general rule can be given. There are many installations in which either can be used.

COIL CONSTRUCTION AND ARRANGEMENT

Coils are basically of two types, those consisting of bare tubes or pipe and those of *extended* surface construction. The former are little used for the applications covered by this chapter, but are often employed where conditions cause frost accumulation, and for cooling surface within spray dehumidifiers.

The heat transmission from air passing over a tube to a refrigerant flowing within it is impeded by three resistances. The same is true when the air is being heated by steam or hot water in the tube. The first resistance is from the air to the surface of the tube, usually called the outside surface resistance or air-film resistance. Second is the resistance to the flow of heat by conduction through the metal itself. Finally there is another surface or film resistance to the flow of heat between the inside surface of the metal and the fluid in the tube. For the applications under consideration both the resistance of the metal wall to heat conduction, and the inside surface or film resistance are usually low as compared with the air-side surface resistance. This is especially the case where sensible

heating or cooling only is accomplished. Where dehumidification accompanies sensible cooling, or where the external surface of the tube is sprayed with large quantities of water, the resistance to heat flow between the tube and the air flowing over it is much decreased. In the case of the water spray, the surface resistance depends on the amount and the method of application of the water. Economy in space, weight and cost make it advantageous to decrease the external surface resistance, where it is proportionately large, to approach that of the tube wall, and that from tube to refrigerant. This is accomplished by increasing the external surface by means of fins. With water spray the external resistance is already low, and the fins are less useful for increasing the overall heat transfer. Sometimes water spray is applied to the same type surface as would have been used without it. The overall heat transfer is not necessarily increased much by such an arrangement, but the water spray may serve other purposes than to increase the flow of heat, such as air and coil cleaning.

In fin or *extended* surface coils the external surface of the tubes is known as *primary* and the fin surface is called *secondary*. The primary

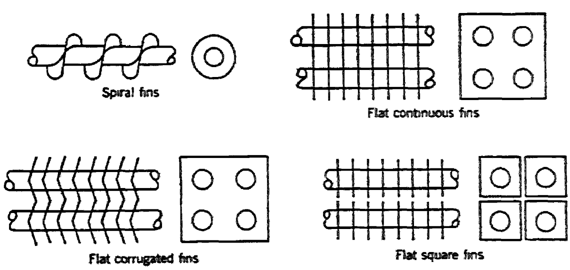


FIG. 1. TYPES OF FIN COIL ARRANGEMENT

surface consists generally of round tubes or pipes. In some cases these are staggered and in others in line with respect to the air flow. The staggered arrangement gives a somewhat higher heat transfer value but also a higher resistance to air flow and in some cases makes the header and return bend arrangement more complicated. A number of types of fin arrangement are used, the most common of which are spiral, flat and flat-crinkled or corrugated, all as shown in Fig. 1. While the spiral fin surrounds each tube individually in all cases, the flat types may be continuous (including several rows of tubes), or they may be round or square, with individual fins for each tube. All of these, as well as other less common types, are in use, the selection for a particular installation being based on economic considerations, space requirements and resistances of individual designs of coils. A most important factor in the performance of extended surface coils is the bond between the fin and the tube. An intimate contact is assured in a number of ways. The assembled coil may be coated with tin, zinc, etc., after fabrication. The spiral type fin may be knurled into a shallow groove on the exterior of the tube. The tube may be expanded after the fins are assembled, or the tube hole flanges of a flat or corrugated fin may be made to override those in the preceding

fin and so compress them upon the tube. There are also types of construction where the fin is formed out of the material of the tube itself. In any case the successful performance of a fin surface depends upon the bond between fin and tube being secure and remaining so in service.

For heating coils the materials most generally used are copper, steel and aluminum. Sometimes aluminum or brass fins are used on copper tubes. Steel is uncommon except in special cases. Some types of heating coils are made of cast-iron. There are sufficient practical installations of each of these to demonstrate that they can all give good service. However for equal performances brass and aluminum fins must be of greater thickness than copper fins on account of their lower coefficients of conduction. The copper coils are frequently tin-dipped and steel coils galvanized to protect them from corrosion and to assure a bond between fin and tube.

Cooling coils for water or for volatile refrigerants are most frequently of copper, both fin and tube. Aluminum fins on copper tubes are also used. For brines such as sodium or calcium chloride and for ammonia, steel fins and tubes are common.

Although there are many variations for special cases, tube and fin sizes and spacings for air conditioning coils, both heating and cooling, fall within fairly narrow limits. The tubes are usually 3/8, 1/2, 5/8, or 3/4 in. OD, and the fins spaced from 4 to 8 per inch, 6 per inch being a common design. The tube spacing generally varies from about 1 1/8 to 2 in. on centers. Small tube size and close fin spacing give large capacity with small space demand, but the resistance, both over the surface and through the tubes, is higher than with larger tubes and more widely spaced fins. Moreover, too close a fin spacing may result in trouble from dirt accumulation, especially on dehumidifying coils, and may also cause trouble from water *hold-up* between the fins, particularly with air flow vertically upward. This condition increases the air resistance and decreases the capacity of the coil. Water hold-up sometimes causes flooding trouble in vertical air flow units by accumulating too much water for the drain to handle all at once when the fan is stopped.

Steam Coils

For proper performance of steam heating coils, condensate and air must be continually eliminated and the steam must be evenly distributed to the individual tubes. This distribution is usually accomplished by individual orifices in the tubes, by distributing plates and orifice in the steam header, or by perforated internal steam-distributing pipes extending into the individual tubes. The latter arrangement has the advantage of distributing the steam throughout the length of each tube, and is conducive to uniform delivered air temperatures. The tendency for freezing of condensate at the bottom of the coil with cold entering air and light heating loads is also minimized. This is especially valuable for outside air preheaters. Methods of air and condensate elimination are discussed in detail in Chapters 13, 14 and 21.

Water Coils

The performance of water coils, for heating or cooling, depends on the elimination of air from the system and proper distribution of water. Air elimination is taken care of in the system piping as described in Chapter

15. To assure a pressure drop sufficient for adequate distribution but at the same time to provide against excessive pumping head where large water quantities are handled, water coils are provided with various water circuit arrangements. For instance, a typical coil 18 tubes high and 6 tubes deep in the direction of air flow can be arranged for 6, 9, 18 or 36 parallel water circuits as conditions may require. Orifices in individual tubes are occasionally employed but are usually unnecessary as the resistance of individual water circuits is generally sufficient to effect a satisfactory distribution. In cases such as well water precooling coils, where there may be considerable sand and other foreign matter in the water, provision for cleaning of individual tubes is of advantage. It is important to arrange water coils for drainage if located where they will be

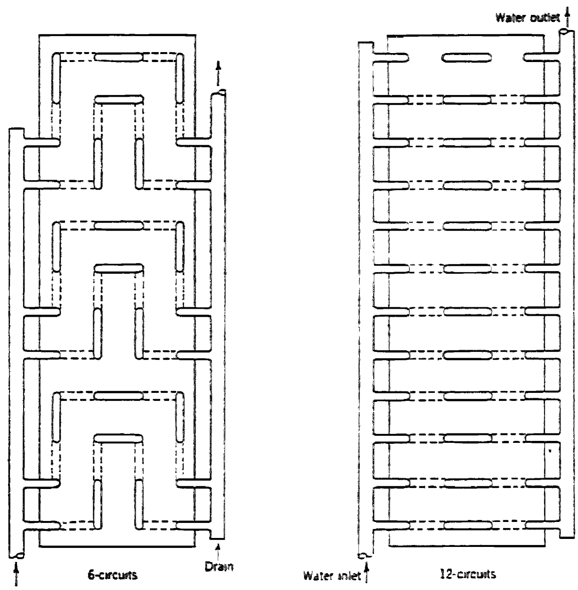


FIG. 2. VARIOUS WATER CIRCUIT ARRANGEMENTS

exposed to freezing. For this reason the circuits should be so laid out that there are no pockets to hold water. Fig. 2 shows such construction. The drains may be provided in the water piping although they are often arranged in the coil headers.

Direct-Expansion Coils

Coils for volatile refrigerants present more complex problems of fluid distribution than do water, brine or steam. It is desirable that the coil be effectively and uniformly cooled throughout, and necessary that the compressor be protected from entrained, unevaporated refrigerant. There are two types; namely, flooded systems, and thermal expansion valve systems, as shown in Figs. 3 and 4. With flooded control the coils are supplied with liquid by the same type of circulation that exists in a water tube boiler, while the level in the surge drum is maintained by the action

of the float regulator, or by properly charging the plant in the case of the high pressure float drainer. The thermal expansion valve system depends upon the thermal valve automatically feeding just as much liquid to the coils as is required to maintain the superheat at the coil suction outlet within predetermined limits which vary from about 6 to 10 F. The

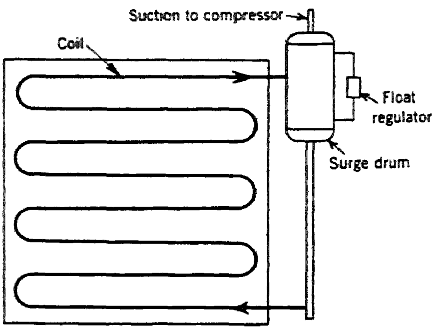


FIG. 3. DIRECT-EXPANSION COIL WITH FLOODED SYSTEM

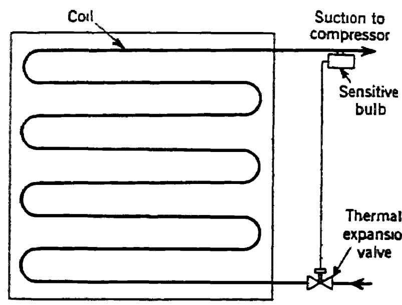


FIG. 4. DIRECT-EXPANSION COIL WITH THERMAL VALVE SYSTEM

thermal valve arrangement is in common use for the type of coils covered by this chapter, while the flooded system is comparatively rare.

With the flooded system the refrigerant distribution through the tubes depends on properly selecting the length of the feeds and the head of liquid imposed upon the liquid inlets. No auxiliary distributing devices

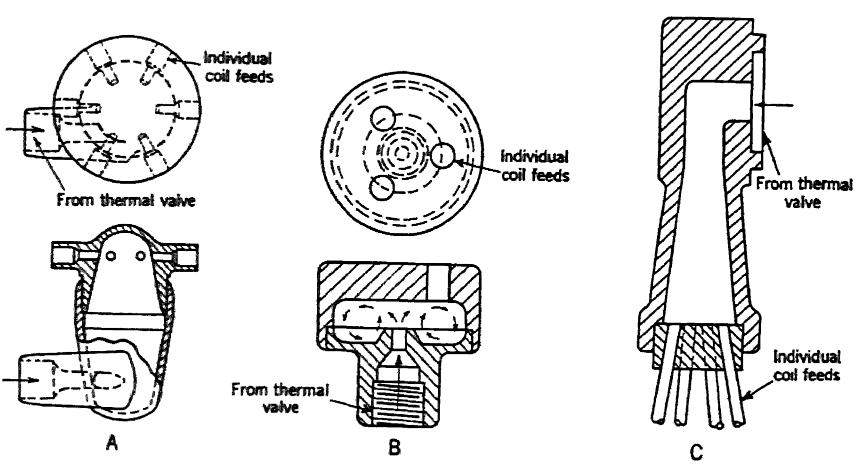


FIG. 5. TYPES OF REFRIGERANT FEED DISTRIBUTING HEADS

are required. With the thermal valve system there are two factors to consider. There must be, generally, more than one refrigerant feed through the coil per thermal valve to keep the pressure drop through the refrigerant circuit within practical limits and to reduce the corresponding penalty in increased evaporating temperature. At the same time the

coil must be so arranged that the required suction superheat can be attained with a minimum sacrifice in the performance of the coil as a whole. It is general practice to attain this superheat within the coil itself and not by the use of external heat exchangers or other auxiliary devices.

With thermal expansion valves it is advantageous to keep the pressure drop through the refrigerant feeds as low as possible. The feeds are laid out to expose each to the same mean temperature difference so that it handles the same refrigerating load. A distributing means is imposed between valve and coil liquid inlets to divide the refrigerant equally among the feeds. Such a distributor shall be effective for distributing both liquid and vapor, since the entering refrigerant is a mixture of the

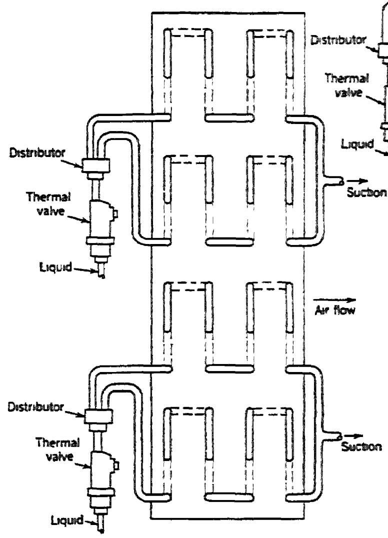


FIG. 6. ARRANGEMENT FOR FACE CONTROL

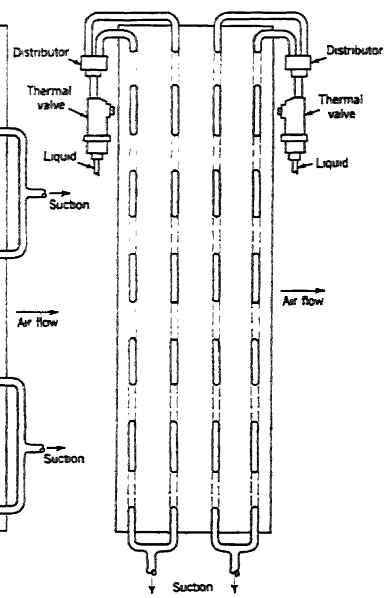


FIG. 7. ARRANGEMENT FOR DEPTH CONTROL

two. Fig. 5 shows three typical types of distributors. In distributor *A* the liquid and gas mixture from the thermal valve is led tangentially into a chamber. The coil feed connections extend outward radially at the top of this chamber. In distributor *B* the refrigerant is discharged at a high velocity through a central jet against the end plate. forming a uniform mixture of gas and liquid within the distributor, from which individual connections are led as shown. In type *C* the refrigerant enters at high velocity from the thermal valve and is discharged against the end plug in which the individual liquid feeds are closely arranged. These distributors can be used in either vertical or horizontal position. Although there are other forms of distributors the above are typical examples. The individual liquid connections from the distributor to the coil inlet are commonly made of small diameter tubing and are all of the same length

and diameter in order to impose the same friction between the distributor and the coil. Since the thermal valves act in response to the superheat at the coil outlet, this superheat should be produced with the least possible sacrifice of active evaporating surface. Sometimes a single thermal valve is used per coil. In other cases multiple valves are used, with the coil divided across the air flow or parallel to the air flow as shown in Fig. 6. The arrangement of Fig. 7 should be avoided since it offers the disadvantage of unequal load on the two parallel circuits.

Flow Arrangement

The relative direction of flow of the air outside the tubes and the medium within them influences the performance of the surface. There are three types of relative flow in common use. Fig. 8A shows parallel-flow in which the air and the medium in the tubes proceed through the coil in the same direction. Fig. 8B shows counter-flow in which the

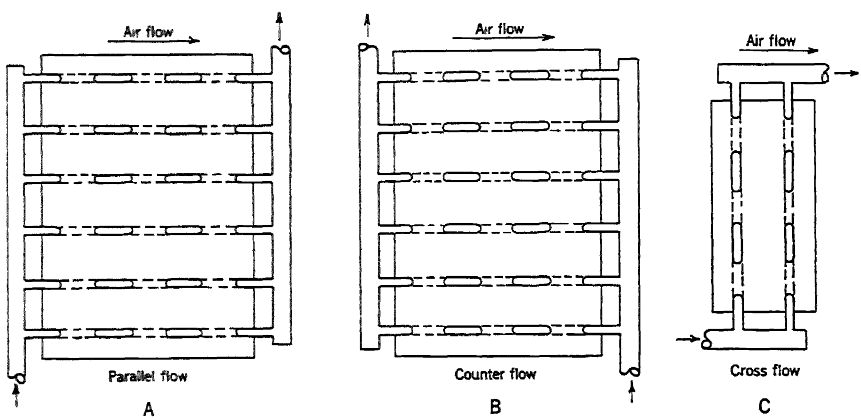


FIG. 8. FLOW OF MEDIA IN TUBES IN RELATION TO AIR FLOW

medium in the tubes proceeds in a direction opposite to the flow of air. Fig. 8C shows cross-flow in which the air and the medium in the tubes pass at right angles to each other. Parallel flow is seldom used for the reason that a lesser mean temperature difference results than with counter-flow. The counter-flow arrangement is almost universally used in brine or water coils to take advantage of the highest possible mean temperature difference for given entering water and air temperatures. It is also invariably used in coils fed with volatile refrigerant to take advantage of the higher air temperature for superheating the leaving gas. This arrangement assists complete evaporation and superheating of the refrigerant which is essential to proper operation of the thermal expansion valve. Cross-flow is common in steam heating coils, the temperature within the tubes being substantially uniform and the mean temperature difference the same whatever the direction of flow, relative to the air. Cross-flow is to be avoided in coils with volatile refrigerants on account of unequal loading of parallel circuits and danger of short circuiting of liquid refrigerant which will disturb proper functioning of the thermal expansion valve.

Applications

Heating coils in field assembled banks are used for a number of purposes as described in Chapter 20. They may be arranged with the air flow vertical or horizontal, although the latter is more common. For steam heating the coils may be set with the tubes vertical or horizontal. In the latter case the coil should be sloped to provide for condensate drainage. Because of the multi-circuit feed arrangement and the necessity for avoiding air and water pockets, water heating coils are generally arranged with the tubes horizontal. Certain precautions must be taken against freezing. Where steam coils are used with entering air below freezing temperature, throttling the steam supply may result in freezing the condensate in the bottom of the coil if the tubes are of the variety not provided with internal distributing pipes, or an equivalent arrangement.

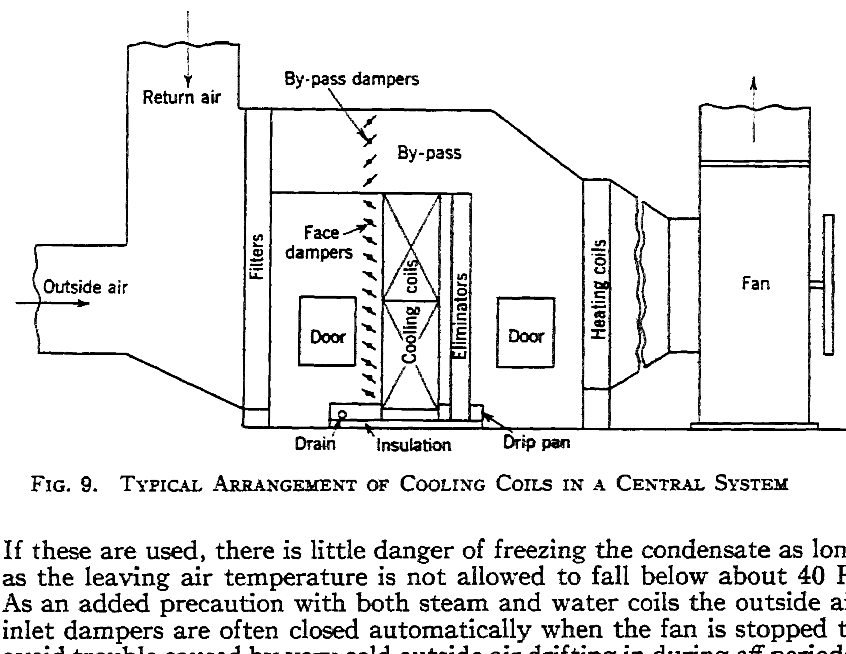


Fig. 9. Typical Arrangement of Cooling Coils in a Central System

If these are used, there is little danger of freezing the condensate as long as the leaving air temperature is not allowed to fall below about 40 F. As an added precaution with both steam and water coils the outside air inlet dampers are often closed automatically when the fan is stopped to avoid trouble caused by very cold outside air drifting in during *off* periods.

A typical arrangement of water cooling coils is shown in Fig. 9. Some means should be provided to filter all the entering air to keep dirt and foreign matter from accumulating on the coils. The assembly is provided with a drip-pan to catch the condensate during summer dehumidifying duty and to collect the non-evaporated water from the humidifying sprays in winter. The drip connection should be made ample in size and liberally provided with plugged tees and crosses for cleaning. It should not be exposed to freezing temperatures in winter if the apparatus is used on winter humidifying duty. Access doors should be provided for servicing filters, humidifying nozzles, and fan bearings and for cleaning the coils. With certain designs of coils when used for dehumidifying, eliminators must

be used beyond the coil to catch any water which may be blown into the air stream. It is customary to include these eliminators when the air velocity exceeds about 450 fpm with the individual fins and about 600 fpm for the continuous flat fin type. Where a number of coil sections are stacked one upon another, and where the velocities are low, so that eliminators need not be used, occasional trouble results when water splashes down from one coil to the next and blows out into the air stream. In such cases drip troughs as shown in Fig. 10 are used to collect this water and conduct it to the condensate pan.

Sometimes finned surface coils on summer cooling and dehumidifying duty are provided with water sprays. These sprays are of two types. In the first type a set of spray nozzles is arranged for intermittent cleaning. The operator can wash the coils off as frequently as necessary. These

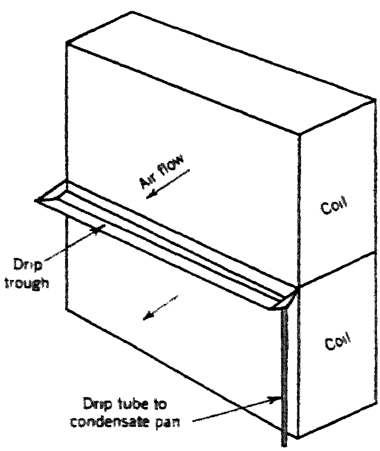


FIG. 10. COIL ARRANGED WITH DRIP TROUGH

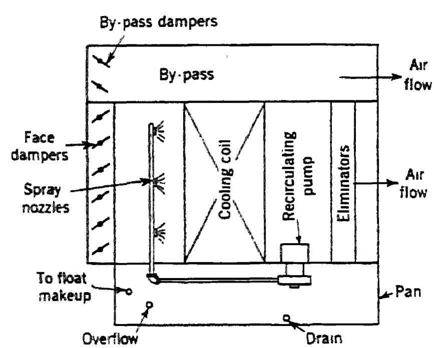


FIG. 11. RECIRCULATING SPRAY SYSTEM FOR CLEANING COILS

sprays are not operative when the system is in use and no recirculating pump is provided. The second arrangement requires a collecting tank and a recirculating pump. The water is in circulation whenever the apparatus is in operation, and assists in keeping the coil clean and in absorbing odors. Fig. 11 illustrates such an arrangement. Wherever air by-passes are used around a coil on summer duty for control purposes, it is of advantage to direct only return air through the by-pass rather than a mixture of return and outside air. The casing should be arranged accordingly. To maintain the air quantity handled by the fan reasonably constant, and to assure the required design quantity of by-passed air when the by-pass damper is open, cooling coil banks are frequently furnished with both face and by-pass dampers as shown in Fig. 9.

Although both heating and cooling coils are made of sufficient strength to take up expansion and contraction arising within themselves, care should be taken to avoid imposing strains from the piping on to the coil connections. (See Chapters 14 and 15).

HEAT TRANSFER AND AIR FLOW RESISTANCE

The transfer of heat between the heating or cooling medium and the air stream is influenced by several variables:

1. The magnitude of the driving force, i.e., the temperature difference.
2. The design and surface arrangement of the coil.
3. The velocity and character of the air stream.
4. The velocity and character of the medium in the tubes.

The driving force is usually taken as the logarithmic mean temperature difference for heating or cooling without dehumidification. For combined cooling and dehumidification, a special measure of the propelling force is used as described later. Logarithmic differences are generally employed in practice although there are special flow relationships used, such as cross-flow, where they do not strictly apply. With volatile refrigerants there is often an appreciable pressure drop and corresponding change in evaporating temperature through the refrigerant circuit. The problem is further complicated by the fact that the refrigerant is evaporating in part of the circuit and superheating in the remainder. In spite of this, heat transfers and ratings for coils using volatile refrigerants are usually based in practice on a refrigerant temperature corresponding to the average pressure in the coil.

The design and surface arrangement of the coil includes such items as materials, type, thickness, height and spacing of the fins, and the ratio of this surface to that of the tube, the use of the staggered or in-line tube arrangement, and provisions to increase the air turbulence such as the use of corrugated as against flat fins. Staggered tubes increase the total heat transfer as against the in-line arrangement and corrugated fins are more effective than flat. Of especial importance is the bond between fin and tube

The velocity of the air usually considered is the coil face velocity. This bears a varied relation to the actual velocity over the surface, depending upon the individual coil design. As long as a fixed design of coil is under consideration face velocities may be used, but they may be unsatisfactory in comparing different designs, as it is the actual surface velocity that is significant. The air volume is often based on standard air at 70 F and a barometric pressure of 29.92 in. Hg. The use of air volume in coil rating information may be misleading. The significant value is mass velocity in pounds per minute and not cubic feet per minute, because for a fixed volume the corresponding weight may vary widely, depending upon the air density, temperature and barometric pressure under consideration.

At the same mass air velocity, varying performance can be obtained depending upon the turbulence of the air flow into the coil and upon the uniformity of distribution of air over the coil face. The latter is very important in obtaining reliable test ratings and in realizing rated performance in practical installations. The resistance through the coils will assist in properly distributing the air, but where the inlet duct connections are brought in at sharp angles to the coil face, the effect is frequently bad and there may even be reverse air currents through the coils. This

reduces the capacity, but can be largely avoided by proper layout or by the use of directing baffles.

The heat transfer depends also upon the velocity of the medium in the tubes and upon its character, whether flowing water, condensing steam or evaporating volatile refrigerant. Heat transfer rates expressed as Btu per square foot of internal surface per degree logarithmic mean effective temperature difference between the fluid and tube wall are, for example, about 150 to 300 for evaporating dichlorodifluoromethane, about 350 to 1200 for water at 2 and 6 fps and about 1200 for condensing steam. The influence of the medium in the tubes on the overall heat transfer rate is, therefore, apparent.

Because of these variables, reliable rating and performance information for any design of coil must be based on actual tests on that coil under the expected conditions of operation. A comparison between the performance of two designs, unless based on such tests on each, may lead to entirely erroneous conclusions.

PERFORMANCE OF HEATING AND COOLING COILS

Heating and cooling coils are essentially heat exchangers and as such their performance depends in general upon:

1. The overall coefficient of heat transfer from the fluid within the coil to the air it heats or cools.
2. The mean temperature difference between the fluid within the coil and the air flowing over the coil.
3. The physical dimensions of the coil.

Thus, for any one definite operating condition, the heating or cooling capacity of a given coil is expressed by the following basic formula:

$$Q = U \times MTD \times A \tag{1}$$

where

Q = total heat transferred by the coil, Btu per hour.

U = overall coefficient of heat transfer, Btu per hour per square foot of external coil surface per degree Fahrenheit temperature difference between the fluid within the coil and the air flowing over the coil.

MTD = mean temperature difference, degrees Fahrenheit between the fluid within the coil and the air passing over it. (This is commonly taken as the logarithmic mean temperature difference).

A = external surface area of the given coil, square feet.

The performances of heating and cooling coils are influenced by the same factors in all but one very important exception, that is, when cooling coils operate wet or act as dehumidifying coils. For this reason, in the discussion which follows, heating and dry cooling coils are treated as one group and dehumidifying coils as another.

OVERALL COEFFICIENT OF HEAT TRANSFER

Of all factors affecting the performance of heating or cooling coils, the overall coefficient of heat transfer is the most difficult to determine as it in itself is influenced by several factors depending upon coil design and conditions of operation.

Considering any coil, whether of bare pipe or of finned type, the overall heat transfer coefficient for a given size and design of coil can always be considered as a combined effect of three individual heat transfer coefficients, namely:

1. The film coefficient of heat transfer between air and the external surface of the coil, usually given in Btu per hour per square foot external surface per degree Fahrenheit mean temperature difference.
2. The coefficient of heat transfer through the coil material—tube wall, fins, ribs, etc.
3. The film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, usually given in Btu per hour per square foot internal surface per degree Fahrenheit mean temperature difference.

These three individual coefficients acting in series result in an overall coefficient of heat transfer in accordance with the basic laws. For a *bare pipe* coil the overall coefficient of heat transfer, whether for heating or for cooling (dry), can be expressed by a simplified basic formula as follows:

$$U = \frac{1}{\dfrac{R}{h_f} + \dfrac{L}{k} + \dfrac{1}{h_a}} \tag{2}$$

where

U = overall coefficient of heat transfer, Btu per hour per square foot external surface per degree Fahrenheit mean temperature difference between air and fluid within the coil.

h_f = film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, Btu per hour per square foot internal surface per degree Fahrenheit mean temperature difference between that surface and the average fluid temperature.

h_a = film coefficient of heat transfer between air and the external surface of the coil, Btu per hour per square foot external surface per degree Fahrenheit mean temperature difference between the mass of air and the external surface.

k = conductivity of material from which the bare pipe is constructed, Btu per hour per square foot per degree Fahrenheit per inch thickness.

L = thickness of tube wall, inches.

R = ratio between external and internal surface of the bare tube, usually varying from 1.03 to 1.15 for the tube used in typical heating or cooling coils. This ratio R is inserted in the formula in order to place internal fluid coefficient of heat transfer on the basis of external surface.

Frequently, when pipe or tube walls are thin and of material having high conductivity (as is the case in construction of typical heating and cooling coils) the term L in Equation 2 becomes negligible and is generally disregarded. (The effect of the term L in typical bare pipe heating or cooling coils seldom exceeds 1 to 2 per cent of the overall coefficient). Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\dfrac{R}{h_f} + \dfrac{1}{h_a}} \tag{3}$$

For finned coils the formula[1] for the overall coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\dfrac{R}{h_f} + \dfrac{1}{xh_a}} \tag{4}$$

[1]Rational Development and Rating of Extended Air Cooling Surface, by H. B. Pownall (*Refrigerating Engineering*, October, 1935, p. 211)

Table 1. Heat Transfer Information on Typical Copper Circular Fin Coils

Dipped Metal Bond or Integral Fins

Test No.	Data Source	Working Fluid	No. of Rows	In-Line or Staggered	Tube Diam In.	O.D. Fins In.	Fins per In.	Per Cent Face Area	U at 500 FPM Face	n
	Air Heating Service									
1	Reported	Steam 2-100 lb	4	Stag	0.375	0.87	6	57	13.0	0.62
2	Research Tests	Steam 2 lb gage	2	In-line	0.625	1.37	7	42	10.8	0.61
3	Commercial	Steam 5-150 lb	1-6	Stag	0.625	1.44	8	50	10.7	0.52
4	Commercial	Steam 5-150 lb	1-6	Stag	0.625	1.37	7	50	10.5	0.57
5	Commercial	Steam 5-100 lb	1-6	....	0.625	....	7	....	10.5	0.60
6	Research Tests	Steam 5 lb gage	4	Stag	0.625	1.47	7	53	10.5	0.65
7	Commercial	Steam 5-150 lb	1	Cast-Iron Sec.	....	....	..	44	10.2	0.62
8	Research Tests	Steam 2 lb gage	2	In-line	0.625	1.12	6	37	10.1	0.55
9	Research Tests	Steam 5 lb gage	1	....	0.625	1.47	7	52	9.3	0.52
10	Research Tests	Steam 5 lb gage	1	....	0.625	1.37	7	56	9.1	0.49
11	Commercial	Steam 5-150 lb	6	Cast-Iron Sec.	....	....	..	44	9.1	0.62
12	Research Tests	Steam 2 lb gage	1	....	0.625	1.37	7	49	9.0	0.48
13	Research Tests	Steam 2 lb gage	1	....	0.625	1.12	6	37	9.0	0.46
14	Commercial	Steam 5 lb gage	1	Cast-Iron Sec.	....	....	..	....	7.0	0.50
15	Commercial	Steam 5 lb gage	6	Cast-Iron Sec.	....	....	..	....	6.5	0.60
	Air Cooling Without Dehumidification									
16	Commercial	Water 2 fps	1-6	Stag	0.625	1.37	7	50	8.9	0.47
17	Commercial	Water 2 fps	1-6	Stag	0.625	1.44	8	50	8.6	0.45
18	Reported	Water 2 fps	....	....	0.625	1.44	8	50	8.5	....
19	Research Tests	Water 2 fps	2	Stag	0.75	1.75	7	55	7.2	0.45
20	Research Tests	Water 2 fps	6	Stag	0.75	1.75	7	55	7.1	0.64
21	Commercial	Direct Expansion	1	....	0.75	1.62	4-6	....	5.8	0.62
22	Commercial	Direct Expansion	6	....	0.75	1.62	4-6	....	4.7	0.61
23	Research Tests	Direct Expansion	4	Stag	0.75	1.75	7	44	6.2	0.51

in which the term x, called the *fin efficiency* is introduced to allow for the resistance to heat flow encountered in the fins.

The term R, in this case, is the ratio of *total* external surface to internal surface. For typical designs of finned coils for heating or cooling, this ratio varies from 10 to 30. The use of this term R is again introduced to place the internal fluid film coefficient of heat transfer on a basis of external surface.

Whereas the Equation 4 appears to be comparatively simple, it is actually extremely difficult to use in practice due to lack of handbook data relating to its component terms h_f, x and h_a.

The difficulty of obtaining the values of terms h_f, x and h_a is due to the fact that they in themselves depend upon several factors.

The internal fluid film coefficient, h_f, depends upon: (1) nature of the fluid—i.e., its chemical composition, (2) velocity of the fluid within the tubes or pipes, (3) temperature of the fluid, (4) whether fluid is boiling or not, e.g., it may be boiling refrigerant or cold water, (5) the rate of boiling or the heat load upon unit area, and (6) whether the fluid is condensing, evaporating or liquid without changing state.

The air film coefficient of heat transfer, h_a, in turn depends upon: (1) air velocity over the tubes and the fins, (2) tube diameter, (3) tube spacing, (4) tube arrangement (staggered or parallel), (5) fin spacing, (6) fin design (flat or corrugated), and (7) air temperature and density.

The fin efficiency term x is even more difficult to obtain by computation because it involves very complicated differential equations. Practically, however, it is known that the fin efficiency term x is affected by: (1) external fin diameter or length, (2) internal fin diameter, actual or effective, (3) fin thickness, (4) material of construction, (5) fin cross-section area in radial direction, and (6) bond between fins and pipe or tubes.

From the foregoing, it is obvious that rating or selection of cooling or heating coils requires careful consideration of all factors involved and that hasty application of the involved theory is apt to produce unsatisfactory results.

Fortunately, for all practical purposes, many of the complex relationships given above can be combined into one and the effect be measured by laboratory tests with a comparative simplicity for any one given coil under fixed conditions of operation.

Thus, for a given coil design, either for heating or for dry cooling, its overall coefficient of heat transfer can be expressed by a simple empirical formula:

$$U = CW^n \tag{5}$$

where

U = overall coefficient of heat transfer, Btu per hour per square foot external surface per degree Fahrenheit mean temperature difference between air and internal fluid.

W = air mass velocity, pounds per hour per square foot of coil face area.

n = exponent depending upon air turbulence which is affected by: (1) tube diameter, (2) tube spacing and arrangement, (3) fin design and spacing, and (4) coil depth. For values of n for typical cooling coils see Table 1.

C = constant which is dependent upon all other factors affecting U as explained in the previous discussion.

GRAPHICAL SOLUTION OF A COIL CAPACITY PROBLEM

After the value of the overall coefficient of heat transfer for any given cooling or heating coil has been determined, it remains to compute the

coil capacity in relation to the total air flow over the coil as well as in relation to the overall coefficient of heat transfer.

The capacity of any heating or dry cooling coil can be expressed by either one of the two following equations:

$$Q_1 = W \times 0.24 \times (t_1 - t_2) \tag{6}$$

where

Q_1 = total coil capacity, Btu per hour.
W = total weight of air passing over the coil, pounds per hour.
0.24 = specific heat of air, Btu per pound per degree Fahrenheit.
$t_1 - t_2$ = difference in temperature of air entering and leaving coil, degrees Fahrenheit.

and

$$Q_2 = U \times MTD \times A \tag{7}$$

where

Q_2 = total coil capacity, Btu per hour.
U = overall coefficient of heat transfer between the air and the fluid within the coil, Btu per hour per square foot per degree Fahrenheit.
MTD = logarithmic mean temperature difference, degrees Fahrenheit.
A = total external coil surface, square feet.

Obviously, the closer the leaving air temperature approaches the temperature of the fluid within the coil, the greater will become the value of the total capacity, Q_1, as determined by Equation 6, because $t_1 - t_2$ increases; and on the other hand, the smaller will become the total capacity as computed by Equation 7, because MTD decreases. Since the Equations 6 and 7 are independent of each other, the graphical balance between them is the simplest method of solution. A typical graphical approach to solution of the coil capacity is shown in Fig. 12. Obviously, the heat loss or gain in the heating or cooling medium must equal the heat gain or loss of the air. Therefore, the actual coil capacity must lie at the intersection of the straight line curve representing Equations 6 and 7.

COIL PERFORMANCE OPERATING WITH DEHUMIDIFICATION

The foregoing theory pertaining to heating and cooling coils (operating dry) applies equally to the performance of cooling coils operating with dehumidification. However, there are also other factors which affect the performance of cooling coils operating wet which have no influence on either heating coils or cooling coils operating without dehumidification.

Since the force which causes condensation of moisture from the air upon the external surface of the cooling coil has its origin in the vapor pressure difference between the mass of air passing over the coil and that corresponding to temperature of the wetted coil surface, it is obvious that the latent coefficient of heat transfer depends upon several factors:

1. The average dew-point temperature of the air passing over the cooling coil.
2. The average temperature of the external coil surface (fins and tubes) exposed to the air stream.
3. Other factors applicable to dry cooling coils, as discussed, also affect the coefficient of latent heat transfer.

Furthermore, the coil may be wet throughout or wet at the cooler points and dry elsewhere. In the latter case the action is partly as a dry, and partly as a dehumidifying surface. As it is difficult to determine the proportions, approximations must be used. Since the dew-point temperature of the air passing over the coil is dependent upon the inside and outside atmospheric design conditions, it is not subject to control at will and must be accepted as found. Consequently, regulation of the average temperature of the external coil surface is the primary means for controlling the moisture removal from the air passing over the coil.

Total Capacity of Coil Operating With Dehumidification

While a coil operating dry transfers heat to or from the air by merely changing the temperature of the air, the cooling coil operating wet also

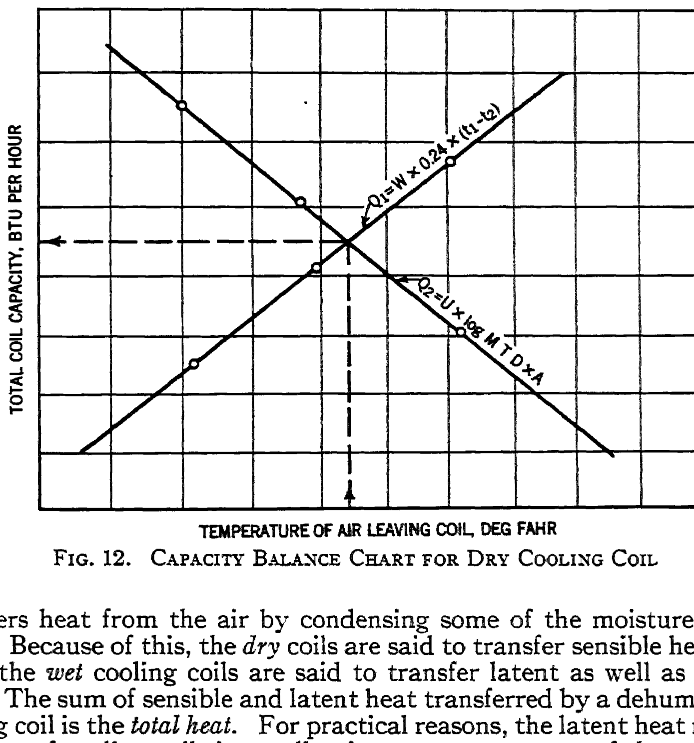


Fig. 12. Capacity Balance Chart for Dry Cooling Coil

transfers heat from the air by condensing some of the moisture carried by it. Because of this, the *dry* coils are said to transfer sensible heat only, while the *wet* cooling coils are said to transfer latent as well as sensible heat. The sum of sensible and latent heat transferred by a dehumidifying cooling coil is the *total heat*. For practical reasons, the latent heat removal capacity of cooling coils is usually given as a percentage of the total heat removal, and this percentage is called the heat load ratio.

Although much research has been already conducted and more is in progress, there is no general agreement as to the most satisfactory and convenient manner in which to rate dehumidifying coils. A large number of methods are now in use, most of them combinations of theory and empirical presentations of test data. Information is given, usually in the form of tables or curves to determine the total cooling capacity, with supplementary devices to ascertain the proportions of sensible and latent.

Some of the methods now used require trial and error solutions and others are of very questionable accuracy, although the error may be small when applied within narrow limits of the several variables.

In rating dehumidifying coils, there are two requirements. The total capacity must be determined and the proportion of sensible and latent heat transfer ascertained. These determinations often involve an average coil surface temperature. This may be determined experimentally by the use of thermocouples or calculated theoretically from other test data. Sometimes, a fictitious effective temperature is used, determined by the point of intersection between the saturation curve on the psychrometric chart and a straight line drawn through points representing the entering and leaving air conditions.

The total cooling capacity is determined in a variety of ways, of which the following are the most usual:

1. The use of surface or overall heat transfer coefficients in conjunction with dry-bulb mean temperature differences. The result is corrected for dehumidification by means of functions for: (1) the temperature differences, (2) the expected total to sensible load ratio, and (3) empirical factors determined from test.

2. The use of surface or overall coefficients for combined sensible and latent heat removal with a wet coil, using as the driving force the difference between enthalpy of the entering air and that of saturated air at either the surface or the refrigerant temperature.

3. The calculation of sensible and latent capacities separately. The sensible is based on dry-bulb mean temperature difference and heat transfer coefficient while the latent is determined using a dew-point mean difference and a corresponding latent heat transfer coefficient.

4. The use of a *contact factor* or ratio of heat removed to heat removable. This factor is a function of coil depth and air velocity, is experimentally determined for each design and used in conjunction with a so-called *surface temperature.*

Average Effective Coil Temperature

The relationship between the average external coil surface temperature and the dehumidifying capacity of cooling coils has been studied by several investigators and their studies have established an empirical but practically accurate rule:

If air at given conditions of dry- and wet-bulb temperature is passed over a cooling coil of *constant external surface temperature*, then the latent heat removed by the coil is always a definite percentage of the total heat removed—this is practically so regardless of the air velocity over the coil, the coil design, the kind of cooling medium within the coil or its flow characteristics.

It is obvious that coil designs and the common cooling mediums employed do not result in a uniform external coil surface temperature; however this uniform condition is not necessary for the rule to be usefully applied. All that is needed is that a method be found for determining the average integrated *effective* external overface temperature. The actual temperatures of the various parts of fins or tubes have relatively little effect upon the ratio of latent to total heat transfer. This rule is of importance in the testing, rating and selection of cooling and dehumidifying coils, when the *straight line method* is used for finding the average effective coil surface temperature.

If a psychrometric chart is constructed so that equal increments along the horizontal axis represent equal changes in sensible heat content and equal increments along the vertical axis represent equal changes in latent

heat content of air, then on such a chart the performance of a dehumidifying cooling coil may be conveniently represented by a straight line, as shown in Fig. 13. Point A represents the condition of return or recirculated air, point B that of outside air, point C the mixture of ⅔ recirculated air with ⅓ outside air, point D represents the condition of air leaving the cooling coil, and point E represents the average effective temperature of the external cooling coil surface. On Fig. 13 the horizontal distance between points C and D represents the sensible heat removed from the air, while the vertical distance represents the weight of moisture or the latent heat removed from the air. Another important practical feature is that a line drawn anywhere else within the body of the chart parallel to the line C-E represents the same ratio of latent to total heat removal

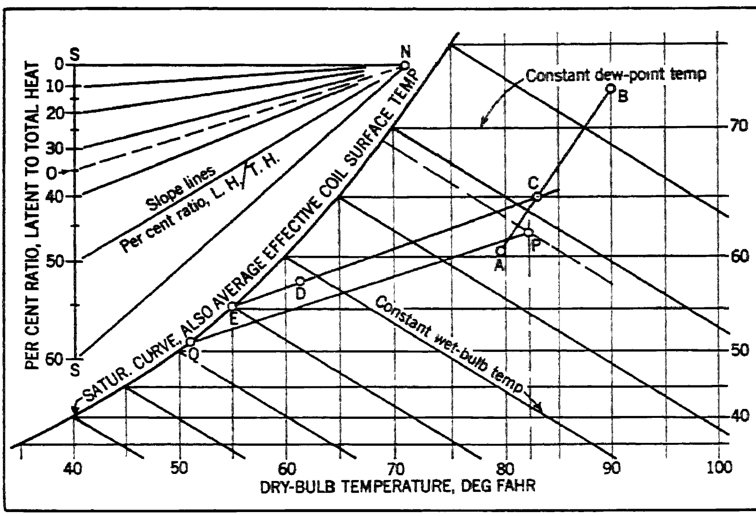


FIG. 13. PSYCHROMETRIC CHART SHOWING STRAIGHT-LINE METHOD FOR REPRESENTING COIL PERFORMANCE

as does line C-E. This is due to the fact that equal vertical distances anywhere on the chart represent equal quantities of latent heat and equal horizontal distances represent equal quantities of sensible heat.

To enhance the practical usefulness of the psychrometric chart illustrated in Fig. 13, a set of marked master slope lines are included. The value of this arrangement is easily illustrated by the graphical example shown.

Example 1. To determine the required average effective external coil surface temperature. *Given:* (1) Air entering cooling coil at temperature of 83 F dry-bulb and 69 F wet-bulb. (2) Ratio of latent to total heat that must be removed from air is 35 per cent. *Required:* To find the average external coil surface temperature.

Solution. (1) Draw through point N, at the origin of the heat load ratio lines, a line N-O with a slope of 35 per cent in accordance with scale S. (2) Mark in the body of the chart, point P representing the condition of air entering the cooling coil at 83 F dry-bulb and 69 F wet-bulb. (3) Through point P draw a line P-Q parallel to line N-O. (4) The line P-Q intersects the saturation curve at 51 F, which means that the effective external coil surface temperature must be maintained at 51 F in order to obtain the desired 35 per cent latent to total ratio of heat removal from the air passing over the given cooling coil.

Overall Coefficient of Heat Transfer

The overall coefficient of heat transfer for cooling coils operating with dehumidification is actually a sum of two overall coefficients: (1) the sensible overall coefficient due to flow of sensible heat under the pressure of temperature difference between the air and the cooling medium and (2) the latent overall coefficient due to flow of the latent heat of condensation from the moisture condensed due to difference in temperature between air dew-point and the external coil surface.

The relationship of the sensible and latent overall coefficients to the average external coil surface temperature and the temperature of the cooling medium within the coil is very complicated and, therefore, not subject to practical computations. Fortunately, an empirical relationship has been established by experimental data which permit a practical means of determining heat transfer in cooling coils operating with dehumidification.

On the basis of experimental data, it has been established that the total heat (sensible plus latent) transferred to a dehumidifying cooling coil by the air passing over it is for all practical purposes a function of the difference between the wet-bulb temperature of the air and the external coil surface temperature. The general formula for *total* (cooling and dehumidification) capacity of any given *wet* cooling coil operating at a definite fixed set of conditions can be expressed by the equation:

$$Q = U \times MTD \times A \qquad (8)$$

where

Q = total (sensible and latent) heat transferred by the coil, Btu per hour.

U = overall total (sensible plus latent) coefficient of heat transfer, Btu per hour per square foot of external coil surface per degree Fahrenheit difference between the wet-bulb temperature of the air passing over the coil and the temperature of the cooling medium within the coil.

MTD = mean temperature difference, degrees Fahrenheit between the wet-bulb temperature of the air passing over the coil and the temperature of the cooling medium within the coil.

A = external surface area of the given coil (fins and tubes), square feet.

The primary difficulty which accompanies the use of Equation 8 is that further computations are necessary to obtain the average (effective) external coil surface temperature before the latent to total heat removal ratio can be ascertained. A practical means of determining effective coil surface temperature has been determined[2].

For all practical purposes the performance of a given cooling coil (operating wet or dry) can be presented in a graphical form without sacrifice of accuracy within the limits of most air conditioning practice. A typical graphical presentation of a cooling coil performance (operating with dehumidification) is shown in Fig. 14. An analysis of this chart indicates how a given coil's performance may be affected by a change in any of the variables enumerated herewith.

1. Wet-bulb temperature of air entering the coil, degrees Fahrenheit.
2. The average (effective) external coil surface temperature, degrees Fahrenheit.

[2]Graphical Method of Determining Finned Coil Capacities Described, by E. P. Wells (*Heating, Pipin and Air Conditioning*, December, 1936, p. 665).

3. Mass air velocity through the cooling coil, pounds per hour per square foot coil face area. (This mass velocity includes linear velocity as fpm and air density as pounds per cubic foot).

Construction of a coil capacity chart such as shown in Fig. 14 is accomplished as outlined herewith:

1. A series of tests are run to determine the sensible, latent and total coil capacity, changing one at a time such variables as: air velocity, inlet air dry-bulb, inlet air wet-bulb, average refrigerant temperature within the coil, and total load upon the coil.

2. From test data obtained in (1) the ratio of latent to total heat removal is computed for various test runs and tabulated against the wet-bulb temperature of air at coil inlet and total coil capacity.

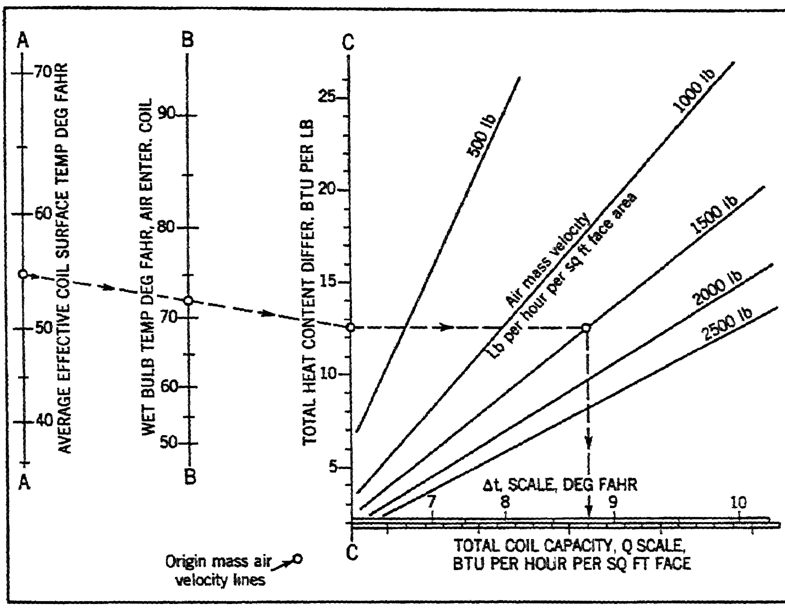


FIG. 14. TYPICAL COIL PERFORMANCE CHART

3. The three axes of the nomogram on the left side of the chart are drawn in such a manner that the C axis represents the differences in total heat content between the air at wet-bulb temperature along B axis and air at wet-bulb temperature along A axis. Thus, the C axis represents the total heat (Btu per pound of air, sensible and latent) which could be removed from the air at some inlet wet-bulb temperature on B axis if the coil heat transfer efficiency were 100 per cent and the wet-bulb temperature of the air could be reduced to some average (effective) external coil temperature on A axis. For example if a straight line is drawn through 72 F wet-bulb temperature of entering air on axis B and the 55 F average effective coil (external surface) temperature on axis B, then this straight line will intersect the C axis at 12.6, which figure represents the difference in total heat content of air between 72 and 55 F wet-bulb temperature.

4. Next scale Q is drawn to cover the range of the likely practical loading for the given coil in Btu per hour per square foot coil face area.

5. Lastly, the diagonal mass air velocity lines are drawn in at the intersection of various values on C axis and the corresponding values on the Q scale. The values on the Q scale corresponding to various values on C axis are obtained by multiplying the

values on C axis by mass air velocity and by heat transfer efficiency of the given coil which depends upon coil design and refrigerant used.

6. Parallel to the Q scale is also shown what is commonly referred to as the Δt scale. This scale gives the difference between the average effective coil (external surface) temperature and the average temperature of refrigerant within the coil. As shown in Fig. 14, the Δt scale applies only to a direct expansion coil of one definite physical design. Any change in coil design usually results in change of Δt values.

Coils with Water as the Cooling Medium

All that has been brought out in discussion of the coil capacity chart in Fig. 14, with the exception of the Δt scale, holds equally well for coils employing either vapor refrigerant or water as the cooling medium.

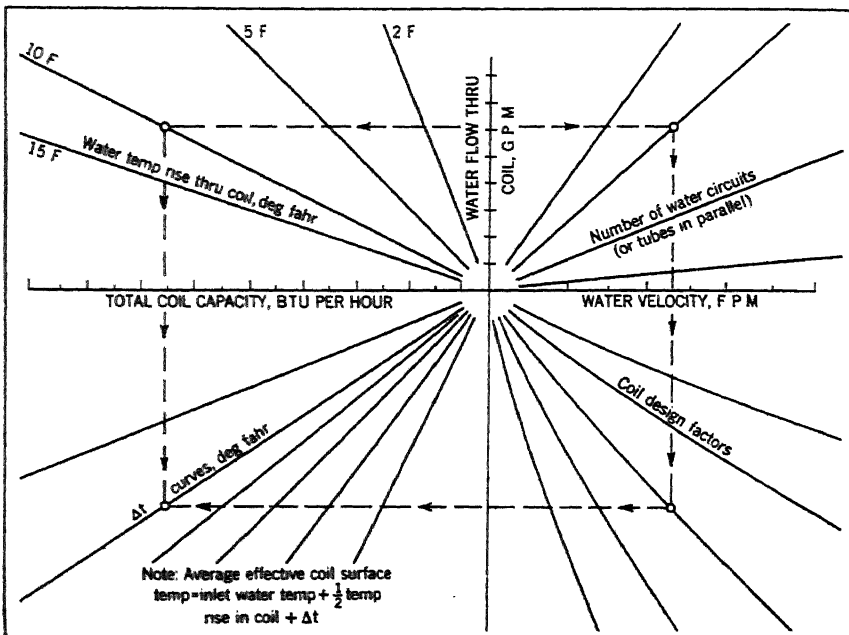


FIG. 15. WATER COIL PERFORMANCE CHART

However, the Δt scale as shown is applicable only to direct expansion coils, i.e. coils employing some volatile refrigerant as the cooling medium. Determination of the difference between the average effective coil (external surface) temperature and the average water temperature within the coil, necessary to transfer the heat from the coil surface to the water, calls for a graphical solution similar to that shown on Fig. 15. When this chart is used in conjunction with Fig. 14 a simple and rapid means is provided for determining the cooling and dehumidification capacity of coils employing water (or brine) as the cooling medium.

Factors affecting the performance of coils employing water as the cooling medium besides those shown on Fig. 14 are:

1. Quantity of water flow through the coil, usually in gpm or lb per hour.

2. Water velocity in tubes, usually in fpm. (This depends on internal tube diameter as well as the number of water circuits).

3. Coil design factor which represents the ratio of external fin and tube surface to internal tube surface.

4. The water temperature rise through the coil, degrees Fahrenheit.

Performance of Coils and Refrigeration Compressor

Practically all data published by various makers of direct expansion cooling coils are based upon maintaining a predetermined refrigerant temperature within the coils. While it is often possible to maintain a definite refrigerant temperature within a given cooling coil, for the greater part it is either impossible or impractical. This is due to the fact that the capacity of standard refrigeration compressors is usually fixed and in

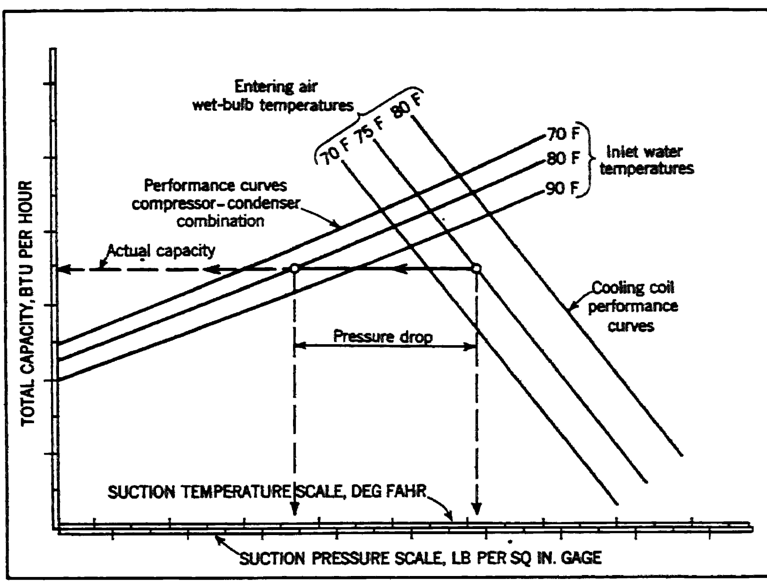


FIG. 16. GRAPHICAL ANALYSIS OF COIL-COMPRESSOR PERFORMANCE

matching a given cooling coil with a standard compressor the capacity of the latter is often somewhat smaller or greater than that of the former. Consequently, very often the refrigerant temperature resulting within a cooling coil and correspondingly the capacity of the coil-compressor combination are not what they were originally calculated to be.

In order to determine the actual performance of a given coil-compressor combination under varying conditions of operation, a graphical solution of the balance or "break-even" point is highly desirable. A typical method of graphical analysis of a coil-compressor combination performance is shown in Fig. 16, which is constructed in a manner described herewith:

1. On a piece of graph paper (with a uniform scale), the equipment capacity scale, total Btu per hour, is laid out along the vertical axis while the refrigerant suction temperature scale is laid out along the horizontal axis.

2. The performance curve of a given compressor with a definite condenser (combination usually called a *condensing unit*) is plotted as a function of suction temperature corresponding to the saturation suction pressure at the compressor suction service valve for a given inlet water temperature and quantity supplied to the condenser.

3. The performance curve of the given cooling coil is next plotted as a function of mean suction temperature within the coil, the mass air velocity over the coil and the wet-bulb temperature of air entering the coil.

4. The refrigerant pressure drop between the center of the cooling coil and the compressor suction service valve is computed and converted into the terms of temperature-difference. This temperature difference is then fitted in horizontally between the performance curves of the cooling coil and the compressor, as shown, and the total capacity of the coil-compressor combination is read along the horizontal line upon which the above mentioned *temperature-difference* segment falls.

COIL SELECTION

In the selection of a coil it is necessary to consider several factors:

1. The duty required—heating, cooling, dehumidifying.
2. Temperature of entering air—dry-bulb only if there is no dehumidification, dry- and wet-bulb if moisture is to be removed.
3. Available heating and cooling media.
4. Space and dimensional limitations.
5. Air quantity limitations.
6. Allowable resistances in air circuit and through tubes.
7. Peculiarities of individual designs of coils.
8. Individual installation requirements, such, for example, as type of automatic control to be used.

The duties required may be determined from information in Chapters 3, 4, 5 and 6. There may or may not be a choice of cooling and heating media, as well as temperatures available, depending upon whether the installation is new or is in combination with present sources of heating or cooling. Space limitations are dictated by the requirements of individual cases. The air quantity is influenced by a number of considerations. The air quantity through heating coils is often made the same as that necessary to handle the summer cooling load. The air handled may be fixed by the use of old ventilating ducts as an air distribution system for new air conditioning apparatus, or may be dictated by requirements of satisfactory air distribution or ventilation. The resistance through the air circuit influences the fan horsepower and speed. This resistance may be limited to allow the use of a given size of fan motor, or to keep the operating expense low, or it may be limited by the maximum fan peripheral velocity which requirements of quietness may permit. The friction through the water or brine circuit may be dictated by the head available from a given size of pump and pump motor. As the fan and pump motor inputs represent a refrigerating load on cooling installations, it is economical to keep them low.

Proper performance of a surface heating or cooling coil depends upon correct choice of the original equipment and upon certain other factors. The usual coil ratings are based on a uniform face velocity or air. If the air is brought in at odd angles or if the fan is located so as to block part of the air flow, the performance as given in the manufacturer's ratings cannot usually be obtained. To obtain this performance it is necessary also that the air quantity be adjusted on the job to that used in deter-

mining the coil selection, and must also be kept at this value. The most common causes for a reduction of air quantity are the fouling of the filters and collection of dirt in the coils. These difficulties can be avoided by proper design and proper servicing. There are a number of ways in which coils may be cleaned. A common method is to wash them off with water. They can sometimes be brushed and cleaned with a vacuum cleaner. In bad cases of neglect, especially on restaurant jobs where grease and dirt have accumulated, it is sometimes necessary to remove the coils and wash off the accumulation with steam, compressed air and water, or hot water. The most satisfactory solution, however, is to keep the filters serviced, and thus make the cleaning of the coils unnecessary.

The proper selection of coils requires an understanding of the necessities of each case and should be based on an economic analysis of the plant design as a whole. No general rule can, therefore, be laid down for the selection of heating or cooling coils. It is possible, however, to point out the limits of usual practice and to indicate the influence of the variables involved in the coil selection.

Heating Coils

Steam and hot water heating coils are usually rated within these limits:

Air Face Velocity—200 to 1200 fpm, sometimes up to 1500 fpm.

Steam Pressure—2 to 200 lb, sometimes up to 350 lb per square inch.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Individual cases may deviate widely, but the tabulation given herewith will serve as a guide to usual heating practice:

Air Face Velocity—500 to 800 fpm face, 500 being a common figure.

Delivered Air Temperature—varies from about 72 F for ventilation only to about 150 F for complete heating.

Steam Pressure—2 to 10 lb, 5 lb being common.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Water Quantity—Based on about 20 F temperature drop through a hot-water coil.

Air Resistance—The total resistance through heating coils is usually limited to from ⅜ to ⅝ in. of water gage for public buildings, to about 1 in. for factories.

The selection of heating coils is relatively simple as it involves dry-bulb temperatures and sensible heat only, without the complication of simultaneous latent heat loads, as in cooling coils. For a given duty, entering air temperature, and steam pressure it is possible to select several arrange-

Table 2. Several Heating Coil Arrangements

Selection	1	2	3
Steam pressure, lb per sq in.	5	5	5
Temperature of air entering coil, deg F	40	40	40
Temperature of air leaving coil, deg F	129	129	129
Air quantity, cfm	10,000	10,000	10,000
Coil face area, sq ft	33.3	12.5	7.50
Coil rows deep	2	3	4
Face velocity, fpm	300	800	1330
Air friction, in. water	0.044	0.396	1.077

ments of the same design of coil depending upon the relative importance of space, cross-sectional area, and air resistance. Table 2 shows an example.

Cooling Coils

The usual range of ratings for cooling and dehumidifying coils are enumerated herewith:

Entering Air Dry-Bulb—60 to 100 F.

Entering Air Wet-Bulb—50 to 80 F.

Air Face Velocities—300 to 800 fpm, (sometimes as low as 200 and as high as 1200).

Volatile Refrigerant Temperatures—25 to 55 F, at coil suction outlet.

Water Temperatures—40 to 65 F.

Water Quantities—2 to 6 gpm per ton, or equivalent to a water temperature range of from 4 to 12 F.

Water Velocity—2 to 6 fps.

The ratio of total to sensible heat removed varies in practice from 1.00 to about 1.65, i.e., sensible heat is from 60 to 100 per cent of total, depending on the application. (See Chapter 20, Table 1). Required ratios may demand wide variations in air velocities, refrigerant temperatures, and coil depth, so that general rules as to these values may be misleading. On usual comfort installations air face velocities between 400 and 600 fpm are frequent, 500 being a common value. Refrigerant temperatures will ordinarily vary between 40 and 50 F where cooling is accompanied with dehumidification. Water velocities will range from 2 to about 6 fps.

When no dehumidification is desired, for which condition the dew-point of the entering air will be equal to or lower than the cooling coil temperature, the coil selection is made on the basis of dry-bulb temperatures and sensible heat transfers only, the same as with heating coils. It is possible also to choose various arrangements of face area, depth, air velocity, etc., for the same duty, as illustrated in Table 2 for a steam coil.

Dehumidifying Coils

The selection of coils for combined cooling and dehumidifying duty is more involved than for heating or sensible cooling and requires consideration of both dry- and wet-bulb air temperatures. It is further complicated by the fact that the proportional amount of dehumidification required is also highly variable. The methods outlined previously under Heat Transfer and Resistance may be used to determine whether it is possible for a coil to perform the duty required. If entering and leaving air conditions are arbitrarily specified, the corresponding duty sometimes cannot be obtained at all without the use of reheat. As with heating and sensible cooling coils, there are combinations of face areas, depth, air velocity and refrigerant temperatures which will give the required performance. This is illustrated in Table 3.

It is possible as shown in Table 3 to perform approximately the same duty at a given refrigerant temperature with small face area and large thickness or vice versa. The large face area coil will give low air velocity and resistance but high air quantities per ton. The coil of small face area and great depth will require small air quantities per ton of refrigeration,

Table 3. Various Cooling Coil Arrangements

Selection	1	2	3	4
Total cooling capacity, tons	100	100	100	100
Sensible cooling capacity, tons	69	69	69	69
Latent cooling capacity, tons	31	31	31	31
Ratio total to sensible heat	1.45	1.45	1.45	1.45
Air quantity, cfm	47,800	41,700	37,100	46,800
Cfm per total ton	478	417	371	468
Face velocity, fpm	325	423	500	600
Resistance, in. water	0.11	0.27	0.51	0.37
Coil face area, sq ft	147	99.0	74.2	78.1
Coil rows deep	4	6	8	4
Coil evaporator temp, deg F	45	45	45	38

high resistance and high air velocities. As shown also in Table 3 the same sensible, latent and total cooling capacity may be obtained with various refrigerant temperatures by proper choice of coil. This makes it possible to keep the evaporating temperature high enough to carry the load with a chosen size of condensing unit. High evaporating temperatures with correspondingly small compressor operating expense can be attained but at the expense of coil surface, air quantity or both. The choice will be determined by the necessities of individual installations.

For a given quantity and condition of entering air the evaporating temperature of a volatile refrigerant coil will be determined by a balance between the condensing unit and the coil. The total, sensible and latent cooling capacity can then be determined from the coil rating information. If the condensing unit and cooling coil have been properly balanced for the required load and, due to miscalculated duct resistance or improper choice of fan speed, the air quantity is reduced, the total cooling capacity will also be reduced. The decrease is generally in the sensible capacity. This is the effect also when the air by-pass or volume control is used.

It is necessary that not only the total capacity but also the sensible and latent cooling requirements both be met. The installation of an excess of coil will result in an increase in total capacity, but not a proportional gain in latent heat capacity. On installations controlled from dry-bulb temperature the operating time will be shortened because of the added sensible cooling capacity. The result will be less moisture pick-up than calculated, and higher relative humidity. If an oversize condensing unit

Table 4. Capacity Balances for Maximum and Minimum Load Conditions

Conditions	Capacity in Tons			Ratio $\frac{\text{Total}}{\text{Sensible}}$
	Total	Sensible	Latent	
Required at peak load conditions	10.90	7.90	3.00	1.38
Required at minimum load conditions	6.62	3.36	3.26	1.98
Peak load equipment balance	10.90	7.90	3.00	1.38
Same equipment balanced at minimum load conditions	9.85	6.58	3.26	1.50
Same equipment balanced at maximum load conditions with 40 per cent by-pass	8.38	5.05	3.33	1.66
Same equipment balanced at minimum load conditions with 38,800 Btu per hour reheat	6.62	3.36	3.26	1.98

is installed the opposite situation will take place. The relative humidity will be lower than estimated. This is not generally a disadvantage except that it results in a greater load from outside air than calculated, as well as in increased power consumption. If oversize equipment is furnished, a balance should be made to assure that the ratio of total to sensible capacity is the same as in the estimated load.

Sometimes arbitrary air quantities are specified for ventilation or other reasons independent of the selection of the cooling coil. As shown in Table 3, the coil selection can be altered to take care of various air quantities for the same duty.

Where coil and condensing unit are selected for the peak load condition, and the sensible load partially disappears due to fall of outside temperature or other cause, the condensing unit and coil rebalance. This may result in more sensible capacity than required at the light load condition and less latent in proportion, with an increased relative humidity in the conditioned space. Such a condition, for a typical installation, is shown in Table 4. If approximately 40 per cent of the total air is by-passed, the condition will be improved as indicated. The situation could be entirely avoided by using reheat. With sufficient reheat, it is possible to handle any ratio of sensible and latent loads and maintain the design temperature and humidity.

Care should be taken to avoid freezing at light loads. In general, freezing occurs when the coil surface temperature falls to 32 F. With usual coils for comfort installations, this will not occur unless the evaporating temperature at the coil outlet is about 20 to 25 F. The exact value depends on the design of coil and the amount of loading. Although it is not customary to choose coil and condensing units to balance at low temperatures at peak loads, there is danger of this occurring when the load decreases. This is further aggravated if a by-pass is used so that less air is passed through the coil at light loads. It may be even worse if the control is arranged for decrease of inside temperature with fall of that outside. Freezing can be avoided by making the full load balance a high evaporating temperature and checking the balance at the minimum load condition.

REFERENCES

A.S.H.V.E. Research Report No. 997—Heat Transfer from Direct and Extended Surfaces with Forced Air Circulation, by G. L. Tuve and C. A. McKeeman (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 427).

A.S.H.V.E. Research Report No. 1029—Performance of Fin-Tube Units for Air Heating, Cooling and Dehumidifying, by G. L. Tuve (A.S.H.V.E. Transactions, Vol. 42, 1936, p. 99).

A.S.H.V.E. Research Report No. 1066—Performance of Fin-Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 367).

A.S.H.V.E. Research Report No. 1099—Performance of Surface Coil Dehumidifiers for Comfort Air Conditioning, by G. L. Tuve and L. J. Seigel (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 523).

Heat Transmission in Cooling Air with Extended Surfaces, by W. L. Knaus (*Refrigerating Engineering*, January, 1935, p. 23, February, 1935, p. 82).

Dehumidification of Air with Coils, by William Goodman (*Refrigerating Engineering*, October, 1936, p. 225).

Extended Surface Cooling Units, by R. H. Swart (*Refrigerating Engineering*, February, 1938, p. 107).

Chapter 26

SPRAY EQUIPMENT

Air Washers, Apparatus for Direct Humidification, Spray Generation and Distribution, Air Dehumidification with Washers, Water Main Temperatures, Atmospheric Water Cooling Equipment, Design Wet-bulb Temperatures for Water Cooling, Cooling Ponds, Winter Freezing

AIR humidification is effected by the vaporization of water and always requires heat from some source. This heat may be added to the water prior to the time vaporization occurs or it may be secured by a transformation of sensible heat of the air being humidified to latent heat as the vapor is added to the air. The thermodynamics of the process are discussed in Chapter 1. The removal of moisture from air may or may not involve the removal of heat from the air-vapor mixture. With spray equipment dehumidification of air always necessitates the removal of heat.

AIR WASHERS

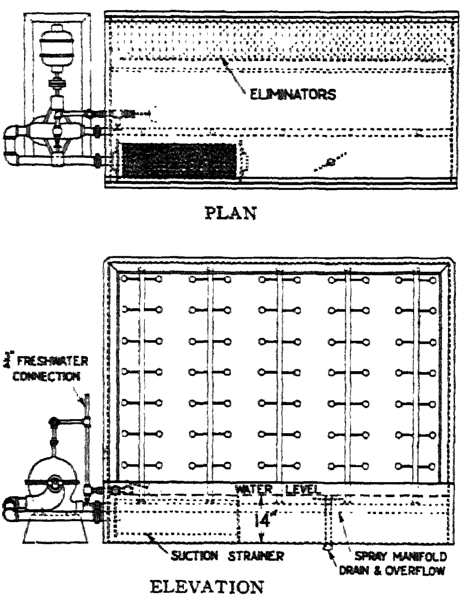
Air washers may be used as either humidifiers or dehumidifiers depending upon the method of operation and the temperature of the spray water. The functions of an air washer are to regulate the moisture and heat content of air passing through it and to remove dust and dirt from the air. Air washers are not as effective as air filters in the removal of dust and dirt.

The construction of commercial air washers is indicated in Figs. 1 and 2. Any air washer consists essentially of a chamber through which the air passes in intimate contact with water. The lower portion of the washer chamber serves as a sump for the spray water.

Contact between the air and the washer water is secured: (1) by breaking the water into a very fine mist, (2) by passing the air over surfaces which are continuously wetted by water, or (3) by a combination of water sprays and wetted plates. Scrubber-plate types of washers are used largely to wash heavy reclaimable products from the air, and are generally composed of one to three eliminator-type baffle scrubber plates across the air stream. Water is supplied at the tops of the scrubber plates by flooding nozzles placed across the top of the washer. Spray washers have one or more banks of water atomizing nozzles placed in the air stream above the level of the water in the sump. The direction of the water sprays may be against the air stream, with the air stream, or with

one bank spraying with the air stream and one against it. The number of nozzles required depends upon their design, the quantity of air handled, and the arrangement of the nozzles.

Scrubbers generally consist of eliminator-type baffle plates placed in the air stream to cause several reversals of the direction of air flow. The scrubber plates are more effective as air cleaners than as humidifiers. All washer chambers should have inlet diffuser plates to aid in producing more uniform air flow through the washer spray chamber. These inlet vanes also aid in preventing spray water from being thrown into the air duct ahead of the washer. However, if the water spray is against the air flow, the ordinary perforated diffuser plate is not sufficient, and specially designed eliminator baffles must be used to prevent spray from passing



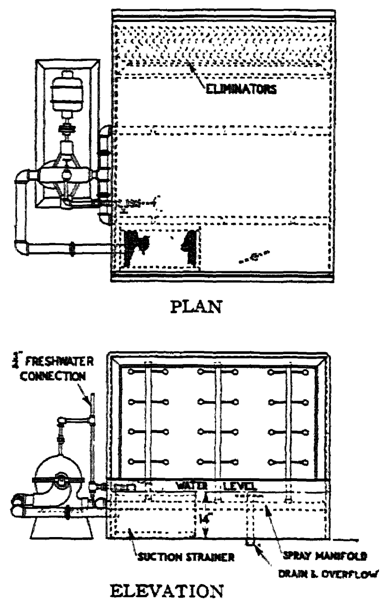


Fig. 1. Typical Single Bank Air Washer Fig. 2. Typical Two Bank Air Washer

into the air inlet duct. At the outlet end of the washer suitable flooded eliminator plates, which will cause from 4 to 6 reversals of the direction of air flow, should be installed for the purpose of removing drops of unvaporized water from the leaving air. When the air carries certain substances mixed with it, the spray water may become acidulated and special consideration must be given to the materials used, to reduce the corrosive action.

Essential items in air washer operation are: uniform distribution of the air across the chamber section above the level of the water in the sump; moderate velocities of air flow, 300 to 600 fpm in the spray chamber; an adequate amount of spray water broken up into a fine mist throughout the air stream; sufficient length of air travel through the water spray and over thoroughly wetted surfaces; and the elimination of free moisture from the air as it leaves the unit.

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the overall efficiency of heat transfer between the air and the heating or cooling medium. A multi-stage washer is equivalent to a number of washers in a series arrangement. Each stage is in effect a separate washer.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 fpm through the gross inlet area of the unit. At this rating spray type washers handle about 2½ gpm of water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed.

For a single stage air washer, a 15 F drop in dry-bulb temperature of the air passing through the washer is about the maximum that should be anticipated. For greater decrease in dry-bulb temperature, multi-stage washers should be utilized. A rise of 6 F should be the calculated maximum for the spray water.

The width and height of a washer may be dictated by space limitations outside the washer, such as headroom, or by the inside space requirements, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils. Roughly, a spray space of about 2 ft 6 in. in length is required for each bank of sprays; *leaving* eliminators require about 1 ft 6 in., and *entering* eliminators about 1 ft.

The resistance to air flow through an air washer varies with the type of eliminators, number of banks of sprays, direction of spray, air velocity, type of scrubber plates, and size and type of cooling coils if located in the washer. Manufacturers should be consulted to obtain the resistance for a particular installation.

HUMIDIFICATION WITH AIR WASHER

Air humidification can be accomplished in three ways with an air washer. These are: (1) use of recirculated spray water without prior treatment of the air, (2) preheating the air and washing it with recirculated spray water, and (3) using heated spray water. In any problem of air washing the air should not enter the washer with a dry-bulb temperature less than 35 F so that there will be no danger of freezing the spray water.

Method 1. Except for the small amount of energy added from outside by the recirculating pump in the form of shaft work, and for the small amount of heat leak from outside into the apparatus, including the pump and its connecting piping, the process would be strictly adiabatic. Evaporation from the liquid spray would therefore be expected to bring the air immediately in contact with it to saturation adiabatically; and, since the liquid is recirculated, its temperature would be expected to adjust to the thermodynamic wet-bulb temperature of the entering air.

It does not follow from the above reasoning that the whole air stream is brought to complete saturation, but merely that its state point should move along a line of constant thermodynamic wet-bulb temperature as

explained in Chapter 1. The extent to which the final temperature approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached is conveniently expressed by a ratio known as *humidifying efficiency* or saturating efficiency and defined as follows:

$$e_h = \frac{t_1 - t_2}{t_1 - t'} \tag{1}$$

where

e_h = humidifying efficiency, per cent.
t_1 = dry-bulb temperature of the entering air, degrees Fahrenheit.
t_2 = dry-bulb temperature of the leaving air, degrees Fahrenheit.
t' = thermodynamic wet-bulb temperature of the entering air, degrees Fahrenheit.

The humidifying or saturating efficiency of a washer is dependent upon the number of spray banks and nozzles, the effectiveness of the nozzles in breaking an adequate quantity of water into a fine spray, the velocity of air flow through the water sprays, and the time of the contact of the air with the spray water. Other conditions being the same, low velocities of air flow are more conducive to higher humidifying efficiencies. The following may be taken as representative humidifying or saturating efficiencies of air washers for the conditions stated:

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

The air leaving the washer may require reheating to produce the required dry-bulb temperature and relative humidity.

Method 2. The preheating of the air increases both the dry and wet-bulb temperatures, lowers the relative humidity, but does not alter the humidity ratio (pound water vapor per pound dry air). At a higher wet-bulb temperature but the same humidity ratio, more water can be absorbed per pound of dry air in passing through the washer, assuming that the humidifying efficiency of the washer is not adversely affected by operation at the higher wet-bulb temperature. The analysis of the process occurring in the washer itself is the same as that explained under Method 1. The final desired conditions are secured by adjusting the amount of preheating to give the required wet-bulb temperature at entrance to the washer and then reheating when necessary after passage through the washer.

Method 3. Even if heat is added to the spray water, the mixing occurring in the washer itself may still be regarded as adiabatic. The state point of the mixture should move in a direction determined by the specific enthalpy of the heated spray as explained in Chapter 1. By sufficiently elevating the spray water temperature it should be possible to completely saturate the air and even raise its temperature above the dry-bulb temperature of the entering air.

APPARATUS FOR DIRECT HUMIDIFICATION

Humidifiers may be divided into the following general types, according to the method of operation: (1) indirect, such as the air washer, which

introduces moistened air; and (2) direct, which sprays moisture into the room or introduces moisture by means of steam jets.

As in the cases of humidification by use of an air washer, the heat necessary for the vaporization of the moisture added to the air by direct humidification is secured either from heat stored in the spray water or by a transformation of sensible to latent heat in the air humidified. In the latter case the enthalpy of the air remains constant but the dry-bulb temperature of the air is reduced.

Direct humidification is usually preferable where high relative humidities must be maintained, but where there is little cooling or ventilation required. In comfort air conditioning, where both humidification and ventilation are required, the indirect humidifier is preferable. In industrial applications, where the cooling or ventilation load is large and where very high relative humidities must be maintained, a combined system employing both direct and indirect humidifiers is sometimes used.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained

at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray. These cannot occur when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is evaporated and the resulting vapor diffused. This distribution of fine spray over the maximum possible area promotes complete and rapid vaporization even at high humidities.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

AIR DEHUMIDIFICATION WITH WASHERS

Moisture removal from an air-vapor mixture can be accomplished by use of an air washer so long as the temperature of the spray medium is lower than the dew-point of the air passing through the unit. The final dry-bulb temperature and the relative humidity of the air leaving a dehumidifier washer are dependent upon: the air velocity, the length of air travel through the sprays, the dry- and wet-bulb temperatures of the entering air, the spray temperature, the number of spray banks and nozzles, the quantity of spray medium handled, and the effectiveness of the nozzles in breaking the spray into a fine mist.

Both sensible and latent heat are removed in the process of dehumidification by cooling. Abstraction of sensible heat occurs during the entire time that the air is in contact with the spray medium. Latent heat

removal takes place as condensation occurs. Therefore, the lower the spray temperature the greater the amount of moisture removal per pound of dry air, all other conditions remaining the same. Washers with two or more banks of sprays are usually selected for comfort air conditioning installations. Such washers will cool the air to within 1 or 2 F of the leaving spray water temperature.

Where a limited supply of cold water is available multiple stage washers may be used to an advantage. The cool water is pumped through the multiple spray systems in series. By this arrangement the entering air is cooled first by the warmer water and finally by the cooler water which gives the maximum amount of cooling with the minimum amount of water. The approximate temperatures of water from wells at depths of 30 to 60 ft are given in Fig 3[1]. Frequently the temperature of the city water main supply is low enough during the summer to permit an appreciable cooling effect. Table 1 lists the maximum city water main temperatures for various localities in the United States and Canada.

Air washers using refrigerated spray generally have their own recirculating pumps. These pumps deliver to the sprays a mixture of water from the washer sump, which has not been re-cooled, and refrigerated water. The quantities of each are controlled by a three-way or mixing valve actuated by a dew-point thermostat located in the washer air outlet or by humidity controllers located in the conditioned space.

ATMOSPHERIC WATER COOLING EQUIPMENT

In the operation of a refrigerating plant or a condensing turbine, one of the main problems is the removal and dissipation of heat from the compressed refrigerant or the discharged steam. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger, from which water it may then be dissipated in a number of ways. If the plant is situated on the banks of a river or lake, an intake may be taken up-stream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of cooling water is a city supply or a well, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where city water is too expensive or where the water available contains dissolved salts which would form scale on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-exchanger by exposure to air in an atmospheric water cooling apparatus.

Air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact. The rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. Because the changes in rate do not occur in direct proportion to changes in the governing factors, data on the performance of atmospheric water cooling equipment are largely empirical.

[1]Temperature of Water Available for Industrial Use in the United States, by W. D. Collins (*U. S. Geological Survey*, Water Supply Paper No. 520 F).

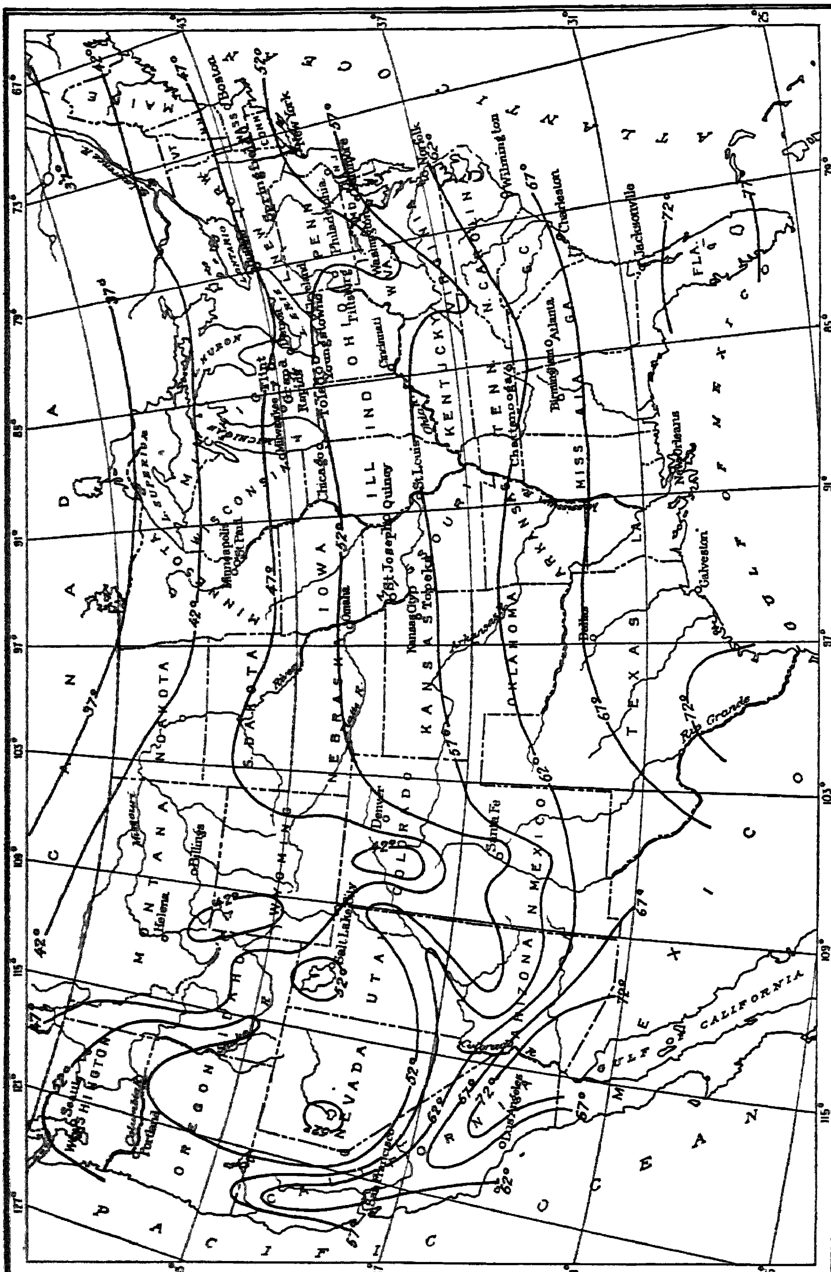


FIG. 3. APPROXIMATE WELL WATER TEMPERATURES AT DEPTHS OF 30 TO 60 FT

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURES[a]

State	City	Temp. F	State	City	Temp. F
Ala.	Birmingham	84	Mass.	Boston	80
	Mobile	73		Cambridge	70
Ariz.	Phoenix	81		Fall River	76
	Tucson	80		Lowell	50
Calif.	Anaheim	60		Lynn	68
	Berkeley	69		New Bedford	70
	Fresno	72		Salem	68
	Fullerton	75		Worcester	76
	Glendale	68	Mich.	Detroit	77
	Los Angeles	75		Flint	70
	Oakland	69		Grand Rapids	84
	Ontario	70		Highland Park	77
	Pasadena	82		Jackson	56
	Pomona	75		Kalamazoo	53
	Riverside	78		Lansing	64
	Sacramento	72		Saginaw	82
	San Bernardino	65	Minn.	Duluth	55
	San Diego	82		Minneapolis	80
	San Francisco	62		St. Paul	77
	Whittier	75	Mo.	Jefferson City	82
Colo.	Denver	75		Kansas City	84
Conn.	Bridgeport	66		St. Joseph	84
	Hartford	73		St. Louis	85
	New Haven	76		Springfield	70
	Waterbury	72	Nebr.	Lincoln	87
D. C.	Washington	84		Omaha	87
Del.	Wilmington	83	Nev.	Reno	61
Fla.	Jacksonville	80	N. H.	Manchester	76
	Miami	80	N. J.	Jersey City	63
	Tampa	77		Newark	74
Ga.	Atlanta	87		Paterson	78
	Macon	80		Trenton	79
Ill.	Chicago	76	N. Y.	Albany	68
	Cicero	76		Buffalo	75
	Evanston	73		Jamaica	56
	Peoria	67		Mt. Vernon	74
	Rockford	59		New Rochelle	75
	Springfield	82		New York	72
Ind.	Evansville	86		Rochester	70
	Gary	75		Schenectady	60
	Indianapolis	80		Syracuse	74
	South Bend	61		Utica	69
	Terre Haute	82		Yonkers	70
Iowa	Cedar Rapids	78	N. C.	Asheville	74
	Des Moines	77		Charlotte	85
	Sioux City	62		Winston-Salem	82
Kans.	Concordia	57	N. M.	Albuquerque	65
	Kansas City	86	Ohio	Akron	76
	Topeka	88		Canton	50
	Wichita	72		Cincinnati	84
Ky.	Louisville	85		Cleveland	74
La.	Baton Rouge	85		Columbus	82
	New Orleans	85		Dayton	60
Me.	Augusta	60		Lakewood	82
Md.	Baltimore	75		Springfield	72
				Toledo	83

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURE[a] (Concluded)

State	City	Temp. F
Okla.	Oklahoma City	82
	Tulsa	85
Ore.	Eugene	60
	Portland	64
Pa.	Altoona	74
	Erie	75
	Johnstown	74
	McKeesport	82
	Philadelphia	83
	Pittsburgh	81
R. I.	Providence	68
S. C.	Charleston	80
	Greenville	81
	Spartanburg	78
S. Dak.	Rapid City	55
Tenn.	Chattanooga	84
	Knoxville	89
	Memphis	70
	Nashville	90
Texas	Amarillo	65
	Austin	90
	Beaumont	86
	Dallas	86
	Fort Worth	84
	Galveston	90
	Houston	84
	Port Arthur	83
	San Antonio	76
	Wichita Falls	85
Utah	Logan	44
	Salt Lake City	60
Va.	Fredericksburg	75
	Lynchburg	73
	Norfolk	80
Wash.	Olympia	58
	Seattle	62
	Spokane	51
	Tacoma	57
W. Va.	Charleston	85
	Huntington	78
	Wheeling	78
Wis.	LaCrosse	54
	Madison	58
	Milwaukee	70
	Racine	68
Province		
Alta.	Calgary	64
B. C.	Vancouver	60
Ont.	London	50
	Toronto	63
P. E. I.	Charlottetown	48
Que.	Montreal	78
	Quebec	68

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 1.) Because it is impractical to leave the air in contact with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least possible expenditure of power.

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design, however, in that the size of the equipment is determined by the quantity of air that must be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application. With water cooling apparatus, its size and the

quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Sizes of Equipment

Assuming a definite quantity of water to be cooled, the size and design of atmospheric cooling equipment are affected by the following factors:

1. Temperature range through which the water must be cooled.
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced.
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air.)
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service (refrigerating, internal combustion engine and steam condensing).
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 lb head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 lb and power consumption increases. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the

TABLE 2. CONDENSER DESIGN DATA

Gas	Maximum Pressure Desired in Condenser	Gas Temperature in Condenser Deg F	Leaving Hot Water Temperature Deg F	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vacuum	101.2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.9	120	114
Ammonia	185 lb gage head pressure	96.0	92	88
Carbon dioxide	1030 lb gage head pressure	86.0	83	81
Methyl chloride	102 lb gage head pressure	100.0	96	92
Dichlorodi-fluoromethane	117 lb gage head pressure	100.0	96	93

quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 or 4 F of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 F difference.

Table 2 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

The temperature range, once the hot water temperature is approximately known, depends upon:

1. Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.
2. Efficiency of the atmospheric cooling equipment considered.

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1934, inclusive, there were but 8 hours per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September inclusive) when the wet-bulb temperature was 68 F or above. As these 975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for refrigeration design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City) with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on a refrigerating plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 8, shows design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmospheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb

temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

Percentage cooling efficiency of atmospheric water cooling equipment =

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}.$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is given in Table 3 although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various processes of the cooling equipment are:

Compressor refrigeration	220 to 270 Btu per minute per ton.
Condenser turbine	950 to 980 Btu per pound of steam.
Steam jet refrigerating apparatus	1030 to 1150 Btu per pound of steam.
Diesel engine	2800 to 4500 Btu per horsepower.

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line. Natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift away with the air movement. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-

TABLE 3. EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

Equipment	Cooling Efficiency—Per Cent		
	Minimum	Usual	Maximum
Spray Ponds	30	45 to 55	60
Spray Towers	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers	35	50 to 70	90
Mechanical Draft	35	55 to 75	90

bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased volume of air used, and (4) the vastly increased area of contact between air and water.

Spray pond efficiencies are increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the nozzles from 3 to 7 ft above the edge of the basin, to supply from 5 to 12 lb pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.1 gpm per square foot in a small pond to 0.8 gpm per square foot in a large pond.

Increasing the pressure, spacing the nozzles farther apart, or increasing the elevation of the nozzles will increase the cross-section of spray cloud exposed to the air, and therefore increase the quantity of air coming in contact with the water. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

Spray ponds may be located on the ground, or they may be placed on roofs. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae growths, during warm weather, in cooling towers and spray ponds may be eliminated while the plant is in operation by the use of potassium permanganate. This chemical can be dissolved at the rate of 1 lb in 1¼ to 1½ gal of hot water. About 10 parts of permanganate should be used per million parts of cooling water.

The permanganate attacks the algae, forms a brown covering over it, and causes it to settle. Enough of the permanganate solution should be added periodically to cause the water to have a pink color for a period of from 15 to 20 min. Small additions of the permanganate daily do not give concentrations which are effective. The best results are obtained when sufficient quantities are added periodically at intervals of several

weeks, the time intervals being dependent upon local operating conditions. The chemical is non-poisonous and is non-corrosive when used as directed.

Spray Cooling Towers

Where not more than 30,000 Btu per minute are to be dissipated, the spray cooling tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray cooling towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years most of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 15 to 80 ft high and from 6 to 30 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks also add to the area of water surface exposed to the air, but since they furnish a resistance to air flow, too many decks are a detriment.

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. These boards or louvers extend outward and upward, and in most designs the top edge of each louver extends above the bottom edge of the one above it.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increase the area of water exposed to the wind, and the latter increases the time of contact of the air with the water.

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling on days when the wind velocity is below average when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 1, Chapter 8, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in

no case for a wind velocity of more than 5 mph. It is obvious that natural draft towers and other natural draft equipment must be so located that they are not obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of vertical shells, constructed of wood, metal, or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood checker-work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles. Air is circulated through the tower from bottom to top by forced or induced draft fans. Since the air flows counter to the water, the air is in contact with the hottest of the water just before leaving the top of the tower, and each unit of air picks up more heat than a similar unit would on natural draft equipment, so the mechanical draft tower cools water by using less air than the other types of equipment need. As movement of the air through the towers is obtained by power-consuming fans, it is essential that the air used be reduced to a minimum so as to secure the lowest possible operating cost.

The efficiency of a mechanical draft tower is increased by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of lowered velocities it increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through mechanical draft towers vary from 250 to 600 fpm over the gross area of the structure.

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged through a duct to the outside. Such apparatus does not have the counterflow advantage of the vertical mechanical draft water cooling equipment, and therefore requires a much larger excess of air for proper operation. Air velocities and operating powers are considerably above those required by vertical mechanical draft water cooling equipment.

Cooling Tower Design

The method of design of equipment for energy transfer from water to an air-water vapor mixture is similar to that used for absorption equip-

ment. Details of this procedure are available[2, 3, 4] and its application to the problem of the cooling tower is illustrated by the following development. The nomenclature used is as follows:

a = wetted area, square feet per cubic foot of tower volume.
c = specific heat of liquid water, Btu per pound per degree Fahrenheit.
e = effectiveness.
ε = natural base.
K = overall mass unit conductance, pounds per (hour) (square foot) (pound per pound).
G = weight rate of flow of air, pounds per hour.
h = enthalpy, Btu per pound.
h_a = enthalpy of air-vapor mixture, Btu per pound.
h'' = enthalpy of saturated air-vapor mixture at water temperature, Btu per pound.
L = water rate, pounds per hour.
S = cross-sectional area of cooling tower, square feet.
t = water temperature, degrees Fahrenheit.
t_{wb} = wet-bulb temperature, degrees Fahrenheit.
V = tower volume, cubic foot.
m = slope of saturation line on a temperature enthalpy diagram.
l_m = logarithmic mean.

Subscripts 1 and 2 refer to water entrance and exit sections respectively, for the counter-flow tower.

A section of a typical counter-flow tower is shown in Fig. 4. If the reduction in water rate due to evaporation within the volume is neglected, the energy balance for this differential section of the exchanger volume may be written as:

$$L\,c\,d\,t = G\,d\,h \tag{2}$$

The potential for energy transfer from the water to the mixture in contact with it may be expressed with reasonable accuracy as the difference between the enthalpy of saturated air at the water temperature, h'', and the enthalpy of the main stream air vapor mixture, h_a. The rate of energy transfer is given by the expression:

$$Ka\,(h'' - h_a)\,dV \tag{3}$$

which equation defines the overall rate coefficient, Ka; the latter being the product of the overall mass unit conductance, K, and the ratio of the transfer surface to the exchanger volume, a.

Equations 2 and 3 are conveniently illustrated by means of the temperature-enthalpy diagram of Fig. 5. Equation 2 indicates that the succession of air and water states existing in the exchanger sections must combine to form a straight line (for L = constant) on the temperature enthalpy diagram. The slope of this operating line is $\frac{dh}{dt} = \frac{Lc}{G}$. Since the heat capacity of water is approximately unity, this slope is the ratio of the water to the air rate. Equation 3 indicates that the potential for

[2] Principles of Chemical Engineering, by W. H. Walker, W. K. Lewis, W. H. McAdams and E. R. Gilliland (McGraw-Hill Co., New York City, 1937, p. 480).

[3] Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., New York City, 1937, p. 91).

[4] Performance Characteristics of a Mechanically Induced Draft, Counterflow, Packed Cooling Tower, by A. L. London, W. E. Mason and L. M. K. Boelter (*A.S.M.E. Transactions*, January, 1940, Vol. 62, p. 41).

energy transfer at any section is the difference between the enthalpy of saturated air at the water temperature of that point and the enthalpy of the air in contact with that water. This potential is the difference in ordinates between the saturation and operating lines for the water temperature at the point considered.

Combination of equations 2 and 3 results in the expression:

$$Gdh = Ka\ (h'' - h_a)\ dV \tag{4}$$

Integrating this equation over the length of the exchanger:

$$\int_2^1 \frac{G}{Ka} \frac{dh}{h'' - h_a} = V \tag{5}$$

The integration of the left side of Equation 5 determines the tower volume required to achieve the desired energy exchange. This summation is readily accomplished for counter and parallel flow arrangements. G and

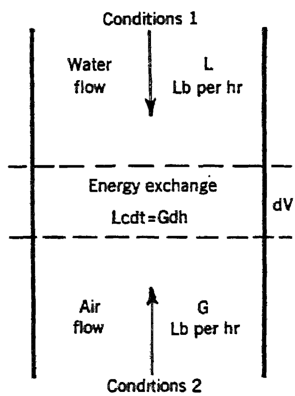


FIG. 4. SECTION OF TYPICAL COUNTER-FLOW TOWER

Ka are usually independent of the tower volume and Equation 5 then becomes:

$$\int_2^1 \frac{dh}{h'' - h_a} = \frac{KaV}{G} \tag{6}$$

$\frac{KaV}{G}$ is defined as the *Number of Transfer Units* (*NTU*).

The integration is made numerically or graphically. In the graphical integration $\frac{1}{h'' - h_a}$ is evaluated as a function of h_a. This determination involves the use of the energy balance equation integrated from one section to the section in question. The area under the curve between any two abscissae is the number of transfer units required to change the air state from h_2 to h_1.

The overall rate coefficient, Ka, must be known if the tower volume is to be determined. Experiments conducted on towers containing different packing construction have yielded some magnitudes of this coefficient,

evaluated on an overall basis. These data are presented in Fig. 6 as a function of the gas mass velocity through the packing, and apply only to the particular packing structure for which they were obtained. *Ka* may also be a function of the water rate, since a reduction of the water rate may reduce the wetted area within the exchanger[5]. The results included in Fig. 6 probably represent values of *Ka* obtained with complete wetting.

A typical design procedure is illustrated in the example following;

Example 1. The rate of air flow, arbitrarily assumed in the data given, is related to the tower volume by economic considerations. A balance between air rate and tower volume rests on consideration of the costs of producing air flow and of the tower construction[6]. A counter-flow forced draft cooling tower is to cool 36,000 lb of water per

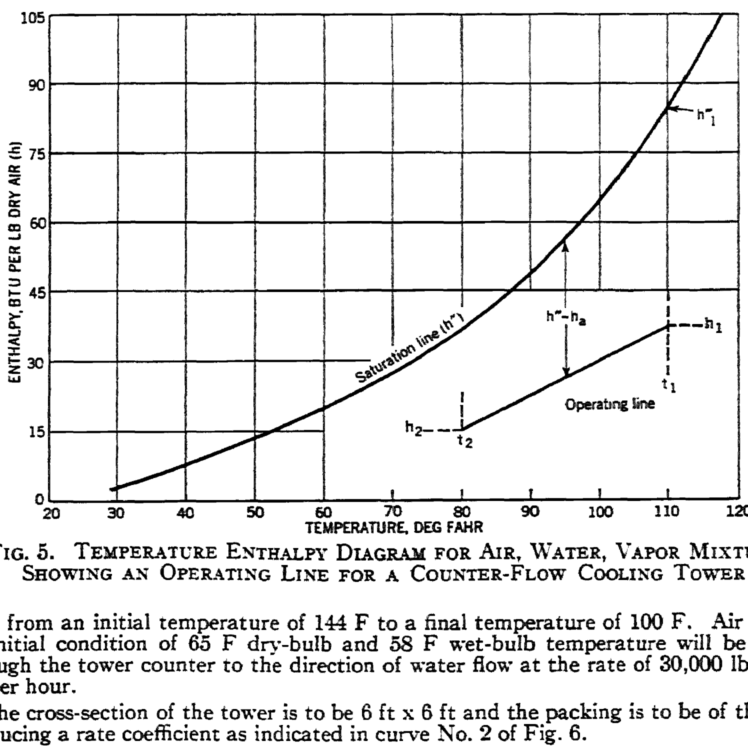


FIG. 5. TEMPERATURE ENTHALPY DIAGRAM FOR AIR, WATER, VAPOR MIXTURE SHOWING AN OPERATING LINE FOR A COUNTER-FLOW COOLING TOWER

hour from an initial temperature of 144 F to a final temperature of 100 F. Air having an initial condition of 65 F dry-bulb and 58 F wet-bulb temperature will be forced through the tower counter to the direction of water flow at the rate of 30,000 lb of dry air per hour.

The cross-section of the tower is to be 6 ft x 6 ft and the packing is to be of the type producing a rate coefficient as indicated in curve No. 2 of Fig. 6.

Solution:

Initial air enthalpy = 25 Btu per pound.

Final air enthalpy:

$$(h_1 - h_2) = \frac{Lc}{G}(t_1 - t_2)$$

$$(h_1 - 25) = \frac{36{,}000 \times 1}{30{,}000}(144 - 100)$$

$$h_1 = 77.8 \text{ Btu per pound dry air.}$$

[5]Loc. Cit. Note 4.

[6]Loc. Cit. Note 2, p. 142.

A numerical integration (Table 4) is employed to determine the Number of Transfer Units (NTU) required. Temperature increments of 2 F are used between successive determinations of the quantity $\frac{1}{h'' - h_a}$. The energy balance indicates that the enthalpy increments corresponding to these temperature increments are:

$$\Delta h = \frac{Lc}{G}\,\Delta t = \frac{36{,}000}{30{,}000} \times 2 = 2.4 \text{ Btu per pound.}$$

The result of the integration is that the Number of Transfer Units required (NTU) = 0.757.

Unit gas mass velocity, $\frac{G}{S} = \frac{30{,}000}{6 \times 6} = 830$ lb per hour per square foot.

From Fig. 6, Ka = 138 lb per hour per cubic foot.

The tower volume required is:

$$V = \frac{G}{Ka}\,(NTU) = \frac{30{,}000}{138} \times 0.757 = 217 \times 0.757 = 164 \text{ cu ft.}$$

Height of the packed section is:

$$\frac{164}{6 \times 6} = 4.6 \text{ ft.}$$

The graphical solution for the Number of Transfer Units required for the desired performance is plotted in Fig. 7. $\frac{1}{h'' - h_a}$ is plotted as a function of h, and the area under the curve from the initial enthalpy of the air, 25 Btu per pound, to the final enthalpy of the air, 77.8 Btu per pound is 0.757, the Number of Transfer Units required. The effect of the rapid decrease in potential due to the cooling of the water is indicated by comparison of area A_1, A_2, and A_3 of Fig. 7. Each represents the Number of Transfer Units required to achieve a water temperature reduction of about 15 F.

$$A_1 = 0.132\ NTU \text{ (144 to 130 F)}$$
$$A_2 = 0.251\ NTU \text{ (130 to 115 F)}$$
$$A_3 = 0.369\ NTU \text{ (115 to 100 F)}$$

TABLE 4. INTEGRATION OF NUMBER OF TRANSFER UNITS

t	h_a	h''	$h'' - h_a$	$\frac{\Delta h}{h'' - h_a}$
100	25.0	71.4	46.4	0.052
102	27.4	75.0	47.6	0.050
104	29.8	78.9	49.1	0.049
106	32.2	83.0	50.8	0.047
108	34.6	87.3	52.7	0.045
110	37.0	91.8	54.8	0.044
112	39.4	96.7	57.3	0.042
114	41.8	101.7	59.9	0.040
116	44.2	107.1	62.9	0.038
118	46.6	112.8	66.2	0.036
120	49.0	118.9	69.9	0.034
122	51.4	125.2	73.8	0.032
124	53.8	132.1	78.3	0.030
126	56.2	139.2	83.0	0.029
128	58.6	146.8	88.2	0.027
130	61.0	154.9	93.9	0.025
132	63.4	163.7	100.3	0.023
134	65.8	172.9	107.1	0.022
136	68.2	182.6	114.4	0.021
138	70.6	193.1	122.5	0.020
140	73.0	204.3	131.3	0.018
142	75.4	216.3	140.9	0.017
144	77.8	229.0	151.2	0.016
				0.757

When the relationship between the enthalpy of the saturated air and the temperature is linear over the range of water temperatures involved, it can be shown[7] that the logarithmic mean of the terminal potentials, Δh_{lm}, is the correct driving force. This is true to a good approximation when the water cooling does not exceed 15 F. The approximation to linearity may be determined by inspection of Table 6, Chapter 1, or the temperature enthalpy diagram of Fig. 5. When the logarithmic mean is a valid potential, Equation 6 may be written:

$$\frac{h_1 - h_2}{\Delta h_{lm}} = \frac{KaV}{G} \tag{7}$$

and the need for the numerical integration for the determination of the tower volume is eliminated.

The application of the foregoing design method to atmospheric towers is difficult because rate coefficients and flow conditions are not yet well

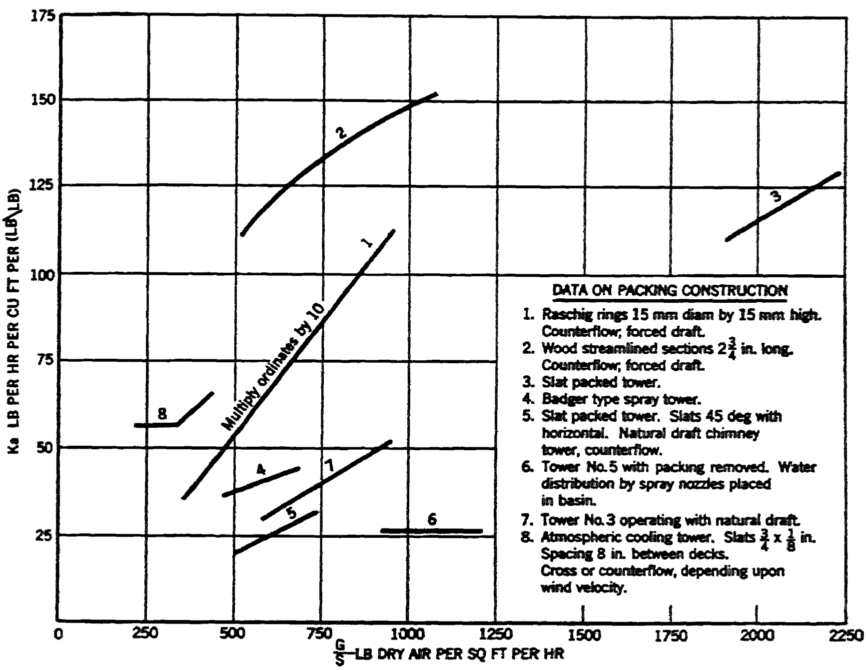


FIG. 6. UNIT CONDUCTANCES FOR VARIOUS TYPES OF PACKING CONSTRUCTION

defined for such equipment. If these are known, application of Equation 7 to sections of the tower small enough to justify use of the logarithmic mean potential will yield the tower volume required for each section. The sections must be taken perpendicular to the path of water flow. A correction to adjust the logarithmic mean potential, evaluated as for counter-flow, to the reduced effectiveness of cross-flow, has been derived for heat transfer and may be applied to this case[8].

Atmospheric towers operate with natural draft, produced in a vertical direction by the stack action of the tower structure, at zero velocity of the

[7]Loc. Cit. Note 3, p. 79.

[8]Heat Transmission, by W. H. McAdams (McGraw-Hill Co., New York City, 1933, p. 157).

approach wind. Approach wind of sufficient magnitude (the magnitude depending on the baffle arrangement which is designed to reduce drift) will cause cross-flow augmenting the natural draft. An adequate design requires the consideration of both flow conditions.

Expression for Cooling Tower Performance

The performance of a cooling tower is described in terms of its effectiveness as an energy exchanger. The effectiveness is defined as the ratio of the energy actually exchanged to the energy available for exchange.

Effectiveness expressions:

Case 1. The slope of the operating line on the $t - h$ diagram exceeds the slope of the saturation line in the region of water temperatures considered.

$$e = \frac{h_1 - h_2}{h''_1 - h_2}$$

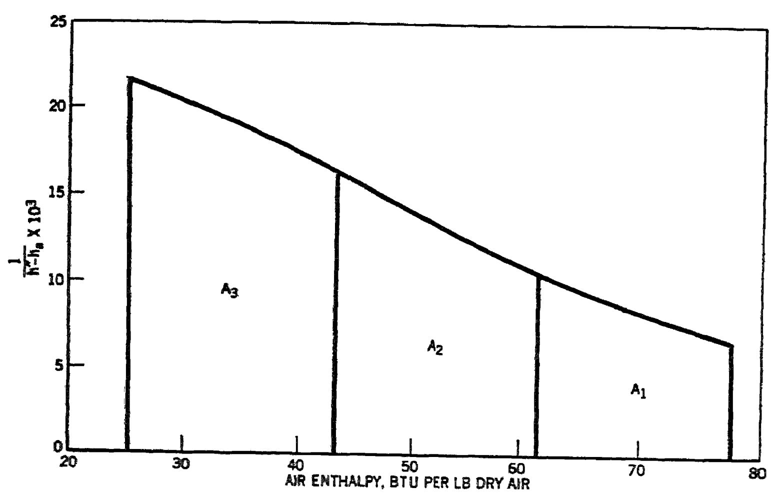


FIG. 7. GRAPHICAL INTEGRATION TO DETERMINE NUMBER OF TRANSFER UNITS REQUIRED FOR DESIRED OPERATING CONDITIONS OF EXAMPLE 1

Case 2. The slope of the saturation line exceeds that of the operating line.

$$e = \frac{h_1 - h_2}{\frac{Lc}{G}(t_1 - t_{wb})} = \frac{t_1 - t_2}{t_1 - t_{wb}}$$

This equation represents the *approach to wet-bulb.*

Usual tower operating conditions conform to Case 1. Because of the curvature of the saturation line, operating conditions may present themselves to which neither Case 1 nor 2 applies. Since a simple expression for the intermediate case is not available, the expression of Case 1 may be utilized for the small number of operating conditions falling into the intermediate classification.

When the logarithmic mean enthalpy difference is a sufficiently accurate

overall potential the effectiveness as defined may be expressed analytically as follows:

$$e = \frac{h_1 - h_2}{h_1'' - h_2} \text{ or } \frac{t_1 - t_2}{t_1 - t_{wb}} = \frac{1 - \varepsilon^{NTU_{\min}(1 - r_f)}}{r_f - \varepsilon^{NTU_{\min}(1 - r_f)}} \tag{8}$$

where $NTU_{\min}$ is $\frac{KaV}{G}$ for Case 1

or where $NTU_{\min}$ is $\frac{KaV}{Lc}$ (m) for Case 2

and r_f is the ratio, $\frac{Lc}{mG}$ or $\frac{mG}{Lc}$, taken so that it is always less than unity.

The average slope of the saturation line is, to a good approximation, the slope of the chord of the saturation line between the entrance and exit water temperatures. Solution of Equation 8 offers a method for determining the performance of a given tower when the number of transfer units, gas, and water rates are known and the slope of the saturation line does not vary greatly in the temperature range involved. If these same data are available, a trial and error solution of Equation 7 will also yield the tower performance.

Make-Up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the remainder, there is a continual drain on the quantity of water in the system, and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 F range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Because the evaporation of the water leaves behind the salts which the water contained, high concentration of salts may make chemical treatment of the make-up water necessary to

TABLE 5. COMPARISON OF VARIOUS TYPES OF ATMOSPHERIC WATER COOLING EQUIPMENT

Figures indicate order of desirability

	Cooling Pond	Spray Pond	Spray Tower	Deck Tower	Mechanical Draft	Indoor Tower
Cost	x	2	1	3	4	5
Area	5	4	3	2	1	x
Height	1	2	3	4–5	4–5	x
Weight per square foot	x	x	1	3	4	2
Independence of wind velocity	6	3	4	5	1–2	1–2
Drift nuisance	1	6	5	4	2–3	2–3
Make-up water required	1	6	5	4	2–3	2–3
Pumping head	1	2	3	4–5	4–5	6
Maintenance	2	1	3	4	5	6
Suitability for congested districts	x	5	4	3	1	2
Water quantity required for definite result	6	5	4	1–2	1–2	3

xNot comparable.

avoid excessive deposits in the condensers. An additional amount of make-up water must be added to replace windage, or *drift loss*. This additional amount of water varies from 0.1 to 3 per cent of the quantity of water being circulated, this percentage depending upon the type of equipment and the wind velocity.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled below freezing temperature so ice forms and collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

A comparison of various types of water cooling equipment is given in Table 5.

Chapter 27

AIR POLLUTION

Classification of Air Impurities, Dust Concentrations, Air Pollution and Health, Occlusion of Solar Radiation, Smoke and Air Pollution Abatement, Dust and Cinders, Nature's Dust Catcher

THE particulate impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silks, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon.

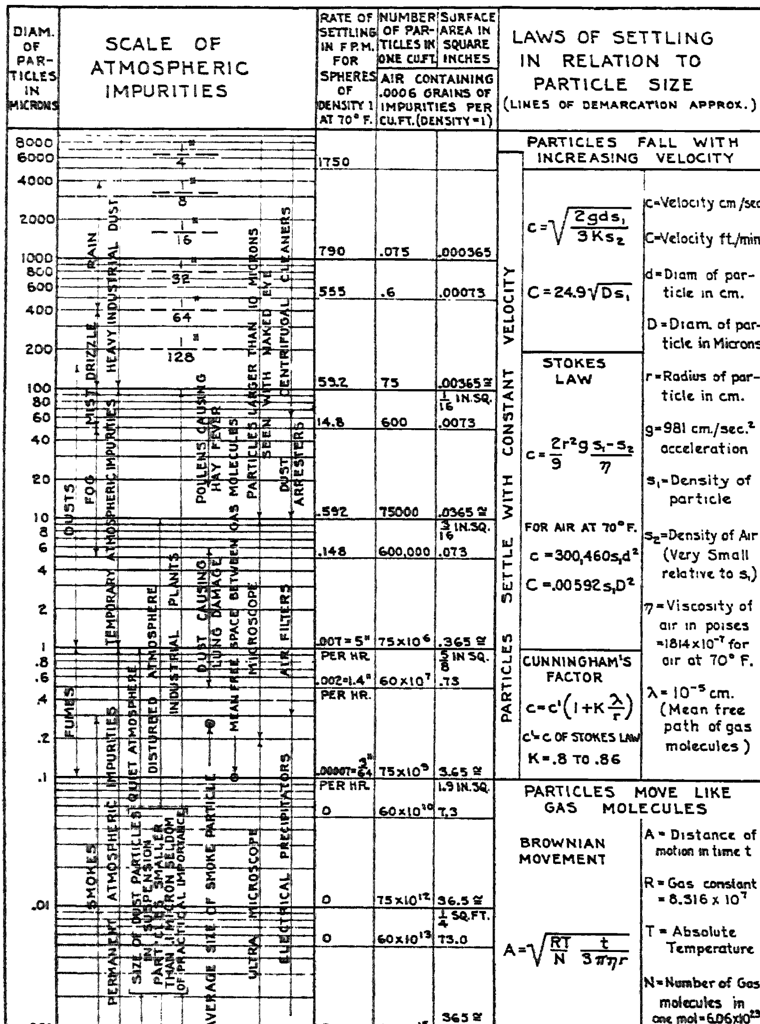
CLASSIFICATION OF AIR IMPURITIES

The most conspicuous sources of atmospheric pollution may be classified in various ways, as dusts, fumes and smoke. In Fig. 1, the classification is by particle size, but recent practice favors differentiation by method of formation. Thus, *dusts* are composed of particles produced by disintegration of larger material, as by crushing or grinding, whereas *fumes* are produced by condensation, and *smoke* consists of the finer carbon particles resulting from incomplete combustion. Similarly, *mists* are formed by the breaking up of liquids and *fogs* by condensation of vapors. There is as yet, however, no general agreement on these terms.

Dusts tend to settle without agglomeration, fumes to aggregate and smoke to diffuse. Particles which approach the common bacteria in size—from 1 to 10 microns—are difficult to remove from air and are apt to remain in suspension unless they can be agglomerated by artificial means. The term fly-ash is usually applied to the microscopic glassy spheres which form the principal solid constituent of the effluent gases from powdered-coal fired furnaces. Cinders denote the larger solid constituents which may be entrained by furnace gases.

Particles larger than 10 microns are unlikely to remain suspended in air currents of moderate strength. Only violent air motion will sustain them in air long enough for them to be breathed. This means that, in hygienic problems, the engineer is concerned mostly with suspensions of particles comparable to the common bacteria in size. A notable exception to this size limitation is the common hay-fever producing pollen such as that from rag-weed. Pollen grains may be anything from fragments 15 microns in diameter to whole pollens 25 microns or more in size.

Although it is possible under certain conditions to recognize the presence of particles smaller than 10 microns, the normal eye is unable to resolve any dimension under about 50 microns, and thus all air floated material of this kind is too small to identify without the aid of a microscope.



Compiled by W. G. Frank and Copyrighted.

FIG. 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

Mineral particles, such as grains of sand, bits of rock, volcanic ash, or fly-ash, can be transported long distances under unusual circumstances. Thus, the dust storms of 1935 in the Kansas district resulted in vast amounts of fine top soil being thrown high into the air. Solar illumination

TABLE 1. APPROXIMATE LIMITS OF INFLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a]

Gas or Vapor	Lower Limit Volume in Per Cent	Higher Limit Volume in Per Cent
Hydrogen	4.1	74.0
Ammonia	16.0	27.0
Hydrogen sulphide	4.3	46.0
Carbon disulphide	1.0	50.0
Carbon monoxide	12.5	74.0
Methane	5.3	14.0
Methane (turbulent mixture)	5.0	15.0
Ethane	3.2	12.5
Propane	2.4	9.5
Butane	1.9	8.5
Pentane	1.45	7.5
Ethylene	3.0	29.0
Acetylene	3.0	....
Acetylene (turbulent mixture)	2.3	....
Benzene	1.4	7.0
Toluene	1.4	7.0
Cyclohexane	1.3	8.3
Methyl cyclohexane	1.2	....
Methyl alcohol	7.0	....
Ethyl alcohol	4.0	19.0
Ethyl ether	1.7	26.0
Benzine	1.1	....
Gasoline	1.4	6.0
Water gas	6 to 9	55 to 70
Ethylene oxide	3.0	80.0
Acetaldehyde	4.0	57.0
Furfural (125 C)	2.0	....
Acetone	3.0	11.0
Acetone (turbulent mixture)	2.5	....
Methyl ethyl ketone	2.0	12.0
Methyl formate	6.0	20.0
Ethyl formate	3.5	16.5
Methyl acetate	4.1	14.0
Ethyl acetate	2.5	11.5
Propyl acetate	2.0	....
Butyl acetate (30 C)	1.7	....
Ethyl nitrite	3.0	....
Methyl chloride	8.0	19.0
Methyl bromide	13.5	14.5
Ethyl chloride	4.0	15.0
Ethyl bromide	7.0	11.0
Ethylene dichloride	6.0	16.0
Dichlorethylene	10.0	13.0
Vinyl chloride	4.0	22.0
Pyridine (70 C)	1.8	12.5
Natural gas	4.8	13.5
Illuminating gas	5.3	31.0
Blast-furnace gas	35.0	74.0

[a]Limits of Inflammability of Gases and Vapors, by H. F. Coward and G. W. Jones, (*U. S. Bureau of Mines, Bulletin* No. 279, 1931).

as far east as Boston was affected noticeably and particles as large as 40 to 50 microns were actually carried half way across the continent before they settled out. In similar manner volcanic ash has been carried even further. It is not surprising, therefore, that fly-ash from furnace gases, cement dust and the like, can be carried for considerable distances and occasionally the engineer is confronted with the problem of removing such material before the air in question is suitable for use in building ventilation.

The physical properties of the particulate impurities of air are summarized conveniently in the chart of Fig. 1.

In the case of gases, the objectionable features are the injurious physiological effects and the danger from inflammability. See Table 1.

Dust Concentrations

It is customary to report dust concentrations as grains per 1000 cu ft or milligrams per cubic meter. Gas concentrations are commonly recorded as milligrams per cubic meter or as parts per million or as per cent by volume. Typical ranges in dust concentrations as now found in practical applications are given in Table 2.

TABLE 2. DUST CONCENTRATION RANGES IN PRACTICAL APPLICATIONS[a]

Application	Grains Per 1000 Cu Ft	Mgs Per Cu M
Rural and suburban districts	0.2 to 0.4	0.4 to 0.8
Metropolitan districts	0.4 to 0.8	0.9 to 1.8
Industrial districts	0.8 to 1.5	1.8 to 3.5
Dusty factories or mines	4.0 to 80.0	10 to 200
Explosive concentrations (as of flour or soft coal)	4000 to 8000	10,000 to 20,000

[a]1 gr per 1000 cu ft = 2.3 mgs per cubic meter; 1 oz per cubic foot = 1 g per liter.

The engineer frequently desires information regarding the effects of various concentrations of gases or dusts upon man, as the success of a particular installation may depend upon the maintenance of air which is adequately clean. At the present time there are several organizations working on this problem all of them publishing literature of various kinds.[1] References to books covering the hygienic significance, determination and control of dust are listed at the end of this chapter.

AIR POLLUTION AND HEALTH

The prevention of various diseases which result from exposure to atmospheric impurities is an engineering problem. It is important for the engineer to insure, by proper ventilation, suitable environments for working or for general living. If the equipment used is to be successful, it must operate automatically as in the modern air conditioned theater or railroad train.

In Table 3 are given data on permissible concentrations of various substances, gases and dusts, which occur in industry. The prudent

[1]National Institute for Health, U. S. Public Health Service; Division of Labor Standards, U. S. Department of Labor; University of Toronto Medical School, Canada; Saranac Laboratories, Saranac Lake, N. Y.; Air Hygiene Foundation, Inc., Pittsburgh, Pa.; Harvard School of Public Health, Boston, Mass.; Haskell Laboratory, Wilmington, Del.; and the Departments of Health and of Labor in the United States and in various provinces of Canada.

engineer will design equipment using these bench marks as the upper limits of pollution. In general it is good practice to avoid recirculation of air which contains originally toxic substances. Obviously there may be exceptions to this rule, but it is one which is generally being followed in current practice.

Bronchitis is the chief condition associated with exposure to thick dust, and follows upon inhalation of practically any kind of insoluble and non-colloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it may aggravate the disease once it has started.[2]

TABLE 3. TOXICITY OF GASES AND FUMES IN PARTS PER 10,000 PARTS OF AIR[a]

Vapor or Gas	Rapidly Fatal	Maximum Concentration for from ½ to 1 Hour	Maximum Concentration for 1 Hour	Maximum Allowable for Prolonged Exposure
Carbon monoxide	40	15–20	10	1
Carbon dioxide	800–1000	......	......	......
Hydrocyanic acid	30	1½	½	⅕
Ammonia	50–100	25	3	1
Hydrochloric acid gas	10–20	½	......	1/10
Chlorine	10	½	......	1/100
Hydrofluoric acid gas	2	1/10	......	1/33
Sulphur dioxide	4–5	½–1	......	1/10
Hydrogen sulphide	10–30	5–7	2–3	1
Carbon bisulphide	......	11	5	½
Phosphene	20	4–6	1–2	......
Arsine	2½	½	½	......
Phosgene	Over ¼	¼	......	1/100
Nitrous fumes	2½–7½	1–1½	......	⅓
Benzene	190	......	31–47	1½–3
Toluene and xylene	190	......	31–47	......
Aniline	......	......	1–1½	1/10
Nitrobenzene	......	......	1/100	1/500
Carbon tetrachloride	480	240	40	1
Chloroform	250	140	50	2
Tetrachlorethane	73	......	......	1/10
Trichlorethylene	370	......	......	......
Methyl chloride	1500–3000	200–400	70	5–10
Methyl bromide	200–400	20–40	10	2
Lead dust	......	......	......	0.15 mg/cu m
Quartz dust	......	......	......	1 mg/cu m

[a]Adapted from Y. Henderson and H. Haggard. (See *Noxious Gases*, *1927*, and *Lessons Learned from Industrial Gases and Fumes*, *Institute of Chemistry of Great Britain and Ireland*, London, 1930.)

The sulphurous fumes and tarry matter in smoke are more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages. The Meuse Valley fog disaster will probably become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons.

Carbon monoxide from automobiles and from chimney gases constitutes another important source of aerial pollution in busy cities. During

[2]Physiological Response of the Peritoneal Tissue to Dusts Introduced as Foreign Bodies, by Miller and Sayers (*U. S. Public Health Reports*, 49:80, 1934).

heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets is found to contain enough *CO* to menace the health of those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of *CO* in city air is insufficient to affect the average city dweller or pedestrian.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[3] by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City[4] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

Recent studies[5] in Pittsburgh indicate that heavy smoke pollution is definitely unhealthful. Heretofore adequate proofs on this point were lacking.

The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution are justified.

SMOKE AND AIR POLLUTION ABATEMENT

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of buildings that operate their own boiler plants.

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals. (See Chapter 7.)

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, pro-

[3]Effects of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H Coblentz and F. A. Korff (*American Journal of Public Health*, p. 7, Vol. 19, 1929).

[4]Studies in Illumination, by J. E. Ives (*U. S. Public Health Service Bulletin* No. 197, 1930).

[5]Pneumoconiosis in the Pittsburgh district, Based on a Study of 2,500 Post Mortem Examinations made in Pittsburgh Hospitals, by Schnurer et al (*Journal Industrial Hygiene*, 17:294, March, 1935).

duces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher gravity oil always produces less smoke than a high volatile coal or low gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is necessary.

Legislative measures at the present time are largely concerned with the smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 34), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious gases, such as sulphur dioxide, nor sulphuric acid fog, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDERS

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the

market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in Chapter 28.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coals when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

NATURE'S DUST CATCHER

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for each rain drop and the rain picks up dust as it falls from the clouds to the earth. However, it was found in recent studies[6] that rain was not a good air cleaner of the material below about 0.7 micron.

REFERENCES

Bulletin, Air Hygiene Foundation, Inc., Pittsburgh, Pa.

Determination and Control of Industrial Dust, by J. J. Bloomfield and J. M. Dalla Valle (*U. S. Public Health Bulletin*, No. 217, 1935).

Journal of Industrial Hygiene and Toxicology, Harvard School of Public Health, Boston, Mass.

Reports of the National Silicosis Conference, Washington, D. C. To be published by the U. S. Department of Labor.

Saranac Symposium on Silicosis, 1937, Saranac Laboratories, Saranac, N. Y.

Industrial Dust, by Philip Drinker and Theodore Hatch, McGraw Hill Co., N. Y.

Noxious Gases, by Y. Henderson and H. Haggard, Chemical Catalog Co., N. Y.

Occupation and Health, International Labour Office.

Preventive Medicine and Hygiene, by Milton J. Rosenau, D. Appleton-Century Co., N. Y.

[6]Atmospheric Pollution of American Cities for the years 1931-1933, by J. E. Ives et al (*U. S. Public Health Bulletin* No. 224, March, 1936).

Chapter 28

AIR CLEANING DEVICES

Damage Caused by Dust, Classification of Air Cleaning Devices, Viscous-Impingement Filters, Dry Air Filters, Air Washers, Electrical Precipitators, Cleaning of Gases from Exhaust Systems

IN this chapter the term *cleaning* is assumed to mean the removal of particulate matter from the air. The removal of foreign gases and vapors requires entirely different methods and is discussed in Chapter 39.

The cleaning of air involves the removal of many kinds of materials having a wide range of particle sizes and concentrations. The degree of air purification required varies widely, consequently, many types of devices having radically different design characteristics are available.

The various materials that pollute the air are discussed in Chapter 27, Fig. 1, which shows characteristics of particles ranging in size from 8000 to 0.001 microns. The importance of particles in the range from 0.1 to 0.001 microns is open to argument. Particles below 0.1 micron can be seen in some microscopes as specks of reflected light, and a few microscopes using ultra-violet light have a resolving power of 0.1 micron, but the smallest particle which is really resolved in microscopes using ordinary light is about 0.25 micron. The performance of particles below 0.1 micron is, therefore, controversial because no means have been developed for reliably counting or measuring the sizes of the particles.

Even if the discussion is limited to the range from 0.1 micron to 50 microns, from the smallest particle observable in the microscope to the smallest particle distinguishable to the naked eye, this range is so far outside the usual experience that it is difficult to visualize. If particles could be examined through a super microscope having a magnification of 250,000 diameters, a tobacco smoke particle of 0.1 micron would appear to be 1 in. in diameter, or approximately the size of a golf ball; a soft coal smoke particle 0.3 micron in diameter would appear like a baseball; a ragweed pollen grain 20 microns in diameter would appear 16.5 ft in diameter, while the 50 micron particle, just visible to the naked eye and able to pass through a 270 mesh screen, would appear to be 50 ft in diameter. Picturing this range in particle size from a golf ball to a sphere 50 ft in diameter may aid in appreciating the problem of cleaning air and the difficulty of devising any single test to adequately measure the performance of air cleaning devices under all conditions of service.

DAMAGE CAUSED BY DUST

Dust may cause damage in many ways, but usually it must first lodge on a surface. Larger particles settle rapidly out onto surfaces. The rate of fall of particles is given in Chapter 27, Fig. 1. Those visible to the unaided eye (50 microns or over) fall so rapidly that few remain in the air. However, any large ones which are carried into a room are almost certain to settle and are so noticeable when they have settled as to be very objectionable.

Any air movement, particularly over fabrics or unpolished surfaces, tends to deposit dust on the surface. The smallest particles observable in the microscope are deposited in this way.

A phenomenon of great importance in air conditioning and one not yet generally appreciated is thermal precipitation of dust. This is the tendency for dust to be deposited on any surface which is cooler than the adjacent air. It is largely responsible for outside walls becoming dirtier than partitions. In the case of plaster on wood lath, the dark streaks following the spaces between laths are very noticeable and frequently beams and other structural members can be traced by the difference in blackness of the wall or ceiling. Thermal precipitation deposits particles of all sizes apparently with very little differentiation as to size.

By keeping the surface temperature higher and more uniform, modern thermal insulation decreases the deposit of dust and makes it more uniform and less noticeable.

REQUIREMENTS

The removal of larger particles is always important because they quickly settle out of the air and become noticeable on surfaces as visible dustiness. In many applications the removal of these larger particles will constitute satisfactory performance, and a relatively simple device can be used.

In cities where large quantities of soft coal are burned, the air becomes contaminated with fine black smoke particles. These are deposited on walls, draperies, and other interior surfaces by air movement and by thermal precipitation. Their removal is much more difficult than that of the large particles.

Hay fever is usually due to pollen in the air, and many people, once this trouble has started, are sensitive to minute amounts of pollen, so that almost perfect cleaning may be required. Asthma many be caused by many things, including pollens and fine dusts. Many afflicted with either disease, who have not obtained relief from usual remedies, have found it in a room where the air is efficiently cleaned from small as well as large particles.

In air conditioning systems the maintenance of constant air flow is essential. In summer cooling, a decrease in air flow may cause a reduction in temperature of discharged air. This, combined with reduced velocity, may completely upset the air distribution objective resulting in drafts. The air cleaning equipment must function so that reduction in air flow due to the normal accumulation of dust will not cause faulty operation of the system.

In addition to requirements which are specified the following features are desirable:

1. Low resistance to air flow.
2. Ease of cleaning and maintenance.
3. Efficiency over a wide range of velocities.
4. Binding liquid to catch particles must not contaminate the air.

TESTING

The wide variety of materials and particle sizes which may be present in the air makes the testing of air cleaning devices difficult. Probably no standardized test can cover all of the conditions which may be encountered in service.

Tests have been devised which compare the ability of air cleaners to remove a certain artificial dust under specified conditions. This seems to be the most practical way of comparing various devices, but it may lead to misleading results if the dust is not representative of the dust to be removed in service.

The most common test is that specified by the A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work[1]. This Code specifies an artificial dust consisting of a mixture of dusts, which has passed through a 200 mesh screen. The efficiency is measured in terms of the ratio of the weight of dust removed to the weight of dust injected into the air. This is a well-worked-out method for testing the ability of a device to remove the coarser particles, but it is not a satisfactory measure of the ability of a device to remove extremely fine particles. In general, any weight method tends to measure the efficiency of removal of the larger particles present in the test dust. The largest particle specified in this Code is one just passing a 200 mesh screen or one 70 microns in diameter. Assuming a dust containing one such particle to 1,000,000 particles having a diameter of 0.1 micron, the 70 micron particle will have 3.43×10^8 times the weight of one of the 0.1 micron particles, all particles having the same density. If the cleaning device removes the one large particle but removes none of the 0.1 micron particles, the weight efficiency will be the ratio of the weight removed to the weight of dust injected or 99.71 per cent by weight even though only one particle out of 1,000,000 has been removed. This is obviously an exaggerated case, but illustrates the weakness of a weight test in measuring efficiency of removal of fine particles.

In testing air filters at the *National Bureau of Standards*, a much finer dust is used and the efficiency is measured by determining the relative blackness of pieces of filter paper through which air is passed[2]. The test dust used is a sample of dust collected by a precipitator in a local power plant, which of course is not as fine as atmospheric dust. However, this method of measuring efficiency can be used with atmospheric dust to test an air cleaning device under actual operating conditions. The efficiency is determined by drawing samples of filtered and unfiltered air through pieces of filter paper, the volumes of air being adjusted until

[1]A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

[2]A Test Method for Air Filters, by Richard S. Dill (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 379).

equal blackness is obtained. If, for example, one unit of volume of unfiltered air gives the same blackness as four units of volume of filtered air, the efficiency is said to be 75 per cent. This method gives a much better measure of the tendency to blacken walls than the weight efficiency.

CLASSIFICATION OF AIR CLEANING DEVICES

Considering the wide variety of materials and particle sizes to be removed and the variety of requirements, it is natural that many kinds of devices are used which cannot be shown satisfactorily in a single simple classification. The following outline gives classifications on three different bases:

1. Methods of cleaning.
   a. Automatic.
   b. Non-automatic.
      (1) Throw-away, replaceable elements.
      (2) Manually cleaned in place.
      (3) Removable for cleaning.
2. Principle of air cleaning.
   a. Viscous-impingement filters.
   b. Dry filters.
   c. Washers.
   d. Centrifugal devices.
   e. Electrical precipitators.
3. Classification according to application.
   a. General air conditioning.
      (1) Central cleaning system.
      (2) Unit ventilator.
      (3) Window installation.
      (4) Warm air furnace.
   b. Removal of smoke and fumes from stack gases.
   c. Collection of dusts from exhaust systems.

VISCOUS IMPINGEMENT TYPE FILTERS

The principle of air cleaning used in viscous filters is that of *adhesive impingement*. Dust and dirt in the air, especially soot and carbons, are trapped and retained by successive impingements on coated surfaces. The arrangements of filtering mediums and the kind of materials used are almost unlimited. Since this type of device depends on impingement, it is more effective in catching large particles than small ones. To secure maximum cleaning efficiency, it is necessary to divide the air into innumerable fine streams, to obtain intimate contact between the air and the viscous-coated mediums.

The following are desirable characteristics of a binding liquid:

1. Its surface tension should be such as to produce a homogeneous film-like coating on the filter medium.

2. The viscosity should vary only slightly with normal changes of temperature.

3. It should prevent the development of mold spores and bacteria on the filter mediums.

4. The liquid should have high capillarity, or ability to wet and retain the dust at all operating temperatures.
5. Evaporation should be low.
6. It should be fire resistant.
7. It should be odorless.

Viscous Type Unit Filters

In the unit type viscous filter, the filtering mediums are arranged in units of convenient size to facilitate installation, maintenance, and cleaning. Each unit consists of an interchangeable cell or replaceable filter pad and a substantial frame which may be bolted to the frames of other like units to form a partition between the source of dusty air and the fan inlet.

To secure the maximum dust-holding capacity, the filter medium is usually arranged with large pores or air passages on the entering air side, the filter density increasing gradually toward the leaving air side. This arrangement provides relatively large spaces for the collection of dirt in

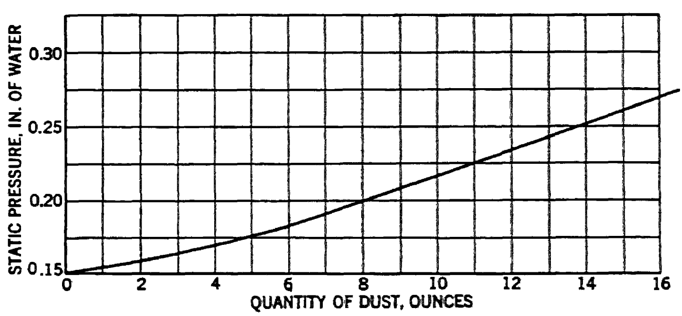


FIG. 1. CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION

the front of the filter where the bulk of the dust is taken out without undue increase in resistance, while in the back of the filter the openings are smaller to secure high efficiency in the removal of the finer dust particles.

The resistance of a well-designed unit filter of the adhesive impingement type usually depends upon the velocity at which the air is handled and upon whether the unit is clean or dirty. The cleaning efficiency is usually highest after it has accumulated a certain portion of its maximum load of dirt because some dust collected in the cell acts as an efficient medium for the further seizing of solids from the air. By periodically cleaning a predetermined number of cells, the resistance and capacity of a built-up filter may be held at any desired figure. The frequency of cleaning any installation depends upon the dust concentration of air being cleaned, and on the amount of dirt which can be accumulated in the medium without causing excessive resistance. (Figs. 1, 2 and 3.)

The dust-holding capacity of a given filter is dependent on the type of dust. Lint is particularly difficult to collect because it tends to build up a layer over the face of the filter. In this way a small quantity of dust may cause a serious increase in the pressure drop of a filter.

A chart showing the increase in resistance of a unit filter of the viscous impingement type, when tested with the standard test dust described in

the Code, is given in Fig. 1. The resistance to air flow of three typical clean viscous impingement type filters having different densities of mediums is shown in Fig. 2. Type A is a dense pack used in bacterium control; Type B is a medium pack used for general ventilation work, and Type C is a low resistance unit for use where low resistance is the important factor and maximum cleaning efficiencies are not essential. The operating characteristics which might be expected under various dust concentrations with air filters having different dust-holding capacities are illustrated in Fig. 3.

Filters consisting of inexpensive frames of cardboard or similar material filled with viscous-coated glass wool, steel wool or the like are available. Because of their construction these units may be discarded when dirty and replaced with new units at relatively little expense. They are used in general ventilation work and with warm-air furnaces and other instal-

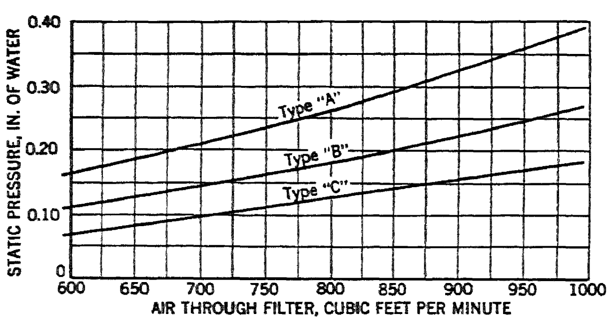


Fig. 2. Resistance to Air Flow of Typical Unit Air Filters

lations where low first cost and low resistance to air flow are essential. The operating characteristics of these units conform in general with those of the rigid frame type.

Viscous Automatic Filters

In this type of filter, the removal of the accumulated dust is done automatically instead of by hand. The automatic cleaning and recoating of these filters is based on the principle that the viscous fluid itself will perform the cleaning function, thereby eliminating a separate washing agent. The dust collected by the filter is deposited finally in the bottom of the viscous fluid reservoir from which it may be removed by different methods, depending on the design of the filter.

There are three general types of automatic filters. They are differentiated from each other according to the process of self-cleaning and renewing of the viscous coating used as follows:

1. The filter medium has the form of an endless curtain suspended vertically, with its lower portion submerged in a viscous fluid reservoir. The curtain moves slowly through this bath, thus performing the cleaning and recoating of the filter medium.

2. The filter screen is arranged in the form of shelves or cylinders, and the viscous fluid is flushed through all parts of the medium in a direction opposite to the air flow.

3. The filter medium is arranged vertically and is stationary. The viscous fluid is flushed from above over the medium, while the air flow is stopped.

The washing and renewing process in automatic filters of the second and third types is usually intermittent. It is accomplished by an electric motor or by other motive power and is controlled by manual or by automatic timing devices. The operating cycle is of a predetermined frequency and should be so timed as to insure a constant static pressure drop across the filter. The customary resistance to air flow is 3/8-in. water gage at an air velocity of 500 fpm, measured at the filter entrance. Automatic viscous filters are made up in units which are delivered either fully assembled or in parts to be assembled at the point of installation.

DRY AIR FILTERS

Dry air filters, in which dust is impinged upon or trapped in screens made of felt, cloth, cellulose, or other fabrics, are available in various types. These filters require no adhesive liquid, but depend on the strain-

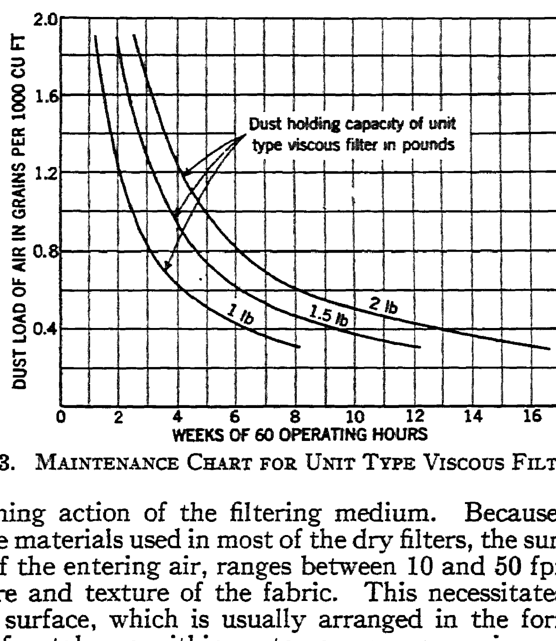


FIG. 3. MAINTENANCE CHART FOR UNIT TYPE VISCOUS FILTERS

ing or screening action of the filtering medium. Because of the close texture of the materials used in most of the dry filters, the surface velocity, or velocity of the entering air, ranges between 10 and 50 fpm, depending on the nature and texture of the fabric. This necessitates a relatively large screen surface, which is usually arranged in the form of pockets to bring the frontal area within customary space requirements.

As with viscous unit filters, an average constant resistance and air volume may be obtained by periodic reconditioning or renewal of the filter screens. Since some materials suitable for dry filtering mediums are affected considerably by moisture which tends to cause a rapid increase in resistance, they should be treated or processed to minimize the effect of changes in humidity.

Filters using felt and similar materials usually depend upon vacuum cleaning for reconditioning. A special nozzle, operated from a portable or stationary vacuum cleaner, is shaped to reach all parts of the filter pockets. Permanent filter mediums should be capable of withstanding repeated vacuum cleanings without loss in dust removal efficiency. While

most dry filters are cleaned by replacing an inexpensive filter sheet, the useful life of these sheets often may be lengthened by vibrating or vacuum cleaning.

AIR WASHERS

Air washers, originally designed as the name implies to wash air, are now used for humidification (see Chapter 26). Their ability to cleanse air depends upon the nature of the dust; fine particles, especially those having no affinity for water, are not efficiently removed.

ELECTRICAL PRECIPITATORS

Electrostatic precipitation has long been used for the precipitation of smokes and fumes from smoke stacks, but until recently such equipment was unsuited to air conditioning and ventilation applications. The operating principles are as follows: A discharge takes place from a small wire to grounded electrodes and as the air passes through this discharge the dust particles receive an electrical charge. The air then passes between parallel plates with a high potential difference between plates. The resulting electric field attracts the dust particles to one set of plates.

The electric force is effective in depositing the particle on the plate, but does not hold it there. Some dusts stick to the plates, but in general there should be a coating of adhesive. The dust is removed by washing. As this device is sensitive to air velocities, care should be taken to avoid any local currents that exceed rated velocity. An advantage of electrostatic precipitation is the ability to remove fine particles at high efficiency and with a low pressure drop, and thus with little change in the volume of air delivered. This feature must be balanced against the greater first cost.

CLEANING OF GASES FROM EXHAUST SYSTEMS

The gases from exhaust systems vary widely, but in general contain heavy concentrations of dust as compared to ventilation practice. A dust loading of one grain per cubic foot of air is not unusual, which is about 1000 times the dirt usually found in ventilating air. For this reason even though the cleaning efficiency is high it is still usually desirable to discharge the exhaust out of doors. The purpose of the cleaning is then to prevent a nuisance in the neighborhood or to collect valuable material.

Gravitational Settling Chambers

Settling chambers are simple but effective only in removing large particles. The rate of fall of various particles is given in Fig. 1 of Chapter 27. Even though the vertical height is kept small to reduce the time of fall, limitations of space and volume of air to be handled usually allow time for only the larger particles to settle, even if the flow is perfectly streamlined. Actually, eddies retard the settling of particles so that the full velocity of fall is not realized.

Cyclones or Centrifugal Separators

The force causing settling can be increased many times that of gravitation by giving the air a whirling motion and introducing centrifugal

force. The settling rate then becomes dependent upon the peripheral air velocity and the radius of curvature as well as upon the other factors.

In cyclone separators, air is introduced tangentially into a vertical cylinder and passes out from the center of the top. The gas velocity and curvature of the cylinder cause whirling which throws the particles to the surface. The particles slide down this surface and are removed through a hopper in the cone bottom, or are thrown through slits in the periphery.

Assumptions regarding streamline flow and turbulence make general calculations of centrifugal settling rate quite involved and rough. Centrifugal separators have wide application in connection with industrial operations such as grinding, screening and combustion, but have little or no effect upon the finer particles.

When a high collection efficiency is desired, or the material is unusually fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation.

Cloth Filters

Filters are used when the material collected by an exhaust system is valuable or cannot be separated efficiently from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. Bag houses used in manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The collected dust particles themselves aid in agglomerating and retaining others. Periodic shaking with the fans off or reversed, at intervals of a few hours, drops the excess dust into a lower header or hopper for removal.

Readily removed filters built in small sections in which filter mediums can be replaced are of distinct advantage where deterioration is rapid. Various styles of construction are available which combine quick interchangeability and large filtering area per square foot cross-sectional area. Use of several independent units in parallel is important for the reconditioning of each unit separately. Both continuous and intermittent shaking and sweeping devices remove excess dust and maintain a low resistance.

Such filters should be frequently inspected for leaks and the bags or screens should be in readily replaceable units. General practice in this field is to use a low flow per square foot of cloth and a high pressure drop, giving a relatively high efficiency as compared to general ventilation practice. However, because of the high dust loading, the discharged air may still carry excessive dust for ventilating purposes.

SMOKE STACKS

Gases discharged from smoke stacks may contain relatively large particles of cinders and carbon as well as the fine smoke which remains in relatively permanent suspension. Large particles settle quickly and are likely to constitute a serious nuisance in the immediate neighborhood. Smoke tends to spread over a large area, but in cities the large number of stacks may pollute the atmosphere over the entire city.

A variety of cinder catchers is available for catching the large cinders. Devices of the cyclone type remove much finer particles, but for anything approaching complete cleaning, electrostatic precipitators are usually required. For this purpose they usually consist of wires (at potential of 30,000 to 100,000 volts negative) suspended between vertical plates or hanging in the center of vertical cylinders. The dust collects on the plates or cylinders which are periodically vibrated or rapped to cause the dust to fall down into hoppers.

REFERENCES

Design and Application of Oil-Coated Air Filters, by H. C. Murphy (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 73).

Operation and Maintenance of Air Filters, by W. G. Frank (*Heating, Piping and Air Conditioning*, May, 1931, p. 378).

Size and Characteristics of Air-Borne Impurities, by W. G. Frank (*Heating, Piping and Air Conditioning*, January, 1932, p. 35).

Fundamental Principles in the Design of Dry Air Filters, by Otto Wechsberg (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1933, p. 217).

Operation, Maintenance of Cloth-Screened Dust Collectors, by W. F. Terry (*Heating, Piping and Air Conditioning*, May, p. 259, June, p. 304, 1933).

Testing and Rating of Air Cleaning Devices Used in General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

A.S.H.V.E. RESEARCH REPORT No. 1094—Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed, and Air Velocity Through Filter, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 415).

A.S.H.V.E. RESEARCH REPORT No. 1122—Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 339).

A.S.H.V.E. RESEARCH PAPER—The Effect of Lint on Air Filter Performance, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1940, p. 61).

The Dust-Free Space Surrounding Hot Bodies, by H. H. Watson (*Transactions of the Faraday Society*, Vol. 32, 1936, Pt. 2, p. 1073).

A New Electrostatic Precipitator, by G. W. Penney (*Electrical Engineering*, January, 1937, p. 159).

Chapter 29

FANS

Classification, Performance, Fan Efficiency, Characteristic Curves, System Characteristics, Selection of Fans, Volume Control, Fan Designations, Motive Power

IN heating and ventilating practice, fans are used to produce air flow except where positive displacement is required, in which case compressors or rotary blowers are used. Fans are classified according to the direction of air flow as (1) *axial flow* or *propeller* type if the flow is parallel with the axis, and (2) *radial flow* or *centrifugal* type if the flow is parallel with the radius of rotation.

Axial flow fans are made with various numbers of blades of a variety of forms. The blades may be of uniform thickness (sheet metal), either flat or cambered, or may be of varying thickness of so-called aerofoil section (airplane propeller type). Where an axial flow fan is intended for operation at comparatively high pressures the hub sometimes is enlarged in the form of a disc and the fan is known as a *disc fan.*

Radial flow or *centrifugal fans* include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades. The angularity of the blades determines the operating characteristics of a fan; a forward curved blade is found in a fan having slow speed operating characteristics, while a backward curved blade is found in a fan having high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types in each general class can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type. The most common field of service for fans of the propeller type is in moving air against moderate resistances, especially where no long ducts or heavy friction must be overcome and where noise is not objectionable, whereas centrifugal fans are commonly employed for operation at the comparatively higher pressures and where extreme quietness is necessary.

FAN PERFORMANCE

Fans of all types follow certain laws of performance which are useful in determining the effect of changes in the conditions of operation. These

laws apply to installations comprising any type of fan, any given piping system and constant air density, and are as follows:

1. The air capacity varies directly as the fan speed.
2. The pressure (static, velocity, and total) varies as the square of the fan speed.
3. The power demand varies as the cube of the fan speed.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

When the density of the air varies the following laws apply:

4. At constant speed and capacity the pressure and power vary directly as the density.

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.0749 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.0602 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.0602}{0.0749} = 0.80 \text{ in.}$$

$$\text{Power} = 4 \times \frac{0.0602}{0.0749} = 3.20 \text{ hp}$$

5. At constant pressure the speed, capacity and power vary inversely as the square root of the density.

Example 3. If the speed of the fan of Example 2 is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.0749}{0.0602}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.0749}{0.0602}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.0749}{0.0602}} = 4.46 \text{ hp}$$

6. For a constant weight of air:
   (*a*) The speed, capacity, and pressure vary inversely as the density.
   (*b*) The horsepower varies inversely as the square of the density.

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.0749}{0.0602} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.0749}{0.0602} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.0749}{0.0602} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.0749}{0.0602}\right)^2 = 6.20 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the horsepower output to the horsepower input.

The horsepower output is expressed by the formula:

$$\text{Air Horsepower}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356} \qquad (1)$$

When the static pressure is used in the computation it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but, owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static efficiency}^1 = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \qquad (2)$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total or dynamic pressure. This efficiency is variously known as the total, dynamic, or mechanical efficiency, and may be expressed as follows:

$$\text{Mechanical or Total efficiency}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \qquad (3)$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different per-

[1]See Standard Test Code for Centrifugal and Axial Fans, Third Edition of 1938.

centages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans are determined by tests performed in accordance with the Standard Test Code for Centrifugal and Axial Fans[2] prepared jointly by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.* The results of tests are plotted in different ways: the abscissae may be the

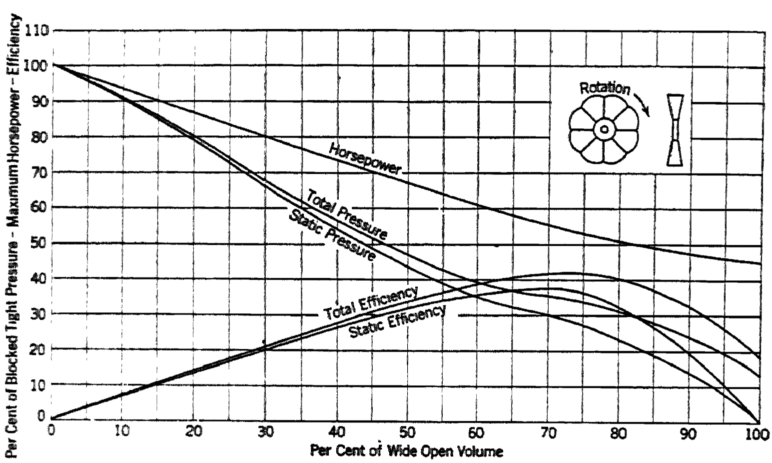


FIG. 1. OPERATING CHARACTERISTICS OF AN AXIAL FLOW FAN

ratio of delivery, assuming full open discharge as 100 per cent, and the ordinates may be static pressure, dynamic pressure, horsepower and efficiency. Pressures may be expressed in per cent of the maximum pressure in the manner shown in the illustrations in this chapter, but in engineering calculations they are sometimes expressed in proportion to the pressures due to the peripheral velocity.

It should be noted that characteristic curves of fan performance are plotted for a constant speed. Some variation in values of efficiency may occur at different speeds but such variation is usually slight within a wide range of speeds. Fans of similar design but of different size will also show some difference in efficiency. Figs. 1 to 4 show characteristic curves for different types of fans using blades of various shapes, but without reference to the design of housing employed. The efficiency curves are therefore not serviceable for making rigid comparisons of efficiencies obtainable with blades of the various shapes but are intended merely to show reasonable values and more particularly to show the manner in which variations occur with changes in fan capacity.

[2]A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407. Amended in A.S.H.V.E. TRANSACTIONS, Vol. 37 1931, p. 363. Third Edition of 1938.

Axial flow fan characteristics are indicated by Figs. 1 and 2. These fans, when properly designed, have a satisfactory efficiency at low resistance, comparing favorably in this respect with centrifugal fans. They are low in cost and economical in operation and occupy relatively little space. Although this type of fan can operate against considerable resistance, the noise often becomes objectionable, so that it does not always compare favorably with centrifugal fans for such service. With most of the designs which employ blades of uniform thickness the power increases rapidly with an increase in resistance.

The curves (Fig. 1) show the rapid reduction in capacity and increase in power as the resistance increases. The low efficiency when overcoming

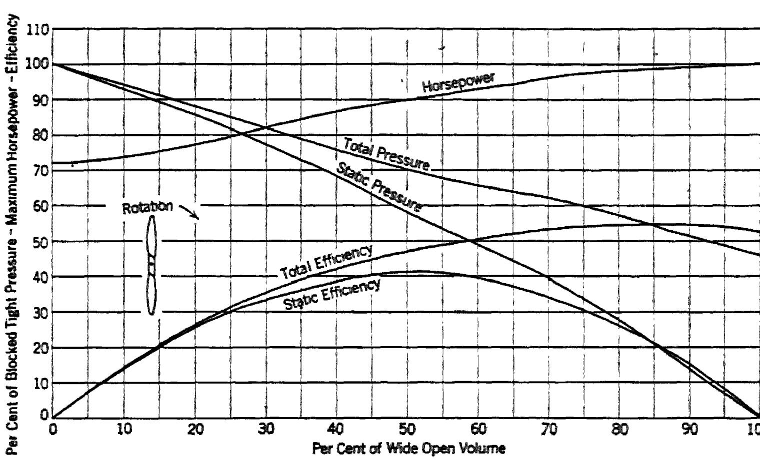


FIG. 2. OPERATING CHARACTERISTICS OF AN AIRPLANE PROPELLER FAN

heavy resistance is due to the low speed of the blades near the hub as compared to the relatively high peripheral or tip speed. The air driven by the blade area near the rim can pass back through the less effective blade area at the hub more easily than it can overcome the duct resistance.

Fig. 2 shows the performance of the *airplane propeller fan* in which the blades are similar in shape to those of an airplane propeller but of varying number according to the pressure to be developed. This fan usually operates at a higher speed than does the former type of propeller fan, and with a different power characteristic, the power remaining fairly constant throughout the range of pressures, being somewhat less at the higher than at the lower pressures. The flatness of the horsepower curve indicates the advantage of this type of fan in preventing overloading of motors where fluctuations in pressure occur. Variations in the diameter, width, pitch, camber, and the thickness of the blades provide a considerable degree of flexibility in design, so that the peak of total efficiency may be made to occur at wide-open volume or at various percentages of that volume.

Another advantage of this type of axial flow fan is its low resistance to air passage when standing still. There are some installations in which such a characteristic is desirable.

The straight blade (*paddle-wheel*) or partially backward curved blade type of fan is practically obsolete for ventilation. Its use is largely confined to such applications as conveyors for material, or for gases containing foreign material, fumes and vapors. The open construction and the few large flat blades of these wheels render them resistant to corrosion and tend to prevent material from collecting on the blades. This type of fan has a good efficiency, but the power steadily increases as the static pressure falls off, which requires that the motor be selected with a moderate reserve in power to take care of possible error in calculation of duct resistance.

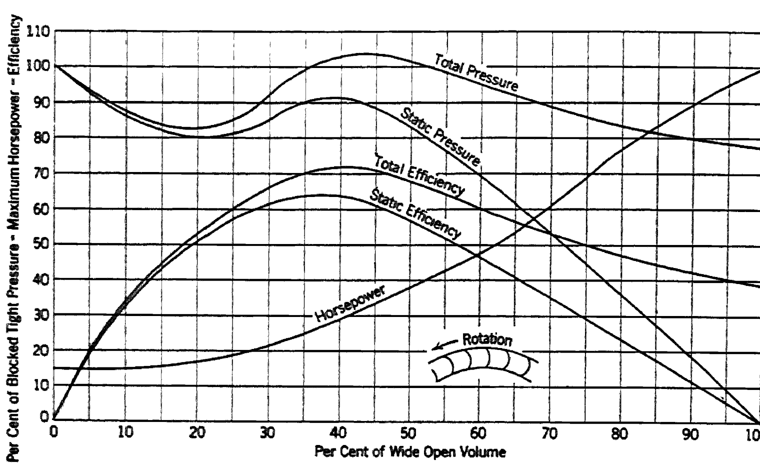


FIG. 3. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

The *forward curved multiblade fan* is the type most commonly used in heating and ventilating work, as it has a low peripheral speed, a large capacity, and is quiet in operation. (See Fig. 3.) The point of maximum efficiency for this fan occurs near the point of maximum static pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continually from low to peak capacity, but if reasonable care is exercised in figuring resistance there is no danger of overloading the motor.

The outstanding characteristics of the *full backward curve multiblade type fan* are the steep pressure curves, the non-overloading power curve, and the high speed. (See Fig. 4.) This fan operates at a peripheral speed of approximately 250 per cent of the forward curve multiblade type for like results. The pressure curves begin to drop at very low capacity and continue to fall rapidly to full outlet opening. The steep pressure curves tend to produce constant capacity under changing pressures. Where wide fluctuations in demand occur, this type of fan is desirable to prevent overloading of motors. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type

makes it adaptable for direct connected electric motor drives. The high speed may necessitate somewhat heavier construction and more operating attention or service. The dimensional bulk for a given duty often is 150 per cent of that of a forward curve multiblade type fan.

Between the extremes of the forward and the full backward curve blade type centrifugal fans a number of modified designs exist, differing in the angularity or in the shape of the blades. Common among these designs are the straight radial blade type, the radial tip type, and the double curve blade type with a forward angle at the heel and a slight backward angle at the tip of the blade. Characteristic curves of these types show

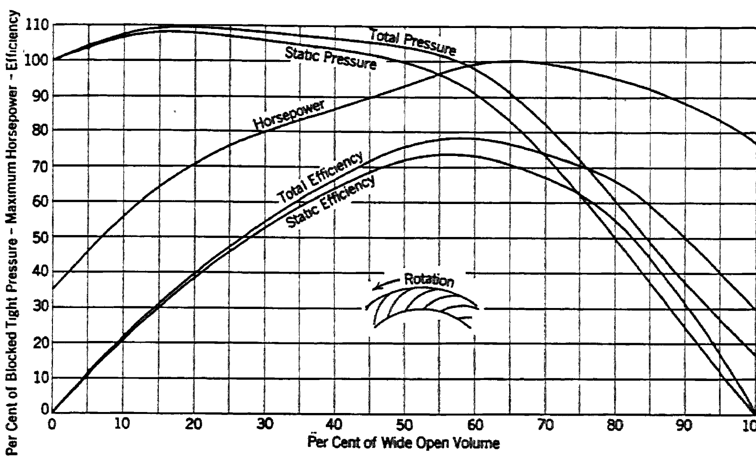


FIG. 4. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

varying degrees of resemblance to the curves of Figs. 3 and 4, according to the degree of similarity to one or the other of the two designs of fan considered.

SYSTEM CHARACTERISTICS

A given fan performs as determined by the real characteristic of the system to which it is attached. When a different performance of a fan is desired, it is necessary to either change the speed of the fan (as *A* to *B* or *C* to *D* in Fig. 5), or to change the system (as by moving a damper from *A* to *C* in Fig. 5). If the speed of the fan is changed, the new point of operation is the intersection of the constant speed static pressure—cubic feet per minute curve for the new speed with the system characteristic. If the system is changed, the new point of operation is the intersection of the constant speed static pressure, cubic feet per minute curve with the new system characteristic.

Heating and ventilating systems follow the simple parabolic law quite closely but other types of systems follow some other more or less complex relation. The more complex systems can be separated into their component parts whose individual characteristics are known and the summation of the characteristics of the several parts of a system will give the composite characteristic of the system.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures.

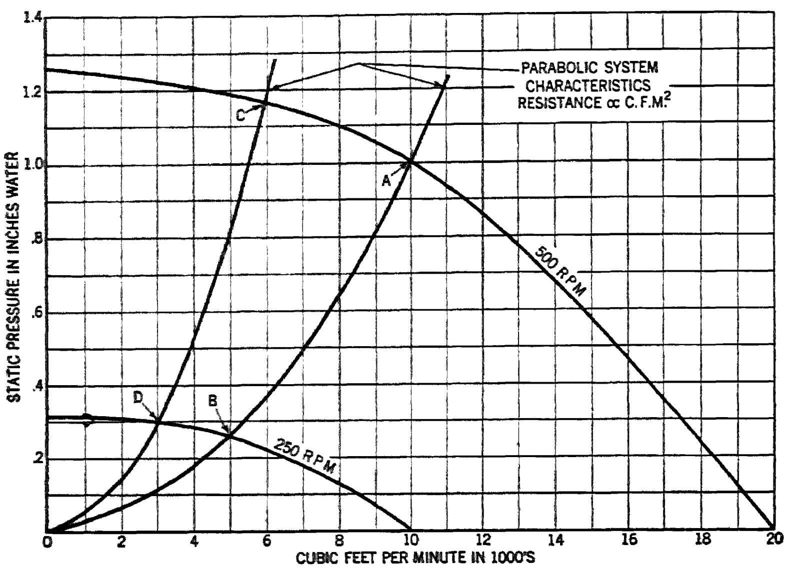


FIG. 5. ILLUSTRATION OF OPERATING POINTS OF A GIVEN FAN AT TWO SPEEDS ON THE SAME AND DIFFERENT SYSTEMS

Knowing the requirements of the system, the main points to be considered for fan selection are (1) efficiency, (2) speed, (3) noise, (4) size and weight, and (5) cost.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures:

1. Volume of air in cubic feet per minute (68 F, 50 per cent relative humidity, 0.07488 lb per cubic foot).
2. Outlet velocity.
3. Revolutions per minute.
4. Brake power.
5. Tip or peripheral speed.
6. Static pressure.

The most efficient operating point of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Fans for Ventilating and Air Conditioning Systems

Two important factors in selecting fans for ventilating systems are efficiency (which affects the cost of operation) and noise. First cost and space available are secondary. The fans should be selected to operate at maximum efficiency without noise. Because noise in a ventilating system is irritating and a cause for complaint, fans must be selected of proper size in order to reduce it to a minimum. Noise may be caused by other factors than the fan, namely, high velocity in the duct work, unsatisfactory location of the fan room, improper construction of floors and walls, and poor installation. Where noise is chargeable directly to the fan, it is caused either by excessive peripheral speeds, or the fan is of insufficient size. It should be remembered, however, that the tip speed

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR FORWARD CURVED MULTIBLADE VENTILATING FANS

Static Pressure Inches of Water	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
1/4	1000–1100	1520–1700
3/8	1000–1100	1760–1900
1/2	1000–1200	1970–2150
5/8	1100–1300	2225–2450
3/4	1200–1400	2480–2700
7/8	1300–1600	2660–2910
1	1500–1800	2820–3120
1 1/4	1600–1900	3162–3450
1 1/2	1800–2100	3480–3810
1 3/4	1900–2200	3760–4205
2	2000–2400	4000–4500
2 1/4	2200–2600	4250–4740
2 1/2	2300–2600	4475–4970
3	2500–2800	4900–5365

required for a specified capacity and pressure varies with the type of blade, and that a tip speed which may be excessive for the forward curved type is not necessarily so for the backward or slightly backward type. A noisy fan usually is one which is operated at a point considerably beyond maximum efficiency.

For a given static pressure there is a corresponding outlet velocity and peripheral speed wherein maximum efficiency is obtained. If a fan is selected to operate at this point, the cost of operation and the noise can be held within control.

To aid in selecting fans as near as possible to the point of maximum efficiency, there are listed in Tables 1 and 2 for each static pressure corresponding outlet velocities and tip speeds which will give satisfactory results. The proper tip speed for a given static pressure varies with the design of wheel and with the number of blades or vanes in the wheel.

Lower outlet velocities than those listed in Table 1 may be employed, but care must be exercised to avoid selecting a fan for operation below its useful range. It is typical of all multiblade fans to have a sudden drop in the static pressure curve near a fully closed position and the selection

of fans in this zone must be avoided. The useful range of the fans of Table 2 extends over the full length of the performance curve.

In exhaust ventilating systems where the air column moves toward the fan, noise due to the higher tip speeds and outlet velocities will not be so readily transmitted back through the air column to the building as when the air column is moving toward the rooms. Therefore higher outlet velocities may be used, but this will be at the expense of increased horsepower.

Amply large fans should always be used for both exhaust and supply systems, as there may be and usually is leakage despite the most careful workmanship, necessitating the delivery of more air at the fans than is exhausted from or supplied through the openings in the various rooms.

Long runs of distributing ducts, heaters, and air washers require definite increments of the total pressure which a supply fan in a ventilating system must overcome. These static pressures should be considered when selecting the fan characteristics, speed, and power.

Table 2. Good Operating Velocities and Tip Speeds for Multiblade Ventilating Fans with Backward Tipped and Double Curved Blades

Static Pressure Inches of Water	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
¼	800–1100	2600–3100
⅜	800–1150	3000–3500
½	900–1300	3400–4000
⅝	1000–1500	3800–4500
¾	1100–1650	4200–5000
⅞	1200–1750	4500–5300
1	1200–1900	4800–5750
1¼	1300–2100	5300–6350
1½	1400–2300	5750–6950
1¾	1500–2500	6200–7550
2	1600–2700	6650–8050
2¼	1700–2800	7050–8550
2½	1800–2950	7450–9000
3	2000–3200	8200–9850

Fans picked within the limits of Table 1 will operate close to the point of maximum efficiency. No attempt has been made to select these limits for quiet operation, since this is a relative term and varies with the type and location of the installation.

The connection of a fan to a metallic duct system should be made by canvas or a similar flexible material so as to prevent the transmission of fan vibration or noises. Where noise prevention is a factor the fan and its driver should have floating foundations.

Fans for Drying

Both axial flow and centrifugal types of fans are used for drying work. Propeller fans are well adapted to the removal of moisture-laden air when operating against low resistance and when handling air at low temperatures. Motors on these fans usually are of the fully-enclosed moisture-

proof types so that saturated air or air containing foreign material will not injure the motors.

Unit heaters employing axial flow fans are widely used in the drying field. In drying, these fans may be used with unit heaters where not too much duct work is required and where air is to be delivered against pressure, since the noise developed from the high peripheral speed of these fans is not ordinarily objectionable in process work.

Centrifugal fans of the multiblade type generally are selected to supply air for drying, as they are capable of delivering large volumes of air against all pressures likely to be encountered.

Belt-driven fans usually are to be preferred to direct-connected fans since efficient motor speeds do not usually coincide with efficient fan speeds. Replacement of a standard motor is quick and easy if it is belted.

Wherever drying is done throughout the year and where air requirements change as the drying conditions change, the drying can be speeded up or reduced through control of the fan capacity. This may be done by changing the fan speed or by varying the outlet area with dampers. A throttled outlet reduces the volume and reduces the power.

Due to the low speeds of forward curved multiblade or paddle-wheel type fans, these can be direct-connected to reciprocating steam engines, and the exhaust steam from the engines may be used in the heating apparatus. In selecting engine-driven fans for drying processes, where a large quantity of exhaust steam is used in the heaters, a smaller fan and greater power consumption may be used, because power economy is not essential under this condition.

Where static pressure in a drier varies, and where several fans must operate in parallel, fans are to be preferred which have a continuously rising pressure characteristic, such as is given by backward-curved or double-curved blades. This type of fan is well adapted for direct-connected motors of the higher speeds. (See Chapter 40 on Drying Systems).

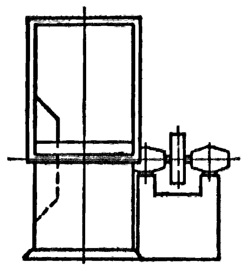
Fans for Dust Collecting and Conveying

The application of fans for handling refuse, dust, and fumes generated by machine equipment is covered in Chapter 39. Information is given regarding the methods for determining air quantities, the velocity required for carrying various materials and the method of determining maintained resistance or total static pressure at which the fan is to operate. The selection of a proper size fan is at times governed by the future requirements of the plant. In many instances, additional future capacity is anticipated and should be provided for.

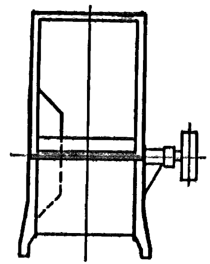
Having determined the necessary volume of air and the maintained resistance or static pressure required, the proper size fan may be selected from the fan manufacturers' performance charts or capacity tables. The fan chosen should be the size that will provide the required ultimate quantities with the minimum power consumption.

FAN VOLUME CONTROL

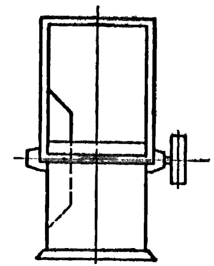
Some method of volume control of fans usually is desirable. This may be done by varying the peripheral velocity or by interposing resistance, as



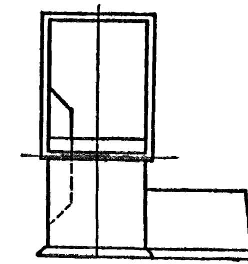
Arr. 1.
For belt drive.
Wheel overhung. Bearings on pedestal.



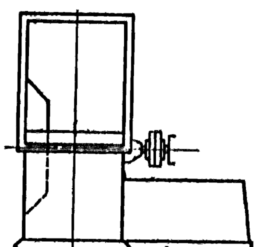
Arr. 2.
For belt drive.
Pulley and wheel overhung. Bearings in bracket on fan housing. Made only in smaller sizes for reversible discharge.



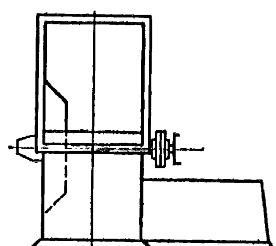
Arr. 3.
For belt drive.
Pulley overhung. Bearings supported on fan housing.



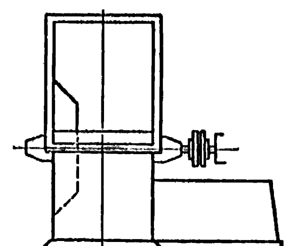
Arr. 4.
For direct drive.
Wheel overhung. No bearings on fan. Wheel mounted on motor or engine shaft. Pedestal for motor or engine.



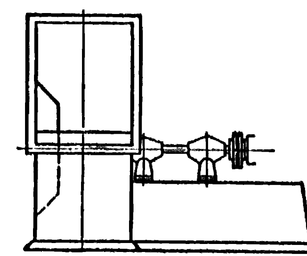
Arr. 5.
For direct drive.
Wheel overhung. Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine.



Arr. 6.
For direct drive.
Three-bearing arrangement with fan bearing at inlet side. Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine.



Arr. 7.
For direct drive.
Similar to Arr. 6, but with two bearings on fan, and flexible instead of rigid coupling.



Arr. 8.
For direct drive.
Similar to Arr. 5, but with two bearings on pedestal with motor, and flexible instead of rigid coupling.

FIG. 6. ARRANGEMENT OF FAN DRIVES

by throttling-dampers. Both methods, since they reduce the volume of air, reduce the power required. In many installations adjustments of volume are desirable during varying hours of the day. In others an increased supply of air in summer over that needed for winter is demanded. Experience is required in deciding whether speed-control or damper-control shall be used for specific cases. Where noise is a factor, it is desirable to obtain reduction of volume by reduction of speed as throttling by damper may be a source of noise. When controlling capacity by speed, the fan law given on page 544 applies.

FAN DESIGNATIONS

Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is clockwise, the fan, blower, or blast wheel will be designated as *clockwise*. If the proper direction of rotation is counter-clockwise, the designation will be *counter-clockwise*. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.)[3]

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word "hand," but specify "clockwise" or "counter-clockwise."

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of air discharge is vertically up.

Down blast: If the line of air discharge is vertically down.

All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

In order to prevent misunderstandings, which cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 6 are suggested.

If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. The *backward curved* and *double curved* types with backward tip operate satisfactorily in double or in parallel operation.

MOTIVE POWER

It is no easy matter to predetermine the exact resistance to be encountered by a fan or, having determined this resistance, to insure that no changes in construction or operation shall ensue which may increase air resistance, thus requiring more fan speed and power to deliver the required volume, or which may reduce air resistance, thus causing delivery of more air and a consequent increase of power even at constant speed.

It is recommended, therefore, for centrifugal type fans that the rated power to be supplied shall exceed the rated fan power by a liberal margin, when *forward curved* types are used. When *backward* or *double curved*

[3]Recommendations adopted by the *National Association of Fan Manufacturers*.

blade types are used, motors with ratings very close to that of the fan horsepower demand can be employed, provided the fan has a limiting horsepower characteristic.

Justification for liberal power provision exists also in the possibility of varying demand due to changes in ventilation requirements, intensity of occupation, and weather conditions.

The motive power of fans should be determined in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.*

Fans may be driven by electric motors, steam engines (either horizontal or vertical), gasoline or oil engines, and turbines, but as previously stated the drive commonly used is the electric motor.

REFERENCES

Mine Ventilation, by J. J. Walsh (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 659).

Fan Blower Design, by H. F. Hagen (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 175).

The Specific Characteristics of Fans, by M. C. Stuart and J. B. Lusk (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1936, p. 507).

Non-Dimensional Fan Characteristics, by H. Carlton Moore (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1937, p. 580).

Section X, A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

Coal Miners Pocket Book.

Constructive Mechanism and the Centrifugal Fan, by George D. Beals.

Fans, by Theodore Baumeister, Jr.

Fan Engineering, Buffalo Forge Company.

Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932.

Mechanical Engineers' Handbook, by Kent.

Mechanical Engineers' Handbook, by Lionel S. Marks.

The Centrifugal Fan, by Frank L. Busey.

The Fan, by Charles H. Innes.

The Theory and Performance of Axial-Flow Fans, by L. S. Marks and J. R. Weske.

Theories and Practices of Centrifugal Ventilating Machines, by D. Murgue, translated by A. L. Stevenson.

Chapter 30

AIR DISTRIBUTION

Definitions, Grille Locations, Standards for Satisfactory Conditions, Factors Affecting Distribution for Cooling and Heating, Air Outlet Noises, Selection of Supply Outlets, Balancing System

CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor. Supplying the proper amount of air is one problem; properly distributing it from the point where it leaves the fan is another. The distribution problem may be further divided into: (*a*) distribution *to* the various spaces served by the system, (*b*) distribution *in* these spaces. This discussion is primarily limited to division (*b*), reference being made to the duct system only insofar as it affects the performance of the air distribution outlets.

Definitions

1. *Supply Opening or Outlet:* Any opening through which air is delivered into a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

2. *Exhaust Opening:* Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

3. *Outside Air Opening:* Any opening used as an entry for air from outdoors.

4. *Grille:* A covering for any opening and through which air passes.

5. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.

6. *Multiple Louver Damper:* A damper having a number of adjustable blades.

7. *Single Louver Damper:* A damper having one adjustable blade.

8. *Face:* A grille with provision for attaching a damper.

9. *Register:* A face with a damper attached.

10. *Flange:* The portion (either integral or separate) of a grille, face, or register extending into the duct opening for the purpose of mounting.

11. *Frame:* The portion (either integral or separate) of a grille, face, or register extending around the duct opening for the purpose of mounting.

12. *Margin:* The margin of a grille, face, or register is one-half of the difference between the duct dimension and overall dimension measured either horizontally or vertically.

13. *Fret:* The member separating the openings of a grille, face, or register.

14. *Free Area:* The total minimum area of the openings in the grille, face, or register through which air can pass.

15. *Core Area:* The total plane area of the portion of a grille, face, or register bounded by a line tangent to the outer edges of the outer openings through which air can pass.

16. *Mean Area:* The total of the core and free areas divided by two.

17. *Duct Area:* The area of a cross-section of the duct based on the inside dimensions at the point where the grille, face or register is mounted.

18. *Percentage Free Area:* The ratio of the free area to the core area expressed in percentage.

19. *Aspect Ratio:* The ratio of length of the core of a grille, face or register to the width.

20. *Throw:* The distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which air motion reduces to 50 fpm.

21. *Envelope:* The outer boundary of an air stream.

GRILLE LOCATIONS

The location of supply and exhaust openings is extremely important if a satisfactory installation is to be secured. Very frequently, however, the room or building is planned and constructed with practically no consideration of this problem. The engineer of today is more likely than not to have as his problem a building that was constructed long before any consideration whatever was given to air conditioning it. Consequently, the room shapes, the location of columns and beams, and other details of architecture frequently make it difficult to properly locate the supply openings. In general, for a cooling installation, the grilles should be located high enough from the floor to prevent the discharge of air directly upon the occupants of the room, and far enough down from the ceiling to

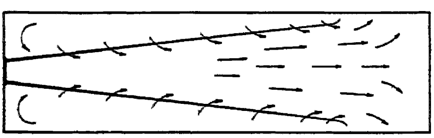


FIG. 1. PLAN VIEW LONG THROW SUPPLY OPENING

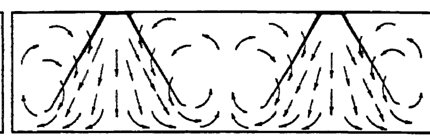


FIG. 2. PLAN VIEW SHORT THROW SUPPLY OPENINGS

minimize the possibility of streaking, and to permit induction of air from all sides of the stream. If the stream actually strikes the ceiling, but at a small angle, the throw will be increased somewhat if the ceiling is smooth. If the angle at which the stream hits the ceiling is 20 deg or more, or if the flow along the ceiling is obstructed by panel mouldings or beams, air velocity may be rapidly lost and a decreased throw result. The air stream also should be so directed that it will not strike nearby columns or beams in such a way as to cause misdirection of the air stream or drafts. Where the room is of irregular shape, as an ell, or where it has an alcove in one side, consideration should be given to obtaining satisfactory circulation in these corners. Frequently this cannot be done except by the use of multiple supply openings. In using multiple supply openings, care must be taken that the several air streams do not interfere with each other, until their velocities have been reduced to values which will not cause high turbulence and a drafty condition. Beams and offsets in the ceiling will cause little difficulty when substantially parallel to the direction of flow, unless they are of considerable depth, but, when positioned across the air stream, may cause drafts and failure to secure satisfactory circulation in that portion of the room farthest from the supply opening. In the case of a heating installation, down-drafts produced by such obstructions may not be serious, because the air will rapidly lose its downward motion, but the possibility of failure to obtain satisfactory circulation still exists.

The location of supply openings should, if possible, be such as to take advantage of the maximum velocity permissible from a noise standpoint. For instance, the spaces illustrated in Figs. 1 and 2 may be satisfactorily served by either arrangement. However, by taking advantage of the long throw, to which the arrangement in Fig. 1 lends itself, fewer supply openings are required and additional savings are effected in the sheet metal work.

In solving the problem of properly conditioning a room of irregular shape, where multiple wall supply grilles are objectionable, a ceiling supply opening of the type illustrated in Figs. 3 and 4 may very often be the best solution.

In choosing the most desirable location for the return air grille, consideration should be given to its effect on circulation of the air through the room. It is generally true that the return air grille should be placed on the same wall as the supply and near the floor level. This results in a U-shaped air path (Fig. 5) which covers the room thoroughly. The arrangement shown in Fig. 6 should be avoided, because it tends to create

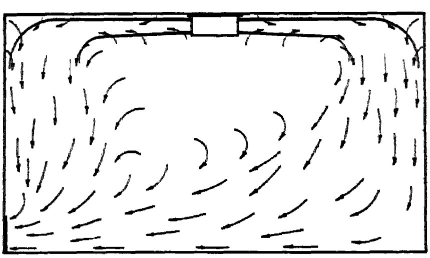


FIG. 3. ELEVATION VIEW, CEILING SUPPLY OUTLET WITH RETURN OPENING IN WALL

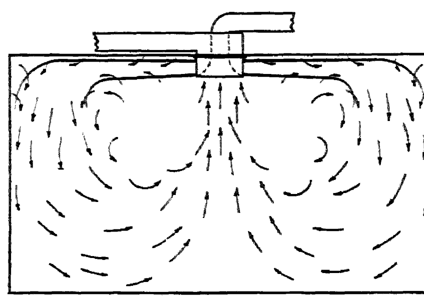


FIG. 4. ELEVATION VIEW, CEILING SUPPLY AND RETURN OPENINGS

a stagnant section below the supply grille. What would otherwise be an unsatisfactory dead spot in a room may in some instances be taken care of by location of the return air grille near that area (Fig. 7).

STANDARDS FOR SATISFACTORY CONDITIONS

The most satisfactory air condition cannot be definitely stated for any particular individual without conducting a series of tests with that individual as subject; some persons are less sensitive than others to variations in temperature, humidity, air velocity and noise. The best that can be done is to attempt to set limiting conditions leaning toward the values of these variables which produce a condition of comfort for the greatest number of individuals. On a cooling installation, the allowable deviation from average room temperature, that is, the temperature of puffs of air which may strike a person momentarily, is a function of the room temperature as well as the velocity of the air. For instance, in a room controlled at 72 F, a puff of air at 70 F might be uncomfortable to an individual, even at relatively low velocities, whereas if the average room temperature were 80 F, air at 78 F, even at moderate velocities, might be very satisfactory. However, air at 78 F in an average room

temperature of 83 F would be cold. In general, other conditions being equal, for the range of temperatures normally encountered in living quarters on cooling installations, the permissible deviation from average room temperature varies from approximately 1 F at the low end of the range to about 3 F at the high end of the range. In this matter, it is important to consider the particular problem in the light of the type of occupancy. For instance, greater deviations from room temperature and higher velocities may be permitted in a garage or a hotel hallway than would be permissible in an office or living room. The velocity which may be considered the permissible maximum differs with the temperature deviation for a given installation, but an absolute maximum under any conditions might be considered that which would produce a mechanical

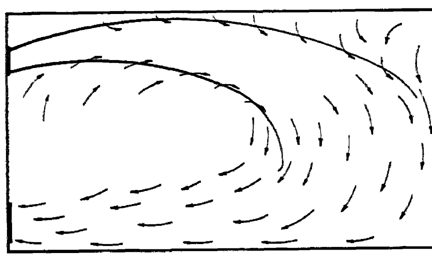


Fig. 5. Elevation View Correctly Located Return Opening

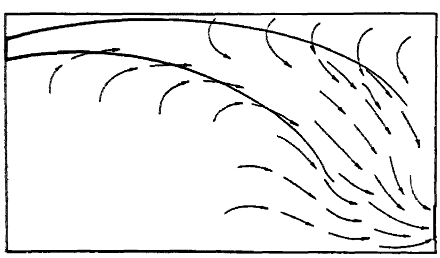


Fig. 6. Elevation View of Improperly Located Return Opening

disturbance, such as the movement of a person's hair or disturbance of papers on a desk. Humidity is an important consideration in the determination of one's feeling of comfort; however, if the room generally is assumed to be at a satisfactory value of relative humidity, the designer is justified in neglecting this factor when considering permissible fluctuations in temperature and velocity in the occupancy zone. This is true because the maximum allowable temperature fluctuation results in an unnoticeable humidity change.

The standards that might be set up for maximum allowable room temperature deviation and air velocity would not be the same for both heating and cooling installations. In the former case, any appreciable temperature deviation is likely to be above rather than below the average room temperature, whereas the reverse is most likely to be true on a cooling installation. Further, because air movement has a cooling effect in itself, the feeling of warmth due to temperatures above room tem-

perature is counteracted to a certain extent so that an individual may be subjected to higher velocities of warm air without the feeling of discomfort occasioned by the same velocities of cool air. In every case, it should be the purpose of the designing engineer to keep the conditions within the zone of occupancy as nearly uniform as possible, securing minimum temperature deviations and low velocities.

It is impractical to measure momentary temperature differences with any degree of accuracy in the field, but in checking a given installation it will generally be found satisfactory to measure velocity only, since on cooling installations high velocities normally occur with low temperatures, and on heating installations high velocities occur with high temperatures. That is, in the former case, the chilled supply air loses its velocity and undergoes an increase in temperature as it settles into the occupancy zone, whereas in the latter case the heated supply air loses its velocity and undergoes a decrease in temperature during this process. Therefore, if the

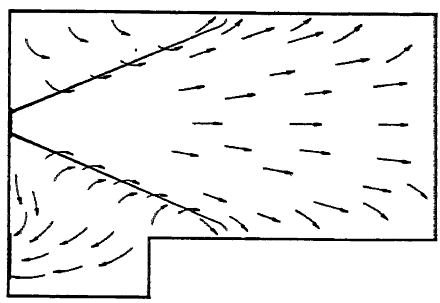


FIG. 7. PLAN VIEW CORRECTLY LOCATED RETURN OPENING ELIMINATING STAGNANT SPACE

average velocities within the occupancy zone are not excessive, one is fairly safe in assuming that the temperature difference is also within permissible limits.

The subject of sound control is covered in Chapter 32 and it is recommended for detailed review before consideration of the problem of air supply opening noise. An understanding of the relation between sound intensity and loudness level in decibels, as well as the effect of the presence of sound absorbent materials in the room, is particularly necessary. A more detailed discussion of the nature of this problem appears later; whereas the following comments refer to what constitutes a satisfactory noise condition.

Obviously, the nature of the conditioned space is important when considering the allowable supply opening noise. In factories, press rooms, and similar spaces where the noise level is 65 db or higher, no complaints of grille noise are likely to be made. On the other hand, some homes, offices, hospitals, and, most of all, radio broadcasting and movie sound studios, present a real problem which must be intelligently attacked if a satisfactory installation is to be made. In this chapter the noise of the air supply openings (and returns) only is considered, it being assumed that the noise or sound level of the room without the supply opening noise

includes that which may be contributed by fans, motors, duct work, and other items of conditioning equipment. The control of noise from these sources is another problem (see Chapter 32). Where sound control is important, the actual room sound level without conditioning equipment should be known. If feasible, the contribution of the conditioning equipment, less supply openings, should be estimated to secure the working sound level. If this correction is not made, the use of the first value errs in the direction of safety.

It is evident that the point within the room which should concern the designer in this problem is that at which the supply opening noise is greatest. A tentative standard *listening point* relative to the supply opening is suggested later in this discussion, and it is assumed that the supply opening noise data are taken with reference to this point. If it is desired that the supply opening noise result in an inaudible addition to the existing noise level, it is safe to assume the total supply opening noise to be 5 db below room level. This results in an increase in total noise of slightly over 1 db, which is unnoticeable. If an increase of 3 db is permissible, the supply opening noise level may be equal to the room noise level alone. All supply openings in the room must be considered, as will appear later; the returns may be ignored only if they are so sized that the velocity of air through them is much less than through the supply opening.

DISTRIBUTION FACTORS IN ROOM COOLING

In attempting to design a satisfactory air distributing system, it is first necessary to properly locate the grilles in accordance with the recommendations already stated. Assuming that the best locations have been selected, it then becomes necessary to choose the proper grille for that location. The considerations involved are the amount of air to be handled, the velocity permissible from the standpoint of noise, and the distance the air should carry. The distance it will carry, assuming no obstructions, is affected by a number of factors which are listed below:

1. The temperature difference between incoming and room air.
2. Height of grille above floor and distance below ceiling.
3. Face velocity.
4. Core area.
5. Core aspect ratio.
6. Design of grille.

The manner in which the above factors affect throw may be generally stated. All other things being constant any one of these will produce a longer throw; a higher temperature of incoming air; a greater height above the floor; a higher velocity; a greater area[1]. A larger aspect ratio will decrease throw. The design characteristics of the grille will, of course, vary the throw.

In consideration of what constitutes the possible throw of a supply opening under a given set of conditions, it is important to remember that the throw may be unsatisfactory for any one of several reasons:

[1]A.S.H.V.E. Research Report No. 1076—Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 77).

1. It may be so long that it will strike the far side of the room and come down the wall with velocities higher than are permissible,

2. It may be so short that it will fail to carry the full length of the room, and short-circuit to the return air supply opening, or

3. It may spill into the center of the room.

In the first case, the system fails for lack of uniform distribution and the presence of cold areas. In the second case, the standards as to velocity and temperature difference in the zone of occupancy may be satisfactorily met, but air distribution and circulation throughout the entire room is not accomplished, with the result that the end of the room away from the outlet would not be satisfactorily conditioned. In the third case, the shortcomings of both case one and case two are present. It is evident, therefore, that for a given supply opening discharging air at a given velocity, there is a maximum and a minimum length of room which can be satisfactorily handled. In the latter, the velocity of the air down the far wall is just within the maximum permissible, while in the former, satisfactory circulation is barely accomplished.

In general, the higher the supply opening is above the floor, the greater may be the difference between room air and incoming air temperatures.

Assuming that proper supply openings for a given installation have been selected, unsatisfactory performance may still result due to the construction of the duct work immediately back of the supply openings. Performance data on the grilles and registers of various manufacturers are based upon results obtained with the air approaching the grille perpendicularly and at uniform velocity over the entire duct cross-section. Where this condition does not exist in practice, performance predictions based on published data cannot be expected to be realized. Every precaution should be taken to secure as nearly ideal conditions in the approaching air stream as are possible.

In addition to disturbances due to the construction of the duct work itself are those which may be created by dampers immediately behind the grille. Where either multiple louver or single blade dampers are used, considerable deflection of the air stream may result, if it is throttled appreciably by these means. This is particularly true when the fins of the register core are perpendicular to the damper blades. If the core has sufficient depth and the fins are parallel to the blades, there is a marked tendency to straighten the air stream, although some deflection may still result.

Any attempt to secure a low face velocity and high duct velocity by the construction of any expanding chamber immediately behind the grille is likely to be unsuccessful. In order to expand from a small duct to a larger one, and have the air stream fill the duct at the end of the diverging section without turbulence, angle A in Fig. 8 should be about 3 deg for four-sided expansion and about 5 deg for two-sided expansion. From this it is apparent that an attempt to secure equivalent results with a short connection would be futile. What actually happens when this is attempted is illustrated by the arrows in Fig. 8. When localized high velocities through the supply opening exist from this cause or any other, the noise produced will naturally exceed that which the supply opening area and average face velocity would lead one to expect. This fact should

be remembered in considering the use of register dampers, particularly in those cases where there must be considerable throttling with the damper to balance a poorly designed system. Where reduction of noise is important, it is recommended that balancing dampers be placed in the duct *ahead* of the acoustic duct lining.

Similar unequal face velocities, aggravated by a deflection of the air stream, are obtained with the arrangement shown in Fig. 9. The latter may be corrected by inserting a turning member in the elbow back of the outlet face as shown in Fig. 10. The importance of straightening the air stream and effecting uniform distribution over the entire face of the supply opening cannot be over-emphasized.

DISTRIBUTION FACTORS IN ROOM HEATING

The problem in the case of a heating installation is substantially the same as in cooling, with a few exceptions. Because the temperature of the incoming air is above that of the room, there is no tendency for it to drop and consequently the throw is not particularly affected by temperature difference in a low ceiling room. In general, the air should be deflected downward where the grille is above the occupancy zone, and this is particularly desirable where the ceiling is high. For the same reason, that is, to keep the heat in the occupancy zone and to avoid excessive temperature at the ceiling, it is desirable to have the grille comparatively low on the wall, and just slightly above the occupancy zone. If the grille is lower than this, it may create an unsatisfactory condition of very warm air at quite high velocities where it can possibly strike the occupants of the room. Where the velocities are very low, the grilles may even be satisfactorily located below the 6-ft level, although the immediate vicinity of the supply openings will probably be useless for occupancy because of high temperature. Essentially, the problem is to keep the incoming air up for cooling, and down for heating, until it is thoroughly mixed with the room air. Grilles and registers which are adjustable for deflection upward and downward, either by moving the fins or inverting the grille, are in general use.

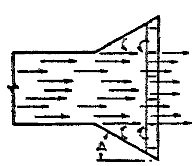


FIG. 8. EFFECTS OF EXPANDING DUCT

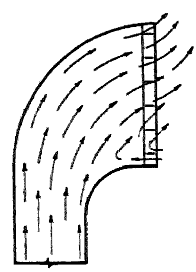


FIG. 9. UNEQUAL FACE VELOCITIES

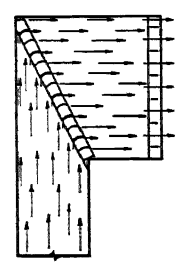


FIG. 10. EFFECT OF TURNING MEMBER

AIR SUPPLY OPENING NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at

a constant rate[2]. Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the supply opening) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units (sabins) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db. However, it is not satisfactory to consider the grille noise on this basis (wherein the sound power received directly from the source is small compared with that received by reflection) since in practice the occupants of the room may be quite close to the grille. The nearer the listener is to the sound source, the greater the proportion of the sound intensity which is due to direct transmission.

In the absence of generally accepted standards at this time it is suggested that the loudness level 5 ft from the lower edge of the supply opening, measured downward at 45 deg in a plane perpendicular to the supply opening at its center, represents about the maximum within the zone of occupancy. The cases where persons are nearer to the supply opening than this are rare and are ignored in the consideration of this problem. Although the effect of sound absorbent material on the intensity at the 5-ft station is not nearly so great as at more remote points in the room, it should not be ignored without consideration of the error involved. An average living room may contain 100 sabins (absorption units). If this be decreased to 50 sabins, the *diffuse* or reflected sound level would be increased 3 db. However, at the 5-ft station the increase would be less than 2 db. If the absorption of the room be increased to 200 sabins, one might expect a reduction in diffuse noise of 3 db; but at the 5-ft station the reduction would be less than 1½ db. Furthermore, even though the absorption be increased without limit (as in free space) the reduction would still be less than 2 db because of proximity to the source.

In comparing sound ratings of various grilles, the following must be known if the information is to be intelligently applied:

1. The threshold intensity on which the decibel ratings are based.
2. The distance from the grille at which data were taken.
3. If stated as loudness level versus velocity for a given grille, the *core area* (not nominal area) must be known.
4. The sound absorbing characteristics of the test room.
5. Whether or not corrected for test room loudness level; if not, the room level (without grille noise) must be known.
6. Methods used for recording data. (Characteristics of sound meter).

Data mentioned in this chapter are based on these assumptions:

1. Threshold intensity = 10^{-16} watts per square centimeter[3].
2. Microphone location 5 ft from lower edge of supply opening on a line downward at 45 deg and in a plane bisecting the supply opening perpendicularly.
3. Where data are given as loudness level versus velocity, the rating is per square foot of core area.
4. The room is assumed to have 100 sabins absorption.

[2]The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 81).

[3]American Tentative Standards for Noise Measurement, *American Standards Association*.

5. Plotted data are loudness levels of *supply openings only*, correction having been made for test room level.

6. Data taken with a direct reading sound-level meter with frequency weighing network intended to approximate the response of the human ear.

If the published ratings are in terms of decibels per square foot, correction must be made for area to secure the total sound level of supply openings of more or less than one square foot area from Formula 1.

$$\text{Decibel Addition} = 10 \log_{10} A \tag{1}$$

where

A = core area, square feet.

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. Since total loudness and air flow are both functions of velocity

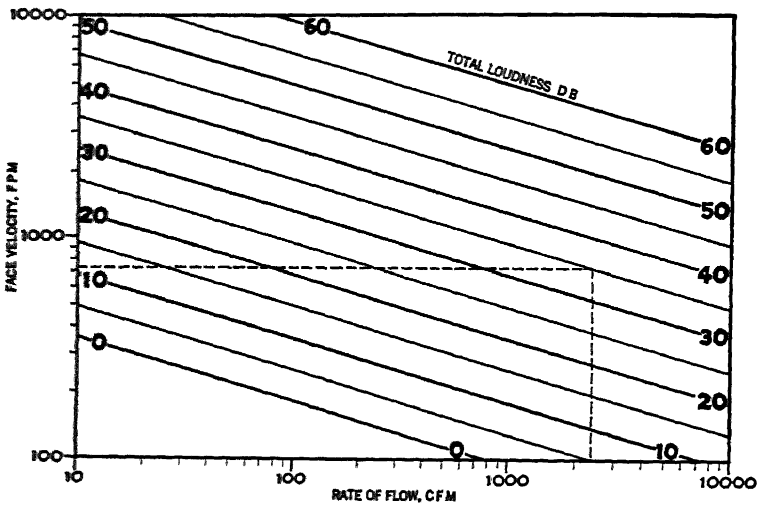


FIG. 11. AIR FLOW AND LOUDNESS CHART

and area, the solution of the problem by use of the previous analysis implies a trial and error method. It has been found possible to present these data with sufficient practical accuracy as a family of uniform curves as illustrated in Fig. 11. With this chart it is possible to find directly the velocity in feet per minute which will give a predetermined total loudness at a predetermined rate of flow expressed in cubic feet per minute. The values used are arbitrarily chosen for the purpose of discussion and do not necessarily represent data referring to any particular design of air supply opening. It is assumed that Fig. 11 is based on a room having 100 sabins of sound absorption. In such a room the sound level due to other sources may be 40 db. As previously stated a supply opening having a noise level of 35 db would be substantially inaudible in such a room.

If 2400 cfm are required with a total noise due to supply opening of 35 db, a velocity (Fig. 11) of about 725 fpm may be used. From this velocity and the rate of flow, the core area can be computed. This determination was on the basis of a room absorption of 100 sabins. If

the absorption is greater, the 725 fpm velocity is safe, since the loudness level will go down. However, correction can be made if desired by the use of the chart of Fig. 12. Thus, if the absorption is 200 sabins, a correction of +1.3 db may be made and the permissible velocity becomes that corresponding to a total loudness level of 36.3 db or approximately 800 fpm. If the room is highly reflecting and has an absorption of less

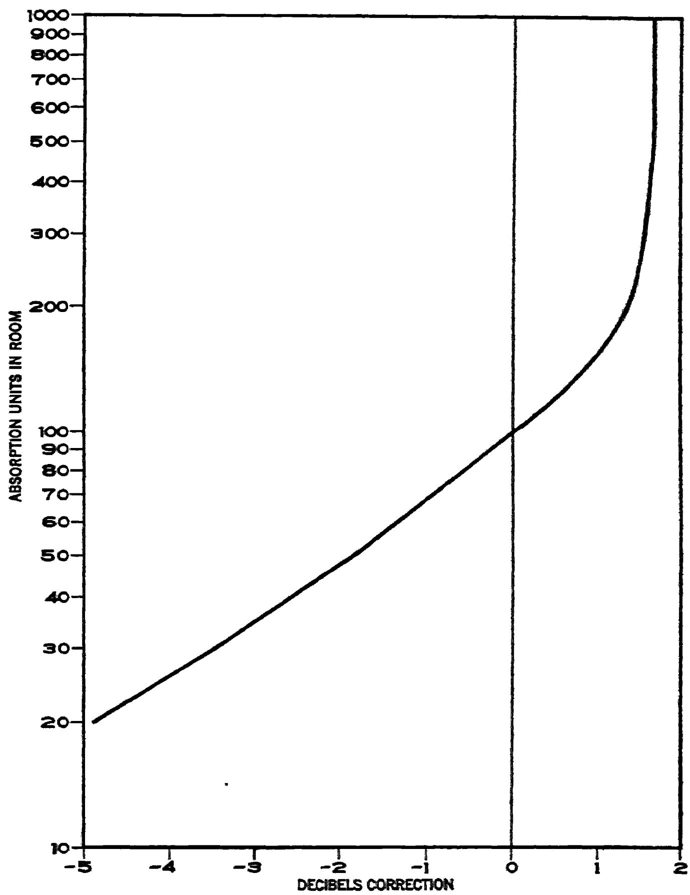


FIG. 12. ROOM ABSORPTION CORRECTION CHART

than 100, correction is much more important. For instance, for 35 sabins a correction of −3 db must be made and the maximum velocity corresponding to 32 db total loudness chosen is approximately 600 fpm.

Where more than one supply opening must be considered, the problem is more complicated. If a similar supply opening is added in a far corner of a highly absorbent room, the change in noise level at the 5-ft station at the first supply opening is small; however, if the room is small, or highly

reverberant or both, the intensity at the 5-ft station may be almost doubled and the noise level increased nearly 3 db thereby. The simplest method of handling this problem, and one which errs in the direction of safety, is to treat the room as though all the air were being supplied by one supply opening. Thus, if two outlets, each supplying 1000 cfm are used, the value 2000 cfm should be used with Fig. 11. Although this method may place an unwarranted limit on velocity when used in a large room, it is seldom that such a room has a noise level low enough to justify a more complicated though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply openings. That is, if 1000 cfm are supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig. 11.

SELECTION OF SUPPLY OPENINGS

After the heating and cooling load calculations have been made (Chapters 5 and 6), and a suitable supply air temperature selected, the volume of air required for each space can be determined. The next step is to determine the velocity at which the air may be introduced into the space quietly and without creating objectionable drafts.

Present day grille design coupled with the introduction of effective acoustical treatment for minimizing fan and duct noises have made grille face velocities in excess of 1500 fpm feasible, and 600 to 1200 fpm is now common practice. This range of velocities is approximately three times higher than common practice values of a few years ago.

Since high velocities make for smaller ducts and supply openings, and therefore savings in space as well as greater flexibility in locating the duct work to the best advantage, selection of design velocities is a very important step. The selection of proper velocity requires that the designer have reliable data applicable to the particular make of supply opening proposed for usage. Even under these circumstances, the problem is one of *cut and try* because permissible velocity may be limited by either noise or throw.

A method for selecting supply openings is outlined below in the form of a sample cooling problem, using numerical values which have no reference to any particular make of supply opening.

1. The load calculations have been made; a suitable temperature differential has been selected (it is to be understood that the data referred to from this point on are based on this temperature differential), and the volume of air required determined. Assume that Fig. 13 represents a small general office having a noise level of 40 db and that 2400 cfm must be supplied for proper conditioning.

2. Select a tentative location for the supply opening or openings, having in mind the type of grille most likely to effect proper distribution. In this particular case, two supply openings having a wide spread appears to be a logical choice.

3. Data from which to determine velocity which corresponds to 2400 cfm and a noise rating at least 5 db below the noise level of the office may be presented in a number of forms, one of which is shown in Fig. 11. (Fig. 11 represents assumed values only. In practice similar data should be obtained from the manufacturer whose supply openings are being considered. Several similar charts or tables may be necessary to cover any one manufacturer's complete line.) From Fig. 11 it will be noted that for 2400 cfm the

type of grille selected may be used at velocities up to 725 fpm without exceeding 35 db; that is, 5 db below the noise level of office.

4. Having determined the velocity, the core area becomes fixed at 3.31 sq ft or 397 sq in. per supply opening. In this problem, the two grilles in question are so close together that consideration of their combined area in determining the permissible velocity from the standpoint of noise introduces little error.

5. The type grille selected has thus far been found satisfactory from a noise standpoint, provided the face velocity does not exceed 725 fpm. The next consideration is throw, which may be assumed to be 16 ft, and by reference to a manufacturer's catalogue the proper correlative test data may be checked with the throw assumed. It is of course evident that one or more types of grilles may satisfy the requirements, and that in any one type there will be a choice of supply opening proportions. It will also be evident that the tentative selection of a supply opening having a wide spread may be unsatisfactory from the standpoint of throw, in which event a second choice should be made and the procedure repeated.

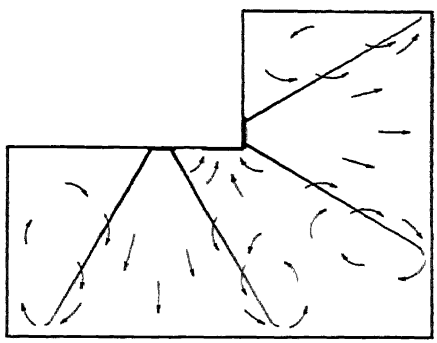


FIG. 13. PLAN VIEW TYPICAL GENERAL OFFICE

In the case of a heating problem, the method of solution is the same, but the manufacturer's data should, of course, be based on tests with air above room temperature. However, if data based on chilled air are used for a heating problem the grille selection will err on the side of safety.

TYPES OF SUPPLY OPENINGS

Grille, register or supply opening design for attaining uniform distribution and minimum air resistance consists of various fixed and adjustable arrangements. Some types are designed with directing air blades, fins, bars, louvers, or thin metal strips shaped into a series of grooves or tubes, all of which may be set into a suitable round, square or rectangular frame. In order to attain desired long or short air throws, the emergence of air from the supply opening may be directed to straight, deflecting, converging or jet air streams depending upon the supply opening design. Designs which direct the air stream to produce an ejector effect within the enclosed space tend to mix the room air with the conditioned air.

There are several types of centrally located ceiling or wall type supply openings. One of these consists of several round, hollow, cone-shaped flaring members with one or more of the smaller members acting as ejectors and injectors. An idea for producing even distribution of air consists of a perforated ceiling made of a suitable architectural surface and installed a small distance below the normal ceiling level of the room. In the space

provided by this suspended ceiling a plenum chamber is formed into which the conditioned air is introduced. From the plenum space the air is permitted to diffuse through the large number of small ceiling openings into the room.

BALANCING SYSTEM

In designing an air conditioning system, it should be the aim of the engineer to so proportion the duct system that proper distribution of air to every supply opening will be obtained. Since this is almost impossible to accomplish in practice, it becomes necessary to have means of balancing the system to secure the desired amount of air in each space. There are a number of ways in which this may be accomplished, some of which are:

1. Dampers on the supply and return grilles.
2. Dampers in the supply and return ducts.
3. Reducing the effective area of some supply openings by blank-offs.
4. Combinations of dampers in both supply and return air.

Dampers on the supply grilles themselves are objectionable because of their effect on the air stream. Dampers on the return grilles are frequently helpful in building up a static pressure in the room to prevent infiltration of outside air, and at the same time reduce the volume of incoming air. However, it is frequently impossible to sufficiently reduce the incoming air by this method alone. A damper in the supply duct some distance back of the supply opening forms a very satisfactory means of regulating the flow without disturbing distribution across the supply opening face. A damper in the return air duct has the advantage over one immediately behind the grille in that it does not tend to create high localized velocities through the grille as the latter might do if nearly closed. Blank-offs consisting of pieces of sheet metal covering a portion of the supply opening face can frequently be used satisfactorily, although determination of just what is required is a matter of experiment, and the balancing of the system is not nearly so conveniently accomplished as with dampers. Dampers in both supply and return air form the most flexible means of controlling the supply to the room and the static pressure within the room. When feasible, these dampers, particularly those in the supply ducts, should be a substantial distance from the supply opening, and ahead of the acoustic duct lining if used. Due consideration should also be given to the use of the several volume control and uniform distribution devices now available. See *Catalog Data Section.*

REFERENCES

A.S.H.V.E. RESEARCH PAPER—The Use of Air Velocity Meters, by Prof. G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1939, p. 703).

A.S.H.V.E. RESEARCH PAPER—The Performance of Stack Heads, by D. W. Nelson, D. H. Krans and A. F. Tuthill (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1940, p. 131).

A.S.H.V.E. RESEARCH PAPER—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August, 1940, p. 501).

A.S.H.V.E. RESEARCH PAPER—Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1940, p. 559).

Chapter 31

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Friction Loss Chart, Elbow Friction Losses, Proportioning the Losses, Duct Sizes, Procedure for Duct Design, Velocities, Main Trunk Ducts, Proportioning the Size for Friction, Velocity Method, Equal Friction Method, Duct Construction Details

THE resistance of an air handling system can be computed from the methods and data given in this chapter. The actual resistance for any given installation, however, may vary considerably from the calculated resistance because of variation in the smoothness of materials, the type of joints used and the ability of the mechanics to fabricate in accordance with the design. It is best to select fans and motors of sufficient size to allow a factor of safety. Volume dampers should be installed in each branch outlet to balance the system. It is improbable that the required quantities of air will be delivered at each outlet without adjustment of the dampers, which usually results in a total pressure exceeding that of the design, unless a liberal factor of safety is allowed.

The flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

In the development of the formulae, diagrams, and tables for the flow of air, use is made of the following basic equation for the flow of fluids:

If H_v be the velocity head in feet of a fluid, and the velocity, V, be expressed in feet per minute, the fundamental equation is

$$V = 60\sqrt{2g\,H_v}$$

The factor g is the acceleration due to gravity, or 32.16 ft per second per second.

It is usual to express the head in inches of water for ventilating work and, since the heads are inversely proportional to the densities of the fluids,

$$\frac{H_v}{\frac{h_v}{12}} = \frac{62.4}{d}$$

$$H_v = 5.2\,\frac{h_v}{d}$$

therefore,

$$V = 1096.5 \sqrt{\frac{h_v}{d}} \qquad (1)$$

where

V = velocity, feet per minute.
h_v = velocity head or pressure, inches of water.
d = weight of air, pounds per cubic foot.

For dry air (70 F and 29.921 in. *Hg* barometer) d = 0.07492 lb per cubic foot[1]. Substituting this value in Equation 1:

$$V = 1096.5 \sqrt{\frac{h_v}{0.07492}} = 4005 \sqrt{h_v} \qquad (2)$$

The relation of air velocity and velocity head expressed in Equation 2 is shown diagrammatically in Fig. 1 for air at 70 F and 29.92 in. *Hg* barometer.

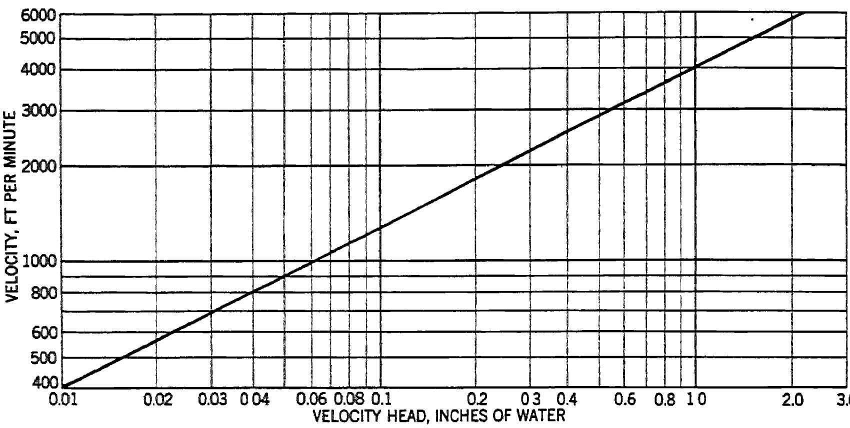


FIG. 1. RELATION BETWEEN VELOCITY AND VELOCITY HEAD FOR DRY AIR

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses for turbulent flow (which occur in all practical air flow problems) are due to the friction of air against the sides of the duct and to internal friction between air molecules. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to 0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design.

FRICTION LOSSES

Theoretically friction losses vary directly as the length of the duct, directly as the square of the velocity, and inversely as the diameter.

[1]See Chapter 46 for definition of standard air.

Since length is a fixed quantity for any system, the factors subject to modification are the area and the velocity, which determine the relation between the first cost of the duct system and the cost of the power for overcoming friction.

The friction between the moving air and pipe surface and internal friction between air molecules causes a loss of head which is numerically equal to the pressure required to maintain a given velocity, and is expressed in the following modification of Fanning's formula:

For round pipe and dry air at 70 F and 29.921 in. *Hg* barometer

$$h_L = f \frac{L}{D} h_V = \frac{L}{CD} \left(\frac{V}{4005} \right)^2 \tag{3}$$

For rectangular ducts

$$h_L = fL \left(\frac{a+b}{2ab} \right) h_V = \frac{L}{C} \left(\frac{a+b}{2ab} \right) \left(\frac{V}{4005} \right)^2 \tag{4}$$

where

h_L = loss of head, inches of water.

$h_V = \left(\frac{V}{4005} \right)^2$ = velocity head, inches of water.

V = velocity of air, feet per minute.

L = length of pipe, D = diameter of pipe, a, b = sides of rectangular duct — all in feet.

f = coefficient of friction, or friction factor.

$C = \frac{1}{f}$ = length of pipe in diameters for one head loss.

For all practical purposes C varies only with the nature of the pipe surface: $C = 60$ for perfectly smooth pipe; $= 55$ for pipe as used in planing mill exhaust systems; $= 50$ for heating and ventilating ducts; $= 45$ for smooth and 40 for rough conduits of tile, brick or concrete. However, Fritzsche states (and numerous tests check very closely) that f varies inversely as the 2/7 power of the pipe diameter, and inversely as the 1/7 power of the velocity, or inversely as the 1/7 power of capacity, which is the same thing. Thus Formula 3 may be revised as follows, based upon a loss of one velocity head (at 2000 fpm) in a length equal to 50 diameters of 24 in. galvanized swedged pipe:

$$h_L = 1.1 \frac{L}{CD^{9/7}} \left(\frac{V}{4005} \right)^{13/7} \tag{5}$$

The preceding formulae are based on dry air at 70 F, and for other conditions the friction varies directly as the air density or inversely (approximately) as the absolute temperature. The increase of friction due to increase of air viscosity with increased temperature is small and is generally neglected.

Friction Loss Chart

A convenient chart is given in Fig. 2 for determining the friction loss for various air quantities in ducts of different sizes. It has been in use for several years and has incorporated in it a factor of safety which should be adequate for any installation of good design. The general form of this chart is familiar, but it should be noted that it is corrected for changes in

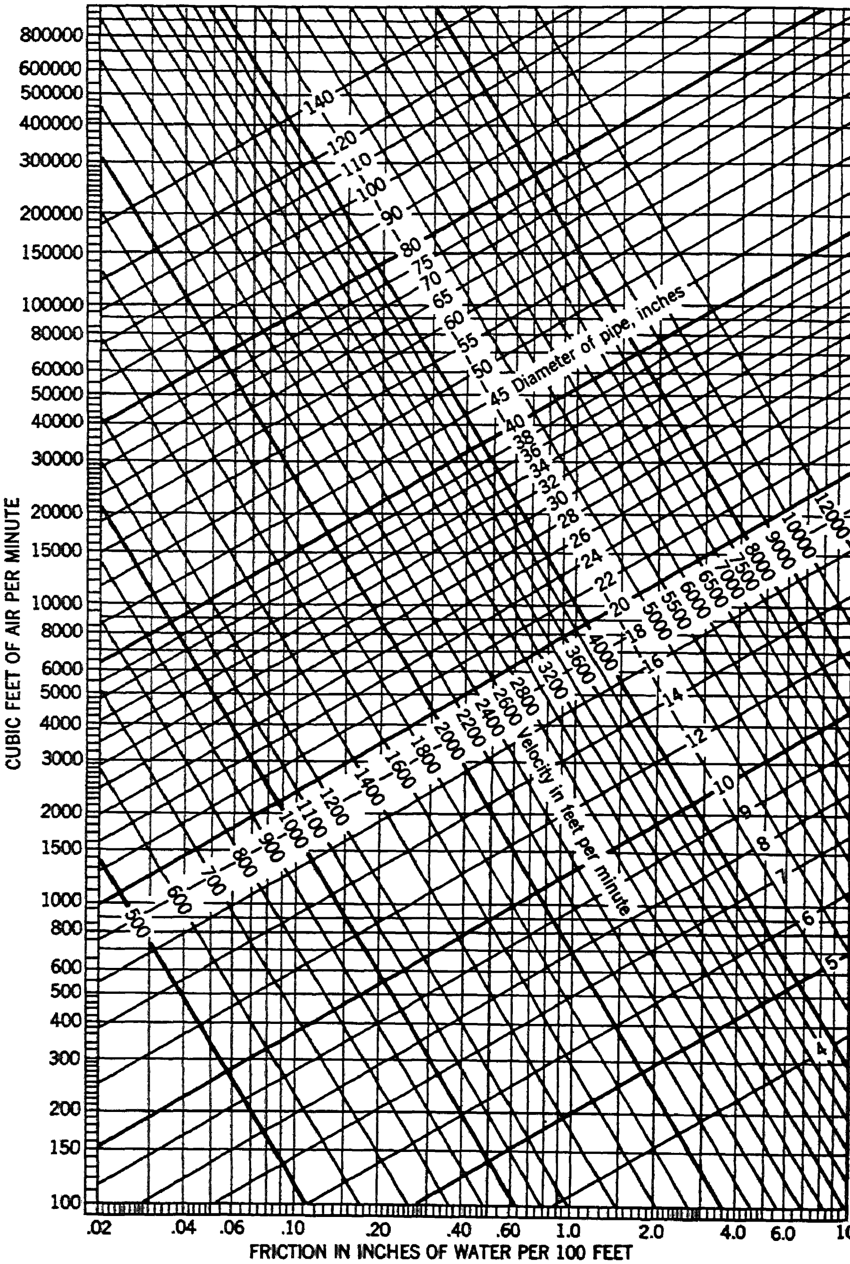


FIG. 2. FRICTION OF AIR IN PIPES

the coefficient of friction based on the rule that the coefficient of friction varies inversely as the 2/7 power of the diameter, and inversely as the 1/7 power of the velocity. The chart in Fig. 2 is based on a loss of one velocity head (at a velocity of 2000 fpm) in a length equal to 50 diameters

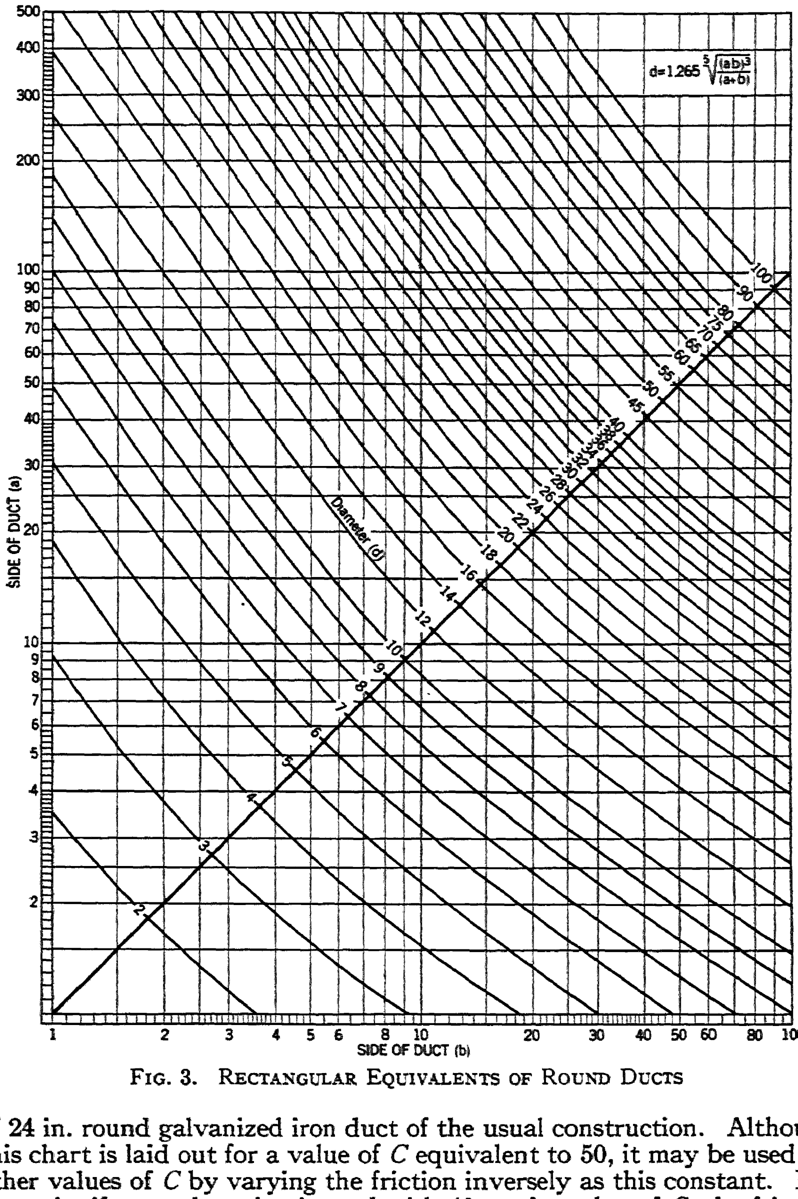


FIG. 3. RECTANGULAR EQUIVALENTS OF ROUND DUCTS

of 24 in. round galvanized iron duct of the usual construction. Although this chart is laid out for a value of C equivalent to 50, it may be used for other values of C by varying the friction inversely as this constant. For example, if a rougher pipe is used with 40 as the value of C, the friction loss as read from the chart should be multiplied by $50 \div 40$.

Example 1. Assume that it is desired to circulate 10,000 cfm of air through 75 ft of 24 in. diameter pipe. Find 10,000 cfm on the right scale of Fig. 2 and move horizontally left to the diagonal line marked 24 in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.59 in.; then for 75 ft the friction will be $0.75 \times 0.59 = 0.44$ in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Circular Equivalents of Rectangular Ducts

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have the same friction per foot of length. Table 1 gives directly the circular equivalents of rectangular ducts for equal friction and capacity, which are based on values determined from Formula 6:

$$d = 1.265 \sqrt[5]{\frac{(a\,b)^3}{a + b}} \tag{6}$$

where

a = one side of rectangular pipe, feet or inches.
b = other side of rectangular pipe, feet or inches.
d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity, feet or inches.

Rectangular equivalents of round ducts are also given in the curves of Fig. 3 which are plotted from data based on Formula 6. To use the chart, locate the curve giving the diameter of the round duct. The width and height of an equivalent rectangular duct may then be read as abscissa and ordinate of any point of the curve.

Since the friction chart, Fig. 2, and Table 1 were prepared, further research on duct friction under the direction of the A.S.H.V.E. Research Technical Advisory Committee on Air Distribution and Air Friction indicates lower pressure loss values than given in Fig. 2. Included in the work by this Committee is a revision of Fig. 2 and Table 1 to bring them into conformity with recent tests. Pending acceptance by the Society of the Committee's report, the friction chart and the table on circular equivalents of rectangular ducts for equal friction presented in earlier editions of the GUIDE have been retained.

Multiplying or dividing the length of each side of a pipe by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24-in. duct is required, it will be twice that of a 40 x 12-in. duct, or $2 \times 23.3 = 46.6$ in.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Side Rectangular Duct	8	8.5	9	9.5	10	10.5	11	11.5	12	12.5	13	13.5	14	14.5	15	15.5	16
3	5.2	5.4	5.5	5.7	5.8	5.9	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.1
3.5	5.7	5.9	6.0	6.2	6.3	6.5	6.6	6.7	6.9	7.0	7.1	7.3	7.4	7.5	7.6	7.7	7.8
4	6.1	6.3	6.5	6.7	6.8	7.0	7.1	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.3	8.4
4.5	6.5	6.7	6.9	7.1	7.2	7.4	7.6	7.7	7.9	8.0	8.2	8.4	8.5	8.6	8.7	8.9	9.0
5	6.9	7.1	7.3	7.5	7.7	7.8	8.0	8.2	8.3	8.5	8.7	8.8	8.9	9.1	9.2	9.4	9.5
5.5	7.3	7.5	7.7	7.8	8.1	8.3	8.5	8.6	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.9	10.1

Elbow Friction Losses

It is customary to express the dynamic and friction losses in elbows as equal to a number of diameters of round pipe or a number of widths of rectangular pipe. The curves in Fig. 4 are arranged to read the number of diameters or widths for determining the lineal feet of pipe having a frictional resistance equivalent to the pressure drop in the elbows. Curves *B* and *C* are based on tests of round and square elbows[2] of ordinary good sheet metal construction having a surface factor of $C = 50$.

Values obtained from Curve *A* should be used when there is any doubt as to quality of duct construction. It is suggested that this curve be used for rectangular elbows and five piece elbows as it will thus allow an additional factor of safety without seriously affecting the design.

As indicated on the chart, long radius elbows will offer much less resistance to the flow of air than short radius elbows. Experience has

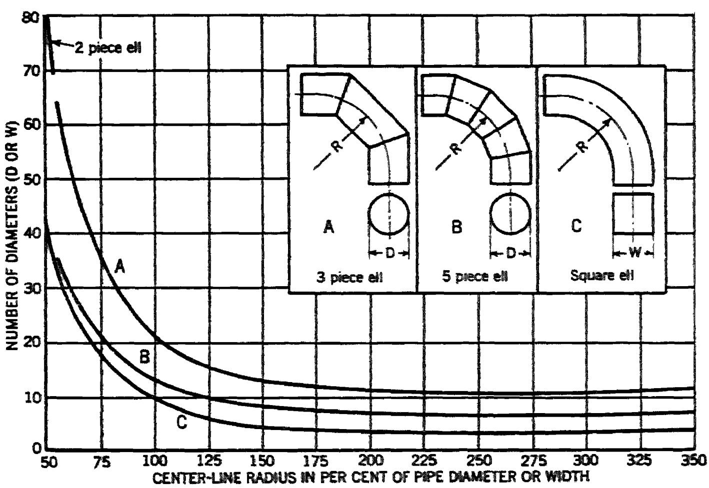


FIG. 4. LOSS OF PRESSURE IN ELBOWS

shown that good results may be expected when the radius to the center of the elbow is 1.5 times the pipe diameter or duct width parallel to the radius. Examination of the curve will indicate that little advantage is to be gained by selecting elbows having a centerline radius of more than two diameters[3]. Elbows having a radius of more than three diameters show a slightly increased resistance due to the increased length of pipe but, when used, they reduce the overall resistance of the system and therefore should not be avoided.

Where space conditions necessitate the use of short radius or square throat elbows in rectangular duct work, turning vanes should be used to reduce the pressure losses. Rough or raw edges on the vanes should be

[2]Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 366).

[3]Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker (*Heating, Piping and Air Conditioning*, July, p. 365, August, p. 427, September, p. 483, 1936).

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION[a]—(Continued)

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

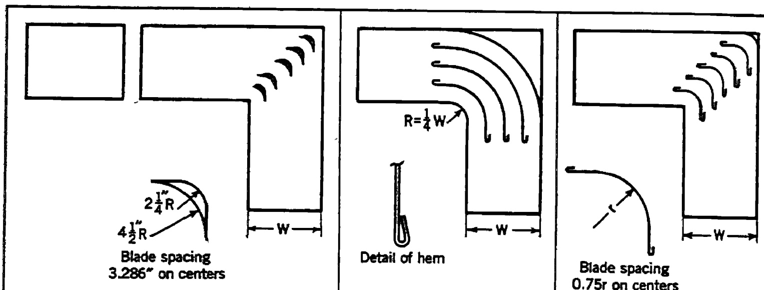
[a]Additional sizes: 4 × 5 = 4.9; 4 × 6 = 5.4; 4 × 7 = 5.8; 5 × 5 = 5.5; 5 × 6 = 6.3; 5 × 7 = 6.5.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Concluded)

Side Rectangular Duct	26	28	30	32	34	36	38	40	42	44	46	48	Side Rectangular Duct	50	54	60	66	72	78	84	88
26	28.6												50	55.0							
28	29.7	30.8											52	56.1							
30	30.7	31.9	33.0										54	57.2	59.4						
32	31.7	32.9	34.1	35.2									56	58.3	60.5						
34	32.7	33.9	35.1	36.3	37.4								58	59.3	61.6						
36	33.7	34.9	36.1	37.3	38.5	39.6							60	60.3	62.7	66.0					
38	34.6	35.9	37.1	38.4	39.5	40.7	41.8						62	61.3	63.7	67.1					
40	35.3	36.7	38.0	39.3	40.5	41.7	42.9	44.0					64	62.2	64.7	68.2					
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2				66	63.2	65.7	69.3	72.6				
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4			68	64.1	66.6	70.3	73.7				
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6		70	65.0	67.6	71.3	74.8				
48	38.5	40.0	41.5	43.0	44.4	45.6	46.9	48.1	49.3	50.5	51.6	52.8	72	65.9	68.5	72.3	75.9	79.2			
50	39.2	40.8	42.3	43.8	45.2	46.5	47.9	49.1	50.4	51.6	52.9	54.0	74	66.8	69.4	73.3	76.9	80.3			
52	40.0	41.6	43.1	44.7	46.1	47.5	48.9	50.1	51.3	52.5	53.8	55.0	76	67.6	70.3	74.2	77.9	81.4			
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52.3	53.5	54.8	56.0	78	68.4	71.2	75.2	78.9	82.5	85.8		
56	41.3	43.0	44.6	46.2	47.7	49.1	50.6	52.0	53.3	54.6	55.9	57.0	80	69.2	72.1	76.1	79.9	83.6	86.9		
58	42.1	43.8	45.4	47.0	48.5	50.0	51.5	52.9	54.2	55.5	56.8	58.0	82	70.1	73.0	77.1	80.9	84.6	88.0		
60	42.7	44.5	46.1	47.8	49.3	50.9	52.3	53.8	55.0	56.4	57.7	58.9	84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
62	43.4	45.1	46.8	48.4	50.0	51.7	53.0	54.5	55.9	57.2	58.5	59.7	86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
64	44.0	45.8	47.5	49.2	50.9	52.4	53.9	55.4	56.8	58.1	59.4	60.6	88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
66	44.7	46.5	48.2	50.0	51.6	53.1	54.7	56.2	57.6	59.1	60.4	61.6	90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
68	45.3	47.2	48.9	50.7	52.2	53.8	55.5	56.9	58.4	59.9	61.3	62.6	92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
70	46.0	47.8	49.5	51.3	52.9	54.5	56.2	57.7	59.1	60.6	62.1	63.5	94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
72	46.5	48.4	50.1	51.9	53.7	55.4	57.0	58.7	60.0	61.3	63.0	64.5	96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

avoided to prevent objectionable noise. Three typical types of vanes are shown in Fig. 5 which gives the approximate number of duct widths recommended to be used in estimating the resistance of each type.

The pressure loss through elbows of less than 90 deg may be assumed to be directly proportional to the ratio of the angle through which the turn is made. The resistance will vary widely for the large degree turns depending upon the aspect ratio and the length of straight pipe between the elbows, but for practical purposes, it may be assumed that the ratio remains proportional to the angle through which the turn is made. Reverse 90 deg elbow turns should be avoided wherever possible but, where used, the friction of the elbows indicated in Fig. 4 should be doubled for the second elbow.



Number of duct widths for equivalent resistance	13 x W	12.4 x W	28 x W

Fig. 5. Design and Corresponding Loss of Pressure in Elbows with Turning Vanes

PROPORTIONING THE LOSSES

The entrance loss through the outside air intake louvers will vary with the design of the louvers and method of connection to the system. The louvers and connecting duct will have a friction resistance of from 0.25 to 1.00 times the velocity pressure. Therefore, the total entrance loss will vary from 1.25 h_v to 2.00 h_v. Common practice is to use 1.5 h_v for a 75 per cent free area louver with connecting duct having 15 deg tapered sides. Wherever air passes through a plenum space having a negligible velocity, allowance must be made for the loss in velocity head. This may be taken as the velocity head corresponding to the difference in velocities in the plenum and the duct. Where the ducts are very smooth with long transformation fittings, a regain in static pressure is sometimes allowed, but generally ordinary construction does not warrant a consideration of this factor, and it is customary to neglect it. When it is allowed, the regain is estimated at one-half the difference between the velocity pressure at the fan outlet and at the last run of pipe.

Other losses of pressure occur through the heating units, at the air washer and at air filters. In ordinary practice in ventilation work it is usual to keep the sum of the duct losses one-third to one-half and the loss

through the other units at less than one-half of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

DUCT SIZES

Ducts and flues for gravity circulation must be sized so that the friction loss will not exceed 50 per cent of the available aspirating effect due to the temperature and height of the column of heated air. Duct systems for mechanical circulation may be sized so as to have much higher pressure losses than gravity systems. The total pressure of these systems is limited to the available pressure from the fan used.

General Rules

The general rules to be followed in the design of a duct system are enumerated herewith:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.
2. Sharp elbows and bends should be avoided unless turning vanes are used.
3. Transformation pieces should be made as long as possible. The angle between the sides and axis of the duct should never exceed 30 deg and where possible, 15 deg should be made the maximum.
4. Especial care should be taken to maintain a true cross-section and not to restrict the air flow either in transformation pieces or in elbows.
5. Rectangular ducts or flues should be made as nearly square as possible. Good practice limits the ratio between the long side and the short side to 3 to 1. In no case should this ratio exceed 10 to 1.
6. Wherever possible, ducts should be constructed of smooth material such as sheet metal. Where masonry ducts are used, proper allowance for the surface coefficient should be made.
7. The use of furred spaces, spaces between joists, etc., should be avoided unless lined with sheet metal.

Procedure for Duct Design

The general procedure for designing a duct system is outlined in the several items listed herewith:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.
2. Arrange the positions of duct outlets to insure the proper distribution of air.
3. Divide the building into zones and proportion the volume of air necessary for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity and throw.
5. Calculate the sizes of all main and branch ducts by either of the following two methods:

   a. Velocity Method. Arbitrarily fix the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections of the continuous run.

   b. Friction Pressure Loss Method. Proportion the duct for equal friction pressure loss per foot of length.

6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained in the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The air velocities given in Table 2 have been found to give satisfactory results in engineering practice. Where the higher velocities are used, the ducts should be cross-braced to prevent breathing, buckling or vibration. High velocities at one point in the system offset the effect of proper design in all other parts of the system; hence the importance of air velocities, elbow design, location of dampers, fan connections, grille and register approach connections, and similar attention to details. For industrial buildings, noise is seldom given much consideration, and main duct velocities as high as 2800 or 3000 fpm are sometimes used but, when these velocities are used, due consideration should be given to duct design, resistance pressure, fan efficiencies and motor horsepower. For department stores and similar buildings, 2000 to 2200 fpm are sometimes used in main ducts where noise is not objectionable and space conditions warrant it. Wherever velocities higher than those shown in Table 2 are used, it is essential that the ducts should be of heavier gages, have additional bracing and be carefully constructed for a minimum resistance.

Where the high velocity diffusing outlets are used, the duct velocity should not be less than the throat velocity of the diffusers, as dynamic losses occur wherever velocities are stepped up or down. One recent trend in grille design is toward the use of much higher grille and branch duct velocities. Some installations have been made with velocities as high as 1600 fpm in branches and through the net area of grilles, but many of these have proven unsatisfactory because of noise and drafts.

TABLE 2. RECOMMENDED AND MAXIMUM DUCT VELOCITIES

Designation	Recommended Velocities, fpm			Maximum Velocities, fpm		
	Residences	Schools, Theaters, Public Buildings	Industrial Buildings	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outside Air Intakes[a]	700	800	1000	800	900	1200
Filters[a]	250	300	350	300	350	350
Heating Coils[a]	450	500	600	500	600	700
Air Washers	500	500	500	500	500	500
Suction Connections	700	800	1000	900	1000	1400
Fan Outlets	1000–1600	1300–2000	1600–2400	1700	1500–2200	1700–2800
Main Ducts	700–900	1000–1300	1200–1800	800–1000	1100–1400	1300–2000
Branch Ducts	600	600–900	800–1000	700	800–1000	1000–1200
Branch Risers	500	600–700	800	650	800–900	1000

[a]These velocities are for total face area, not the net free area.

Grille manufacturers publish selection tables which size the grilles for volume of air, temperature differential and distance of throw. In following these tables, maximums should be avoided and the manner in which the duct connects to the grille should be given careful consideration. Most of the selection tables are based on straight approach to the grille. Elbow connections to supply grilles should be provided with turning vanes to equalize the face velocity. See Chapter 30 for a discussion of grilles.

Fan outlet velocities are discussed in Chapter 29 and will not be dealt with here except to indicate that fan noises should be given proper consideration.

Main Trunk Ducts

Main trunk ducts with branches are commonly used to convey the air from the fan to the grille or register outlets in preference to individual ducts from the fan to these outlets. The velocities in these ducts and branches vary according to the nature of the installation and the degree of quietness desired. The recommended velocities in Table 2, with good construction, should give satisfactory results. The maximum velocities indicated should not be used except in areas where noise is not a deciding factor.

Velocity Method

The velocity method of designing a duct system involves arbitrarily selecting velocities at various sections of the duct system with the highest velocities generally chosen at the fan and progressive lower velocities toward the duct openings to the room. To find the total static pressure against which the fan must operate, the static pressure loss of each section must be calculated separately and the total loss found by adding the individual losses of the various sections of the run having the highest resistance. Usually this is the longest run but in some cases a shorter run may have more elbows, transformations, booster heaters, etc., which will cause it to have a higher resistance pressure. This method requires judgment and experience in choosing the proper velocities to approach equal friction for all lengths of run but many engineers believe that the velocity method is handier to use than other methods and will give satisfactory results for most practical applications. The air velocities given earlier in this chapter are helpful in choosing proper velocities. Adjustable dampers or splitters are used to regulate air quantities delivered.

Equal Friction Method

The equal friction method of design is sometimes preferred because it does not require nearly so much judgment and experience in selecting the proper velocities in the various sections of a system. The usual procedure in this method of design is to select the main duct velocity to be consistent with good practice from a standpoint of noise for a particular type of building. This velocity should be less than the fan outlet velocity. All main ducts and branch ducts are sized for equal friction by the use of Fig. 2 and Table 1 or Fig. 3.

In cases where the fan or factory assembled air conditioning unit has a limited external resistance, it is necessary to divide the available resistance by the total equivalent length of the longest or most complicated run of duct to determine the resistance per 100 ft and then to size all ducts at this resistance value, which will automatically determine the duct velocities and give the desired total duct resistance. A further refinement which is sometimes used in large systems is to size each branch duct so that it has a resistance equal to the resistance of the main system at the point of juncture. Even when this refinement is added, regulating dampers are recommended in each branch.

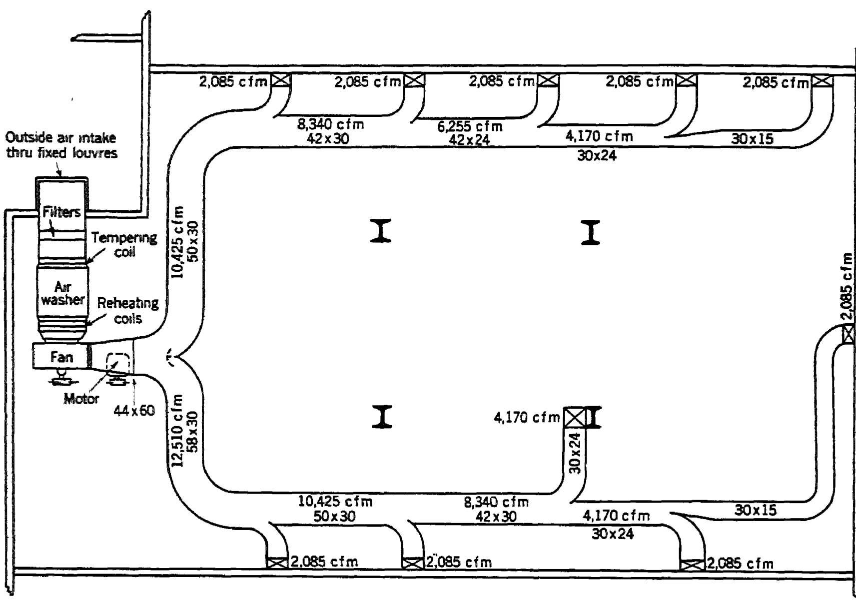


FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

After the duct system is designed the frictional resistance is calculated and tabulated together with the resistance of all component parts. The fan is then selected for the required volume of air, static pressure and outlet velocity.

Example 2. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices. The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-min air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The free area of the fresh air inlet is based on a velocity of 1000 fpm or 22,935 ÷ 1000 = 22.94 sq ft. If the air washer is provided with automatic humidity control, the tempering coil should raise the temperature of the entering air to 32 F. The washer with its automatic control will then raise the temperature from 32 F to 42 F. If the washer is not provided with automatic humidity control, the tempering coil must raise the temperature of the entering air to at least 55 F to allow for some temperature drop in the washer due to evaporation. The reheating coil is selected to raise the temperature of the

air from that leaving the air washer to 70 F. The air washer should have a maximum velocity of 500 fpm through the clear area, which, in this case, is 46 sq ft. For more detailed information on tempering coil and air washer control, see Chapter 33.

The main duct velocity selected from Table 2 is 1250 fpm which gives a fan outlet area of 22,935 ÷ 1250 = 18.354 sq ft (60 x 44 in.). From Table 1 a 60 x 44 in. duct is approximately equivalent to 56 in. diameter.

Referring to Fig. 2, a volume of 22,935 cfm through a 56 in. diameter duct gives a velocity of 1340 fpm and a resistance of 0.041 in. per 100 ft. The amount of air to be handled by each section of pipe is shown in Fig. 6, and by locating each of these values on the 0.041 in. line, the round pipe sizes are obtained and then, referring to Table 1, the equivalent rectangular sizes are selected as shown in Table 3.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser, or in each branch duct, and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 6 where the main duct divides into the 58 × 30 in. and 50 × 30 in. branches.

TABLE 3. PIPE SIZES FOR EXAMPLE 2[a]

Volume of Air (cfm)	Diameter of Pipe (Inches)	Equivalent Size of Rectangular Duct (Inches)
22,935	56	60 x 44
12,510	45	58 x 30
10,425	42	50 x 30
8,340	39	42 x 30
6,255	35	42 x 24
4,170	29½	30 x 24
2,085	23	30 x 15

[a]Velocity through grilles (not shown) to be approximately 300 fpm.

Resistance Losses for the System

(1) Fresh air intake, 1000 fpm velocity (1.5 heads × 0.0625)........ 0.094 in.
(2) Filters (from manufacturer's tables)........ 0.250 in.
(3) Tempering coil loss (from manufacturer's tables)........ 0.074 in.
(4) Air washer loss (from manufacturer's tables)........ 0.250 in.
(5) Reheating coil loss (from manufacturer's tables)........ 0.083 in.
(6) Duct resistance:

The longest run is........ 150 ft

Two 58 x 30 in. elbows (150% ratio) $\frac{2 \times 13 \times 58}{12}$........ = 126 ft

Two 30 x 15 in. elbows (150% ratio) $\frac{2 \times 13 \times 30}{12}$........ = 65 ft

Three 15 x 30 in. elbows (75% ratio) $\frac{3 \times 35 \times 15}{12}$........ = 131 ft

Total equivalent run........ 472 ft

472 ft at 0.041 in. per 100 ft........ 0.194 in.
(7) Allowance for damper adjustment, 25% of 0.194........ 0.049 in.
(8) Supply grille resistance (from manufacturer's tables)........ 0.036 in.

Total static pressure loss of system........ 1.030 in.

The fan is selected from the manufacturer's ratings to deliver 22,935 cfm at a static pressure of 1.03 in. and an outlet velocity exceeding 1250 fpm as outlined in Chapter 29.

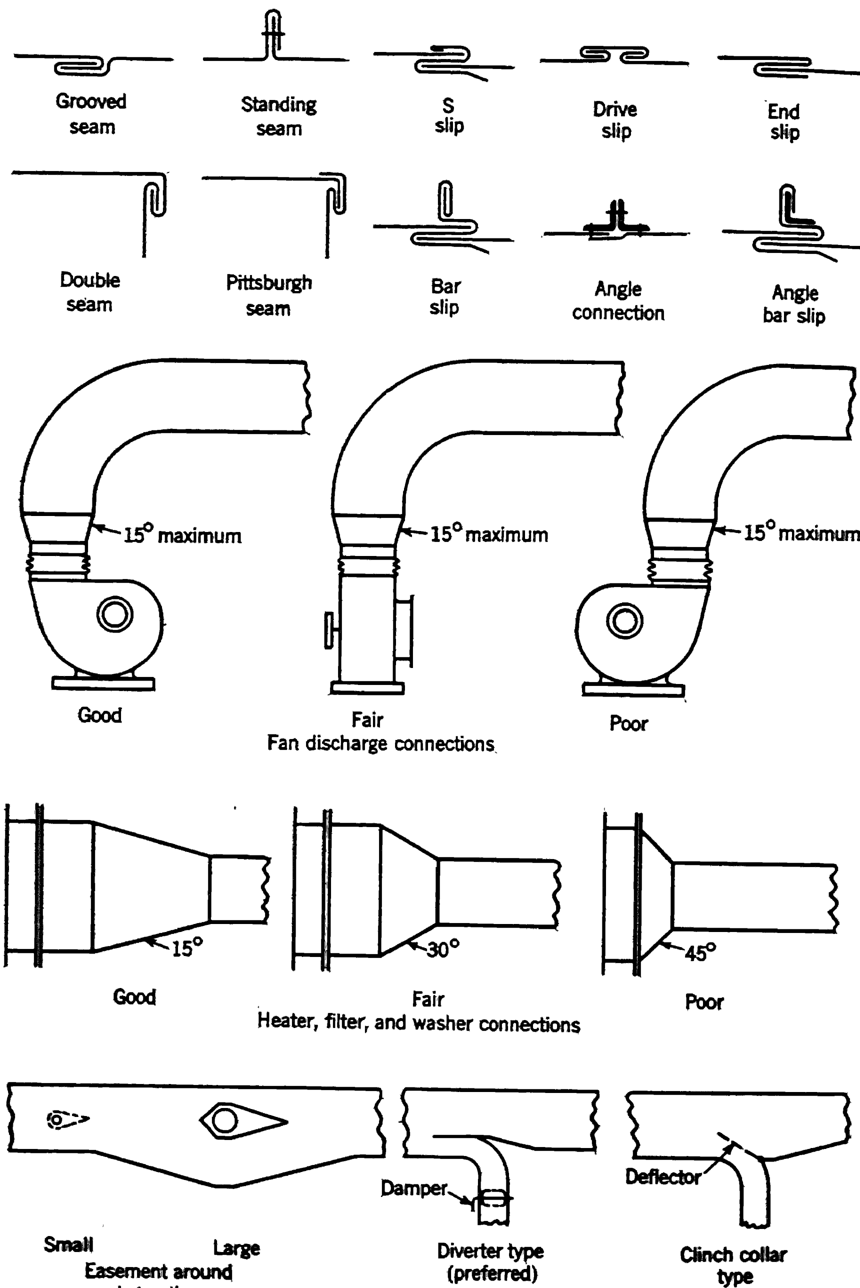


Fig. 7. Sheet Metal Duct and Arrangement Details

TABLE 4. RECOMMENDED SHEET METAL GAGES FOR DUCTS[a]

U. S. STD. GAGE	ROUND DUCTS DIAMETER, INCHES	RECTANGULAR DUCTS Maximum Side, Inches	Type of Joint Connections	Bracing
26	Up to 18	Up to 12	*S* or Drive Slips	None
24	19 to 30	13 to 24	*S* or Drive Slips	None
24		25 to 30	*S* or Drive Slips	1 in. Angle 4 ft from Slips
22	31 to 45	31 to 48	Bar or Drive Slips	1¼ in. Angle 4 ft on Centers
20	46 to 60	49 to 60	1¼ in. Angle Bar Slips or 1¼ in. Angle Connections	1¼ in. Angle 2 ft 8 in. on Centers
18	61 and up	61 to 90	1½ in. Angle Connections	1½ in. Angle 2 ft 8 in. on Centers
18		91 and up	2 in. Angle Connections	2 in. Angle 2 ft 8 in. on Centers

[a]If flat sides are not cross-broken two gages heavier material should be used.

TABLE 5. WEIGHTS OF SHEET METAL USED FOR DUCT CONSTRUCTION

U. S. STD. GAGE	BLACK SHEETS Approximate Thickness, In. Steel	Iron	Weight Per Square Foot Ounces	Pounds	GALVANIZED SHEETS[b] Approximate Thickness, In. Steel	Iron	Weight Per Square Foot Ounces	Pounds
30	0.0123	0.0125	8	0.500	0.0163	0.0165	10.5	0.656
28	0.0153	0.0156	10	0.625	0.0193	0.0196	12.5	0.781
26	0.0184	0.0188	12	0.750	0.0224	0.0228	14.5	0.906
24	0.0245	0.0250	16	1.000	0.0285	0.0290	18.5	1.156
22	0.0306	0.0313	20	1.250	0.0346	0.0353	22.5	1.406
20	0.0368	0.0375	24	1.500	0.0408	0.0415	26.5	1.656
18	0.0490	0.0500	32	2.000	0.0530	0.0540	34.5	2.156
16	0.0613	0.0625	40	2.500	0.0653	0.0665	42.5	2.656
14	0.0766	0.0781	50	3.125	0.0806	0.0821	52.5	3.281
12	0.1072	0.1094	70	4.375	0.1112	0.1134	72.5	4.531
11	0.1225	0.1250	80	5.000	0.1265	0.1290	82.5	5.156
10	0.1379	0.1406	90	5.625	0.1419	0.1446	92.5	5.781

[b]Galvanized sheets are gaged before galvanizing and are therefore approximately 0.004 in. thicker.

TABLE 6. WEIGHTS AND THICKNESSES OF STANDARD COPPER SHEETS[c]
Rolled to Weight

WEIGHT PER SQUARE FOOT Ounces	Pounds	THICKNESS, INCHES Decimal Equivalent	Nearest Fraction	NEAREST GAGE NO. B. & S.	Stubs	U. S. STD.
10	0.625	0.0135	1/64	27	29	29
12	0.750	0.0162	1/64	26	27	28
14	0.875	0.0189	1/64	25	26	26
16	1.000	0.0216	1/32	23	24	25
18	1.125	0.0243	1/32	22	23	24
20	1.250	0.0270	1/32	21	22	23
24	1.500	0.0324	1/32	20	21	22
28	1.750	0.0378	1/32	19	20	20
32	2.000	0.0432	3/64	17	19	19
36	2.250	0.0486	3/64	16	18	18
40	2.500	0.0540	3/64	15	17	17
44	2.750	0.0594	1/16	15	17	17
48	3.000	0.0648	1/16	14	16	16
56	3.500	0.0756	5/64	13	15	14
64	4.000	0.0864	5/64	11	14	13

[c]Variations from these weights must be expected in practice.

Example 3. If the rooms and offices of the hotel building of Example 2 are to be served from a manufactured unit with a capacity of 22,935 cfm against an external resistance of 0.35 in., the known resistances are calculated as:

(1)	Fresh air inlet	0.094 in.
(2)	Allowance for damper adjustment	0.050 in.
(3)	Supply grille resistance (from manufacturer's tables)	0.036 in.
	Total known resistance	0.180 in.

Subtracting this from the total available resistance: 0.35 in. − 0.18 in. = 0.17 in. available for duct resistance.

Known length of run	150 ft

The duct width is then estimated for the following elbow calculations:

Four 150% ratio elbows, 4 x 13 x 3.5 ft	182 ft
Three 75% ratio elbows, 3 x 35 x 1.5 ft	158 ft
Total estimated length	490 ft

The duct friction per 100 ft is then 0.17 ÷ 4.90 = 0.0347 in. and the mains and branches are sized from the 0.034 in. friction line in Fig. 2.

If it is desired to size each branch for equal resistance, the total resistance back to the point of juncture is calculated and the branch is then sized in a manner similar to that outlined in Example 3.

SOUND CONTROL

Frequently the problem of sound prevention in a heating, ventilating or air conditioning system imposes more severe restrictions than the prevention of excessive pressure drop. Tendencies toward higher duct velocities have produced noise control problems which require consideration of enumerable factors in air duct design. Naturally some types of occupancy and application permit relatively higher sound levels to be maintained than others, but the design trend is progressively directed towards noise reduction wherever possible. Sound absorbent materials have been successfully applied to duct construction to reduce noise. The basis used for the selection of the proper amounts of absorbent materials will be found in Chapter 32.

DUCT CONSTRUCTION DETAILS

Straight sections of round duct are usually formed by rolling the sheets to the proper radius and grooving the longitudinal seam. Rectangular ducts are generally constructed by breaking the corners and grooving the longitudinal seam, although some fabricators still use the standing seam due to lack of equipment. Elbows and transformation sections are generally formed with *Pittsburgh* corner seams because this seam is easier to lock in place than the double seam, but complicated fittings such as double compounded elbows are usually constructed with double seam corners.

The construction of these various seams as well as the types of girth connections are shown in Fig. 7. The application of the various slips and connections are outlined in Table 4. The end slip may be used wherever *S* slips are recommended. Where drive slips are used the end slip may be applied on the narrow side of the duct and only the drive slips on the

maximum side. Ducts 25 to 30 in. in size should be reinforced between the joints, but not necessarily at the joint. Ducts 31 in. and up should be reinforced at the joint and between the joints; if drive slips are used the angles are usually riveted to the duct about 2 in. from the slips. It is good practice to cross-break or kink all flat surfaces to prevent vibration or buckling due to the air flow and accompanying variations in internal

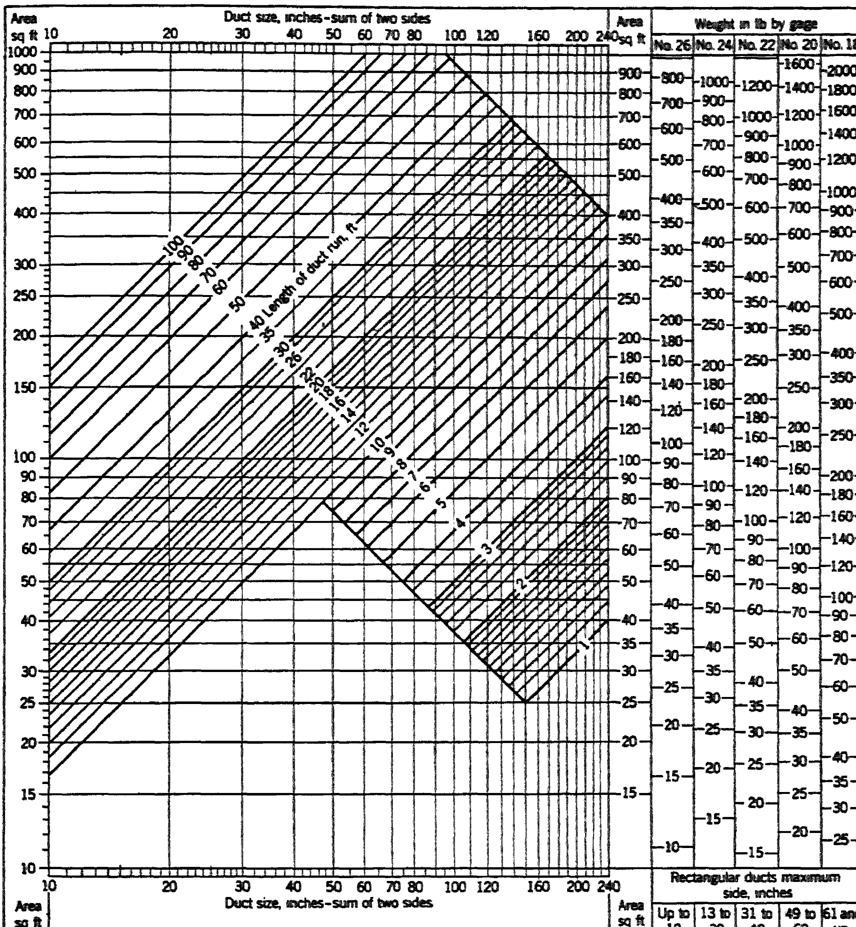


FIG. 8. AREA AND WEIGHT OF RECTANGULAR SHEET METAL DUCTS

pressure. Round ducts are sometimes swedged 1.5 in. from the ends so that the larger end will butt against the swedge and are held in place with sheet metal screws. Where swedges are not used it is general practice to paste the joint with asbestos paper to insure a tight joint.

The construction of elbows and changes of shape cannot be definitely outlined because of the varied conditions encountered in the field, but in

general long radius elbows and gradual changes in shape tend to maintain uniform velocities accompanied by decreased turbulence, lower resistance and a minimum of noise.

Heavy canvas connections are recommended on both the inlet and outlet to all fans. The fan discharge connections shown in Fig. 7 are marked *good*, *fair*, and *poor* in the order of the amount of turbulence produced. An inspection of the heater connections shown in Fig. 7 will readily show that uniform velocity through the heater cannot be expected in the diagram noted *poor*. When obstructions cannot be avoided, the duct area should never be decreased more than 10 per cent and then a *streamlined* collar should be used. Larger obstructions require an increase in the duct size in order to maintain as nearly uniform velocity as possible. Branch take-offs should always be arranged to cut or slice into the air stream in order to reduce as far as possible the losses in velocity head.

The recommended gages for sheet metal duct construction are given in Table 4. Weights of sheet metal per square foot of surface for different gages are given in Table 5. The weights of various gages and the areas for any length of run of rectangular sheet metal ducts may also be determined from Fig. 8. The bottom scale represents the sum of the two sides of the duct and the oblique lines give the length of run in feet. Proceeding horizontally to the right from the intersection of vertical and oblique lines on the chart, the area of the duct may be determined in the first vertical scale. The scales to the right give the weights of the duct run for different gages of metal. In calculating the weights of duct, it is considered good practice to allow 20 per cent additional for weights of joints and bracings. Various weights and thicknesses of standard copper sheets will be found in Table 6.

REFERENCES

Method of Determining Rectangular Equivalents and Weights of Ducts, by Peter Franck (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1940).

The Flow of Liquids, by W. H. McAdams (*Refrigerating Engineering*, February, 1925, p. 279).

A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler (*A.S.M.E. Transactions*, Hydraulics Section, August 31, 1933, p. 7).

Air Conditioning and Engineering, American Blower Corp.

Fan Engineering, Buffalo Forge Co.

Heat Power Engineering, by Barnard, Ellenwood, and Hirshfeld, Part III.

Mechanical Engineers' Handbook, by Lionel S. Marks, McGraw-Hill Book Co.

Chapter 32

SOUND CONTROL

Decibel Defined, Apparatus for Measuring Noise, Problem of Sound Control, Acceptable Noise Levels, Controlling Vibration from Machine Mountings, Controlling Noise through Room Wall Surfaces, Noise Transmitted Through Ducts, Duct Lining Formula

IN ventilating and air conditioning a building or a room, the effect of the mechanical system employed must be considered on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is not assumed that the ventilating and air conditioning engineer will attempt to improve the acoustics of the space that is being conditioned, but the designer should have at least enough fundamental knowledge of the acoustical effects of the system which is being designed to be sure that no damaging effects occur to the existing acoustical properties. It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

By a recently adopted international standard, two terms are used for noise measurement. The *decibel* (db) is the physical unit for expressing intensity or pressure levels. The *phon* is the unit of loudness level. The loudness level, in phons, of any sound is by definition equal to the intensity level in decibels of a thousand cycle tone which sounds equally loud. The decibel is defined by the relation $N = 10 \log_{10} \frac{I_1}{I_0}$, where N is the number of decibels by which the intensity flux I_1 exceeds the intensity flux I_0. The intensity flux is the measure of the energy contained in a sound wave and is defined in terms of micro-watts per square centimeter of wave front in a freely traveling plane wave. It is usually more convenient to select an arbitary reference intensity for I_0 and express all other intensities in terms of decibels above that level. For this purpose a reference intensity of 10^{-16} watts per square centimeter has been selected. This intensity is slightly less than the threshold of audibility

for the average ear at a frequency of 1,000 cycles per second. This reference level also corresponds to a pressure of 0.0002 dynes per square centimeter.

A stated sound level in decibels, unless otherwise defined, will thus be related to a threshold of 10^{-16} watts. For example, a level of 60 db above this reference threshold is 10^{-10} watts. In a similar manner, when sound measurements are given in actual intensity or energy units, they can be converted to decibels by this relation.

Since the decibel is a ratio, it can only be employed when related to a reference threshold level as given. Noise levels, which vary with frequency as well as intensity, must not only be related to this reference threshold level, but also to a reference frequency, which is taken as 1000 cycles. These terms and procedures may be found in Tentative Standards[1] published by the *American Standards Association.*

APPARATUS FOR MEASURING NOISE

Since the relative loudness to the ear, rather than the actual physical intensity, is the quantity in which engineers are usually interested, it has been found necessary to allow for the varying sensitivity of the ear at different frequencies in designing noise measuring equipment. The most satisfactory method of measuring noise is by means of a sound level meter which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels. This meter is calibrated to give readings above the standard reference level and usually contains a weighing network to make it less sensitive at those frequencies where the ear is less sensitive. For complete specifications relative to the approved type of sound level meters refer to the information[2] published by the *American Standards Association.*

GENERAL PROBLEM OF SOUND CONTROL

As previously stated, the function of the ventilating and air conditioning engineer is to add no acoustical hazard to the conditions already present in the room or building and the problem can be stated as:

a. To determine the noise level existing without the equipment.

b. To ascertain the noise level which would exist if the equipment were installed without sound control.

c. To provide as a part of the installation sufficient sound control appliances to reduce the noise level substantially to that found in (*a*).

To accomplish this the engineer should have information of three kinds:

1. A knowledge of the noise levels currently considered acceptable in various rooms in order that he may have a basis on which to proceed.

2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment.

3. A knowledge of how, when necessary, to vary and control the noise level between the equipment and the conditioned space.

In addition, the engineer should have information available to deal with

[1]American Tentative Standards for Noise Measurement, *American Standards Association.*

[2]American Tentative Standards for Sound Level Meters for Measurement of Noise and Other Sounds, *American Standards Association.*

noises which may enter the room due to openings made into it to accommodate the equipment, such as cross talk between rooms connected with common ducts and noise transmitted to portions of duct system outside the conditioned space and through to its interior.

While the general problem may be logically outlined and the items of knowledge necessary to its solution can be listed, the available information at present is lacking in certain respects. However, attention may be directed to that information which is currently available, and to furthermore outline a solution of the noise problem based on these data.

ACCEPTABLE NOISE LEVELS

Measurements of noise levels have been observed by several investigators in various rooms and locations. The information compiled in Table 1 is based on these data, which represent the best opinion on the subject now available. All levels are given in decibels above a reference threshold of 10^{-16} watts (corresponding to a pressure of 0.0002—dynes per square centimeter). Minimum, representative, and maximum levels

TABLE 1. TYPICAL NOISE LEVELS

Rooms	Noise Level in Decibels to be Anticipated		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc	33	40	48
Theaters, Legitimate	25	30	35
Theaters, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	30	38	45
Private Offices, Acoustically Untreated	35	43	50
General Offices	50	60	70
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Buildings, Court Houses, Post Offices, etc	45	55	60
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	65	77	90
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[a]	70	80
Pullman Car	55[a]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	80	85	100

[a]For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

are given for each application. These values are intended to indicate the variation which may be expected in different locations of the same type, but not the time variation which may be expected in each location.

The values shown in Table 1 are typical of those found currently in existing spaces. They are, however, the noise levels of the room and not the noise levels of the ventilating or air conditioning equipment. If the noise level at the room of the equipment is kept at the levels shown in the table the equipment will not add to the acoustical hazard existing without it, provided the equipment noise is heard alone, but if both are heard together the total noise level in the room will be increased about 3 db. This is usually considered an acceptable result.

In some cases it is desirable to keep the equipment noise level at the room at such a value that it actually will not increase the noise level in the room to any measureable degree. This can usually be accomplished if the equipment noise at the room can be kept 10 db below the noise level shown in the table.

NOISE CREATED BY EQUIPMENT

Information concerning the noise levels created by ventilating and air conditioning equipment such as fans, motors, air washers, and similar items is not yet on a basis which permits tabular presentation although certain manufacturers are prepared to offer such data and do state the noise producing properties of their products.

Absence of this information makes it necessary to resort to indirect means in solving certain problems and also prevents a direct logical solution.

KINDS OF NOISE

To solve a sound problem of this type it is desirable to consider separately the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source and places the emphasis on ascertaining the level at the point where the sound enters the room rather than on its point of origin.

The noise introduced into a room or building by ventilating or air conditioning equipment may be divided into two kinds depending on how it reaches the room as:

1. Noise transmitted through the building construction.
2. Noise transmitted through the ducts.

It is convenient to further sub-divide these two methods of delivery as:

1. Noise transmitted through the building construction.
   a. From machine mountings as vibration.
   b. From equipment through room wall surfaces.

2. Noise transmitted through the ducts.
   a. From equipment such as sprays, fans, etc.
   b. From outside, and transmitted through duct walls into air stream.
   c. From air current, including eddying noises.
   d. Cross talk and cross noises between rooms connected by the same duct system.
   e. Noise produced by the grilles.

The next step in the solution of this problem is to present data and discuss methods whereby solutions to the noise problem can be obtained when the allowable room noise level and the path through which the noise reaches the room are known.

NOISE THROUGH BUILDING CONSTRUCTION

It is impossible to select ventilating equipment which will operate without producing some mechanical noise, and since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building itself to such a degree as to make noisy conditions in the rooms which are to be air conditioned.

Controlling Vibration From Machine Mountings

Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations but these mountings must be carefully designed so that they will actually reduce the contact between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if it is improperly engineered, may actually increase the contact between the equipment and the floor upon which it is supported. In general, the flexible material should be loaded as heavily as possible without exceeding its elastic limit[3].

Where the mechanical resistance of the insulating pad is small, the ratio of the vibratory force communicated to the floor of the foundation with the machine resting upon the pad, and with the machine resting directly upon the floor, is given by Equation 1.

$$\tau = \frac{1}{\dfrac{n^2}{n_0^2} - 1} \tag{1}$$

where

τ = the so-called *transmissibility* of the support.

c = the compliance (that is, the reciprocal of the force constant).

m = the mass of the machine to be insulated.

n = the frequency of vibration generated by the machine which is to be insulated, such as the commutation frequency of a motor or the blade frequency of a fan.

n_0 = the natural frequency of the machine upon the elastic pad, expressed as,

$$n_0 = \frac{1}{2\pi}\sqrt{\frac{1}{mc}}.$$

---

[3]C. R. Soderberg (*The Electric Journal*, January, 1924), and succeeding articles. See also V. O. Knudsen, (*Physical Review*, Vol. 32, 1928, p. 324), and A. L. Kimball (*Journal Acoustical Society of America*, Vol. 2, 1930, p. 297).

In most cases of design of resilient machine mounting the effect of frictional resistance is small, and Equation 1 may be used. In such cases it is only necessary to know the natural frequency of the elastic pad or platform used under the desired loading and the transmissibility for any vibrational frequency of the machine may be obtained. However, this formula gives the theoretical maximum insulation which may be obtained and should be used with a liberal factor of safety. (A factor of 2 is common practice when the ratio is converted to decibels.)

If the pad is to be of any value in the prevention of solid-borne vibrations, the value of transmissibility must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be established, and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. It is necessary, therefore, that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 50 vibrations per second, and for best results preferably 20.

When the forced frequency is low, it is frequently impossible to insulate for the fundamental forced frequency due to connecting pipe work and other relevant factors. In cases of this kind an effective installation of sound insulation may be obtained with a mounting which functions far above the fundamental forced frequency. For example, a compressor operating at 500 rpm has a forced frequency of 8.3 vibrations per second. By designing a mounting having a natural frequency of 20 to 25 vibrations per second, it is possible to isolate practically all of the noise.

Controlling Noise Through Room Wall Surfaces

The ventilating equipment is usually housed in a separate room where the noise produced by the mechanical operation of the equipment can be isolated from the rest of the building. If the vibration of the machinery is absorbed by flexible mounting and is not transmitted to the building, the only noise to be eliminated by the walls of the room will be the air-borne mechanical noise. Acoustical measurements on average brick, tile, lath, and plaster walls indicate that the usual wall of these types is sufficient to satisfactorily attenuate this air-borne mechanical noise[4].

Attention should be given to the equipment room door, since this door may leak badly and allow sound to escape into parts of the building which should be quiet. Where the equipment noise is particularly severe, double doors should be used and in all cases, the doors of the equipment room should be fitted with tight thresholds and weather-stripping. The door itself may transmit considerable sound if it is thin but it will not transmit a tenth as much as will be transmitted by a ¼-in. crack between the door and the threshold.

---

[4]Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1932, p. 211).

In cases where the equipment noise is extraordinarily high, it may be necessary to treat acoustically the walls and ceiling of the equipment room. If the equipment room is not entirely closed, partition walls may be necessary.

NOISE TRANSMITTED THROUGH THE DUCTS

After noise reaches the air stream in the ducts it can be controlled by lining the ducts on the inside with a sufficient quantity of sound absorbing material. Lagging material of similar characteristics placed on the outside of ducts serves to prevent noise originating outside the ducts being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined some of the mechanical noise from the equipment room air may be transmitted through the wall of the duct, thus reaching the air stream and be carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material such as cork, pipe covering or other sound damping material to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Measurements in one laboratory have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board. Inside lining material used in the case previously mentioned would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream. Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

Calculating Amount of Duct Lining

To use the lining effectively it must be properly located, well installed and be applied in sufficient quantity to reduce the noise level of the air stream to the level desired. It has been shown theoretically[5] that the reduction, in decibels per linear foot, of sound transmitted through a duct lined with sound absorbing material is related in a rather complicated manner to the size and shape of the duct, to the frequency of the sound, and to the sound absorbing characteristics of the lining. Experimental evidence likewise indicates that there is no simple formula involving the above variables which will apply accurately to all cases.

[5]Sound Propagation in Ducts Lined with Absorbing Materials, by L. J. Sivian (*Journal Acoustical Society of America*, Vol. 9, p. 135, 1937).

One series of experiments[6] made on a commonly used type of duct lining material (1 in. rock wool sheet) has shown that, subject to certain restrictions, the reduction of single-frequency sounds may be expressed by the approximate Equation 2.

$$R = 12.6\, L \frac{P}{A}\, a^{1.4} \tag{2}$$

where

R = reduction, in decibels.
L = length of lined duct, feet.
P = perimeter of duct, inches.
A = cross sectional area of duct, square inches.
a = absorption coefficient of lining.

This formula is accurate within plus or minus 10 per cent for duct sizes ranging from 9 x 9 in. to 18 x 18 in., for cross-sectional dimension ratios of 1:1 to 2:1, for frequencies between 256 and 2048 cycles, and for absorption coefficients between 0.20 and 0.80. In Table 2, the absorption coefficients at different frequencies of a material of the above mentioned type are listed, together with the corresponding values of Equation 2.

TABLE 2. DECIBEL REDUCTION FORMULAE FOR TYPICAL DUCT LINING MATERIAL

FREQUENCY	ABSORPTION COEFFICIENT	REDUCTION, DB
256	0.37	3.0 L P/A
512	0.69	7.5 L P/A
1024	0.78	9.5 L P/A
2048	0.78	9.5 L P/A

Results of other experiments indicate, however, that Equation 2 may be in error when applied to other types of duct lining material and to duct sizes and shapes outside of the range specified. An empirically derived chart[7] representing the average experimental data on a number of different types of materials including the rock wool sheet mentioned as applicable to Equation 2 is shown in Fig. 1. Since individual materials vary somewhat, the curves in Fig. 1 are only given as representing the best available averages for duct sizes of square cross-sections from 6 x 6 in. to 48 x 48 in. As an illustration, the dotted lines in the chart show values calculated from Equation 2 which indicate that the slope for this particular material is somewhat different than from the average curves. The curves in Fig. 1, as well as Equation 2, show that the reduction in decibels is directly proportional to the length of duct lined, and that the larger the duct the greater will be the length which must be lined in order to obtain a given noise reduction.

From Table 2 it will be noted that the noise reduction varies to a considerable extent with the frequency of the sound. In calculating noise reduction, therefore, consideration should be given both to the comparative efficiency of the duct lining material at different frequencies,

[6]The Absorption of Noise in Ventilating Ducts, by Hale J. Sabine (*Journal Acoustical Society of America*, Vol. 12, p. 53, 1940).

[7]The Prediction of Noise Levels from Mechanical Equipment, by J. S. Parkinson (*Heating and Ventilating*, March, 1939, pp. 23-26).

and to the frequency distribution of the noise to be quieted. In the case of fan noise, it is recommended that calculations be based on the frequency 256 cycles, since most of the noise energy is in the region of this frequency. In quieting noise due to air turbulence and eddy currents, in which the high frequencies predominate, the frequency 1024 cycles should be used.

It should be noted that the lining should be installed at or near the outlet, in order to reduce effectively all sounds which may be generated in the system up to this point. The installation of lining near the outlets rather than near the fan is also more economical, because a greater noise reduction per square foot of lining material can be obtained in the smaller ducts leading to the outlets.

Estimating Required Noise Reductions

The amount of noise reduction which is required in any given case will depend both on the noise level in the room with the ventilating equip-

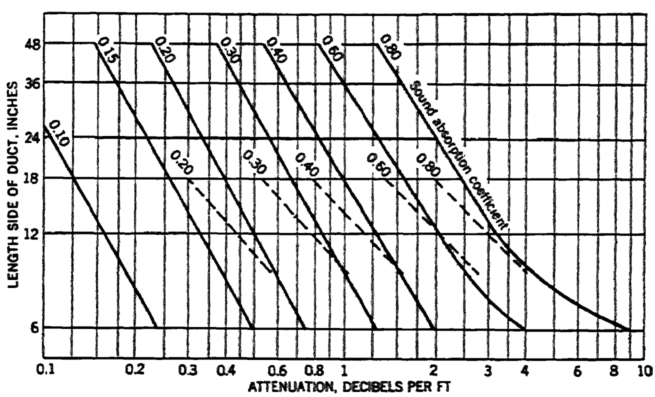


FIG. 1. SOUND ATTENUATION FOR VARIOUS ABSORBING DUCT LINERS

ment shut off, and on the noise level set up in the room by the equipment. Since in most cases it is difficult or impossible to determine these levels accurately, it is best to estimate the required noise reduction on the basis of experience and to allow a liberal safety factor. A guide for this purpose is shown in Table 3, which applies to fan noise. Attention is specifically called to its empirical nature and to the necessity of exercising judgment in applying it. When properly used with Table 1 the values in Table 3 provide a solution which may be both useful and simple. It is important to understand that the reductions referred to in this chart are based on the *average* noise levels set up in the room by the ventilating or air conditioning equipment. In the case of a piece of equipment which generates a noise level of 95 db, when the noise is measured immediately next to the machine, there might be a reduction of 15 db in passing through the duct, and a further difference of 15 db[8] between the noise at the outlet supply

[8]The drop between the level at the grille and the average level in the room will be governed by the absorbing power of the room. This is expressed in sabines and is equal to $\Sigma\, a\, s$ where a is the absorption coefficient of the surface and s the area of that surface. The total absorbing power of the room is thus the summation of the absorbing power of the various room surfaces. For a discussion of the effect of the room absorption see Loc. Cit. Note 7.

grille and the average level in the room, leaving an effective level of 65 db in the room. Reductions of noise level ranging from 5 to 25 db through duct systems have been encountered without the use of sound absorbing linings and the drop from supply opening to average room level may vary from 5 to 20 db.

To determine whether to use the *noisy*, *average*, or *quiet* column in Table 3, in forming an estimate of the relative amount of noise generated by the system, the length of the untreated duct system and the number of bends or elbows or splitters should be considered, since the longer and the more complex the system, the more reduction of noise level will occur before the sound reaches the room grilles. Also the sound absorbing power of the room should be taken into account, since in rooms where there is a great deal of absorptive material, such as rugs, draperies, curtains and furniture, there will be a higher loss between the outlet grille noise and the average room level. The ventilating engineer will have to judge whether the conditions deviate from the typical.

Table 3. Data for Determining Required Reduction of Fan Noise in Decibels

Room Noise Level, db	Reduction Required, db		
	Equipment		
	Noisy	Average	Quiet
15	60	50	40
25	50	40	30
35	40	30	20
45	30	20	10
55	20	10	0
65	10	0	0
75	0	0	0

Manufacturers' ratings on equipment should be considered in connection with the foregoing discussion. The quantity determined involves the noise level which will be produced in the room and the manufacturer's method of rating must be considered before allowances previously mentioned are accepted.

Further discussion of factors affecting equipment noise is given in Chapter 29, Fans.

Predicting Noise Levels

To use Table 3, proceed by consulting Table 1 and determine the probable noise level already existing in the room, and, as suggested, assume that this level is satisfactory for current practice. This gives a noise level in decibels and from Table 3 determine the value of the required noise reduction in the column corresponding to the noisiness of the equipment.

Example 1. A 10 x 20 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate average fan noise satisfactorily, using a lining material of a type to which Equation 2 applies, and having an absorption coefficient of 0.40 at 256 cycles.

Solution: From Table 1 the noise level in this office will be 35 db. From Table 3 a reduction of 30 db will be required.

Transposing Equation 2,

$$L = \frac{R}{12.6 \frac{P}{A} a^{1.4}} = \frac{30}{12.6 \times \frac{60}{200} \times 0.40^{1.4}} = 29.6 \text{ ft}$$

The sound absorbent properties of duct lining are extremely important and materials which have coefficients as high as possible should be used. This is particularly true of the coefficients at the low frequencies. Only certain sound absorbent materials among those listed in various publications[9] will be found to be suitable for duct lining. In addition to a high sound absorbent coefficient a duct lining material should have a low surface coefficient of friction, high resistance to moisture absorption, and should be fireproof and vermin proof. A number of building codes now specify that any sound absorbent material used for duct lining shall have no fire hazard. There are no existing specifications on moisture

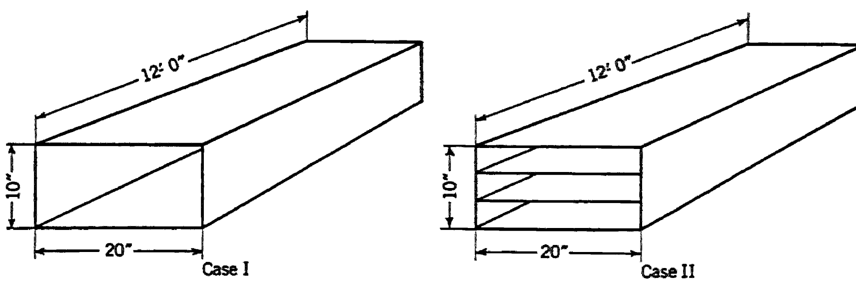


FIG. 2. DIAGRAM OF BRANCH DUCT TREATMENT WHERE LENGTH IS INSUFFICIENT FOR ADEQUATE ABSORPTION

resistance but the manufacturer should be required to show that the material will not absorb sufficient moisture to cause deterioration or to decrease the sound absorbing efficiency.

If, as is often the case, the length of duct from the main duct to a grille is shorter than the length of lining indicated by the calculations, this duct may be sub-divided into smaller ducts, as shown in Fig. 2. The increase in noise reduction thus obtained may be calculated from Equation 3, providing the splitters are installed parallel to the long side of the duct:

$$R_s = R_o \frac{a + bn}{a + b} \tag{3}$$

where

R_s = reduction with splitters, decibels.
R_o = reduction in same length of duct, without splitters, decibels.
a = dimension of short side of duct, inches or feet.
b = dimension of long side of duct, inches or feet.
n = number of channels formed by splitters.

[9]For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Materials Association*, 919 No. Michigan Ave., Chicago, Ill.

Example 2. Assume that the duct of Example 1 is only 12 ft long, and that a 30 db reduction is required in this length.

Solution:

Case 1. (No splitters).

From Equation 2,

$$R_o = 12.6 \times 12 \times \frac{60}{200} \times 0.40^{1.4} = 12.6 \text{ db}$$

Case 2. (Two splitters, three channels).

From Equation 3,

$$R_s = 12.6 \times \frac{10 + (20 \times 3)}{10 + 20} = 29.6 \text{ db}$$

General Suggestions

In some instances where high velocity air is used, a considerable amount of whistle is generated at the grille. This noise is obviously produced after the air leaves the duct and there is no treatment which can be installed in the duct that will reduce this noise. The engineer must take into consideration the type of grille which he intends to use and provide sufficient grille area so that the velocity through the grille is reduced to a point where the grille is not too noisy. Further discussion of grille noise is given in Chapter 30, Air Distribution.

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be sub-divided to provide ample acoustical treatment.

Very often in ventilating duct work the engineer feels that it will not be necessary to line ducts if the sound is traveling against the airflow. This, however, is untrue since sound travels so much more rapidly than does the air in even high velocity systems, that it will travel as easily against the airflow as it does with it.

Sounds which are low in pitch are much harder to eliminate from a duct system than sound which is high in pitch, consequently equipment which produces low pitched sounds should be avoided as much as possible.

Chapter 33

AUTOMATIC CONTROL

Purpose of Automatic Control, Types of Control, Central Fan Systems, Limit Controls, Static Pressure Control, Unit Systems, Control of Automatic Fuel Appliances, Residential Control Systems, Control of Refrigeration Equipment

THIS chapter is prepared with the purpose of acquainting the engineer with the principles underlying the use of automatic control, the general types and varieties of control equipment available and their application.

Automatic control, properly applied to heating, ventilating and air conditioning systems, makes possible the maintenance of desired conditions with maximum operating economy. A properly designed and complete control system has the ability to interlock and coordinate the various functions of heating, ventilating and air conditioning in a manner impossible to accomplish with manual regulation.

Automatic control is an integral and essential part of a heating, ventilating or air conditioning installation and cannot be regarded as an accessory. In order to insure satisfactory results, the control should be designed with and incorporated in the heating, ventilating or air conditioning system. The control equipment should be given careful consideration in the planning of any installation in order that the entire system may operate together with satisfactory results.

In order that proper selection and application of controlling devices may be made it is important that a broad understanding exist as to the types of control available and their principles of operation. Improper selection and application of control equipment will result in unsatisfactory and inefficient operation. Specific control devices and systems are described in the *Catalog Data Section*.

PURPOSE OF AUTOMATIC CONTROL

Automatic control is normally applied to heating, ventilating or air conditioning systems:

1. To insure the maintenance of certain desired or required conditions of temperature, pressure, humidity, air motion or air distribution.

2. To serve a safety function, limiting pressures or temperatures within predetermined points, or preventing the operation of mechanical equipment unless it may function without hazard.

3. To produce economical results and thereby insure operation of the system at a minimum of expense.

TYPES OF AUTOMATIC CONTROL

Operating Medium or Source of Power Supply

Automatic control systems may be classified in three broad groups based upon their primary operating media or sources of power, as follows:

1. *Electric Control Systems.* In such control systems the primary medium utilized to provide for the operation is electricity, and the basic function of these controls consists of switching or otherwise adjusting electric circuits to govern electric motors, relays or solenoids. The individual units of this type of system are interconnected by line voltage or low voltage wiring, and this wiring serves to complete the circuits carrying the commands of the controllers to the controlled valves or damper motors.

2. *Pneumatic Control Systems.* In these systems the source of power for operation is compressed air, furnished by one or more centrally located compressors, and distributed in special piping to the controlling and controlled devices. The pressure is varied by the controlling instruments and this variation operates the controlled devices, which may be valves, damper motors, relays or electric switches.

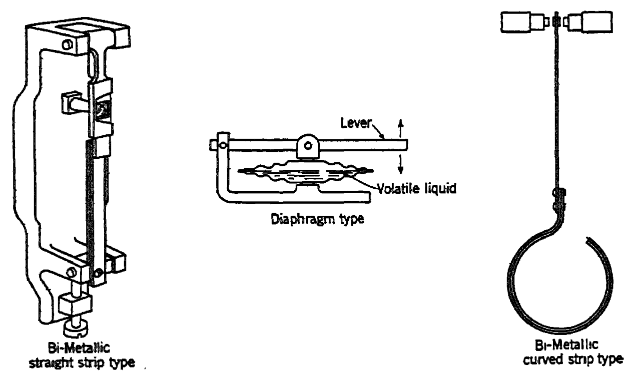


FIG. 1. TYPICAL THERMOSTATIC ELEMENTS

3. *Self-Contained Control Systems.* In self-contained control systems, the primary source of operation is the vapor pressure of a volatile liquid in the closed thermal system of the controller, which is increased or decreased in direct proportion to variation of the temperature in the controlled medium. These pressure changes are transmitted directly to the control valve or damper motor. Applications consist of valves or dampers to regulate the flow of heating or cooling media to coils, radiators, or liquid tanks, as determined by the controller element.

Typical thermostatic elements are shown in Fig. 1.

Motion of Controlled Equipment

Automatic control equipment can also be classified into two general types with respect to the characteristics of the motion imparted by the controls to the controlled equipment, such as two position or positive-acting control and modulating or graduated-action control.

In any control system it is necessary to choose the type of equipment whose characteristics permit the type of control operation desired and in many cases both types of control are used in the same system to best meet various requirements.

1. *Two Position or Positive-acting Control.* This type of control operates positively between two positions such as *on and off* or *open and closed* with no intermediate positions or degrees of motion between the two extremes of operation. A simple thermostat which starts and stops an oil burner or a unit heater motor is an example of this type. As applied to a valve or a damper, the action of the controlling device would serve to fully open or fully close the valve or damper.

In some applications of this type of control, artificial heat is applied to the sensitive element of the room thermostat at the same time that heat is being added to the space under the control of the thermostat in order to produce more frequent operation. This usually results in more accurate control of the heat source.

2. *Modulating or Intermediate Control.* This type of control causes motion in the controlled device in proportion to motion caused in the controller by fractional degree variations in the medium to which the controller is responsive. After a fractional change has been measured at the controller and has effected a new position of the valve or damper in proportion to the amount of such change, the system stands by awaiting further change at the controller before any additional motion occurs. The extent of the motion is limited only by the limits of the controller and by the intensity of the change of conditions as measured. With this type of control, the damper or control valve may be operated in intermediate positions between its extreme limits in order to properly modulate or proportion the flow of air, steam or water, reacting with changes of conditions at the controller. Various modifications of this type of control are available, designed to meet special requirements and conditions, all based on operation of the controlled equipment in intermediate positions.

These controlling devices may be made to operate relatively faster or slower for any change in condition of the fluid being controlled. For example, a thermostat modulating a damper may move it from one extreme to the other in one degree temperature change, or many degrees change may be required to produce this same action. This characteristic is sometimes called the *sensitivity* of an instrument. The sensitivity may be fixed, or adjustable.

This type of control motion cannot be used on valves of one-pipe steam systems as the partial opening of the valves will not permit the condensate to escape against the flow of incoming steam. Where this type of control is used to control the flow of steam to a heater coil of a fan system which is in the direct path of untempered outdoor air at temperatures below freezing, care should be taken that the control point and operating characteristics of the regulator are such that the valve is open far enough, at air temperatures below freezing, to prevent the freezing of condensate in any part of the coil.

Control for Individual Rooms and Small Buildings

Control systems vary considerably with the type, size and occupancy of the building, and with the heating or cooling system, humidity supplying

equipment and ventilating means available for control. In the following paragraphs the general requirements of two types of control are discussed.

1. *Individual Room Control.* The most accurate and flexible form of control for any structure is that calling for the regulation of each individual room by control equipment reacting to conditions in that room only. Such control necessitates a thermostat in each room, located to properly measure the conditions of the room, controlling the radiator, unit heater, damper, unit air conditioner or other heating or cooling source, for that room. This arrangement permits the maintenance of any desired conditions in any room, entirely independent of any other room. In the case of large rooms, where one thermostat location will not serve to properly measure the conditions throughout the room, and where two or more sources of heating or cooling are provided in the room, additional thermostats may be used, each controlling its respective section. This form of control, due primarily to the number of control devices required over the entire building, normally is the most expensive. However, where maximum flexibility and the most accurate control are desired, individual room control should be used.

Room thermostats are available for various functions. *Dual* thermostats operate heating devices at normal temperatures during periods of normal occupancy but at lower temperatures, for economy, at other times. The change-over may be by clock or manual switches, and one or any number of thermostats may be on a single switch. *Summer-Winter* thermostats, as described for All Year Central Fan Systems, are used for reversing the operation of certain dampers or valves to make them function for both heating and cooling.

One precaution to be observed in the location of room instruments is to make sure that each is in control of all the heating and cooling devices that affect its temperature, except where two thermostats are used to operate at different temperatures.

2. *Single Thermostat Control.* A great majority of the buildings under automatic control have the comfort temperature maintained by a single thermostat operating directly on the source of heat or cooling for the entire building. In average size residences and in other small buildings, it is possible to select a thermostat location which will give entirely satisfactory results throughout the structure. This location must be one which truly represents average conditions and one which will not have unusual temperature effects. For example, a thermostat near an outside door may function improperly when the door is open. After the proper location is selected, the system is balanced to provide the proper temperature distribution.

Details of control by single thermostats will be found under the heading, Control of Automatic Fuel Appliances, in this chapter.

Zone Control

As the size of buildings increases, it becomes increasingly difficult to provide proper regulation for the entire structure from a single thermostat control. In such instances, where the advantages of individual room control are not obtainable by reason of its cost, an intermediate form of control system is available, commonly described as *zone control.* In this

form of control system a building is divided into areas or zones such that the general requirements and the general conditions through the areas are relatively constant as to exposure and occupancy, and then each zone is provided with control equipment which functions to regulate the conditions in that particular zone. As in the case of individual room control, each zone may be regulated to its own needs which may vary from the needs of other zones within the same structure.

The number of zones to be used is determined by several factors, such as:

1. Size of building.
2. Number and character of exposures.
3. Variation in occupancy or other inside conditions.
4. Cost of additional zones.

The greater the number of zones, the closer is the approach to the results and cost of individual room control. However, zone control has advantages even where individual room control is installed as it lightens the work of the room control. With room control, fewer zones are needed. In buildings of large floor area, it is usually desirable to have a separate zone for each exposure. If one wall is protected by an abutting building for half its height, two zones may be necessary. First floor conditions may vary enough from those of the rest of the building to justify a separate zone. In large buildings with several exposures toward any compass point, as occurs in wings and courts, all the northern exposures, for example, may be put on one zone control, or each north wall may have its own control. Court exposures are apt to be affected by surrounding walls and thus to require separate treatment.

In high buildings it is often important to consider zoning for stack or chimney effect in winter, caused by the difference in density between the warm air on the inside of a building and the colder air on the outside. Where the lower eight or ten stories are protected from winds by surrounding buildings, it may accentuate the need for zoning to correct the chimney effect, and on windy days there will be a marked difference in the heat requirements for the different horizontal sections at different elevations. An arrangement to provide for difference in heat requirement for exposure and chimney effect would give 12 zones; namely, north, east, south, and west lower, middle and top zones.

For steam heating systems the automatic control arrangement varies with the means of obtaining reduced temperatures. Some of the methods in common use are described in Chapter 13. From the control standpoint they are classified as follows:

1. Throttling steam to allow flow in proportion to the needs for heating.
2. Turning the steam on and off, leaving it on for longer or shorter periods as required.
3. Varying the pressure differential between supply and return lines, and varying the absolute pressures in both, so as to change the amount of steam passing through the radiators, due to the differences in pressure drop and due to the differences in volume per pound of steam.

The controlling thermostats may be inside or outside instruments or a combination of the two. Ordinary inside thermostats alone are likely to give disappointing results because an unusual condition at the thermostat upsets the whole zone, and because a slight temperature drop may allow

too much steam to pass before its heating effect reaches the thermostat. Therefore some device is needed to vary the flow in accordance with the weather. This may be a simple long range thermostat that restricts the flow as the weather moderates, or one that turns the steam off and on, on oftener and for longer periods in cold weather. One device is designed to directly control radiator temperatures at progressively lower points as the weather becomes warmer. Most outside thermostats have provision for sun and wind effect. They do not produce close control of indoor temperature, and are usually accompanied by hand switching devices for raising and lowering the control point, where individual room control is not included. They are, as stated previously, valuable adjuncts of room control.

For a hot water heating system, zone control consists of an outdoor thermostat varying the temperature of the water in accordance with the weather. This may be done by changing the amount of heat applied to the water, or by mixing hot water with recirculated water so as to produce the proper temperature. Inside zone thermostats may be used to correct improper action of weather thermostats, or, where only one outside instrument is used for a number of zones, to start and stop circulating pumps in accordance with the demand for heat in the various zones.

For both steam and hot water systems, zone control is primarily to reduce the general heating effect in moderate weather. Thus the term is used to describe a type of control system, though a building may have but a single zone.

In air conditioning systems, zone control may be applied to separate fan systems in different parts of a building or to two or more sections of the air distributing system from a single fan. The zone thermostat may be room type, or insertion type located in the return air duct from the zone. Where each zone has its own fan, the control may be the same as for an independent system. If one fan serves more than one zone, there will be heating and cooling coils for each, or a damper to mix air volumes of two temperatures to provide the proper conditions for the zone.

Zone control for an all-year air conditioning system presents problems that do not arise in either the heating or the cooling cycle alone. As a zone is normally selected for similarity of conditions, and the distribution of temperature effect to the various rooms adjusted so that one control point is sufficient, it is important that the similarity of conditions applies equally to heating and cooling. Two rooms that have like heating loads and that work well together in the heating season, may have entirely different cooling loads. This difficulty can be overcome by the use of sub-zones or individual room control where necessary.

CENTRAL FAN SYSTEMS

Central systems for air conditioning are described in Chapter 20. For explanation of the control problems for such systems, the various functions, such as heating, humidification, and cooling, are treated independently.

Ventilating Systems

A control system for a central fan ventilating system using all outdoor air and discharging air at a predetermined temperature is illustrated in

Fig. 2. Thermostat T_1 located in the outdoor air intake is set just above freezing, and controls valve V_1 on the first heating coil. This arrangement, where the valve is held completely open or closed to avoid danger of freezing, must be used where the coil is not specially designed for uniform steam distribution across its face. The by-pass damper around the heaters and the other two valves V_2 and V_3 are controlled by thermostat T_2 located in the discharge duct from the fan. When the temperature of the discharge air is too high, T_2 closes V_3 and V_2, gradually and in sequence, then if V_1 is open and supplying too much heat, T_2 opens the by-pass damper. The control of the damper and valves V_2 and V_3 must be gradual to prevent wide fluctuation in temperature.

In ventilating systems it is customary to supply air to the ventilated spaces at an inlet temperature approximately equal to the temperature maintained in the rooms. The radiators therefore are designed to take care of all the heat losses from the rooms and in order to maintain controlled room temperatures it is necessary to control the radiators independently of the ventilation control.

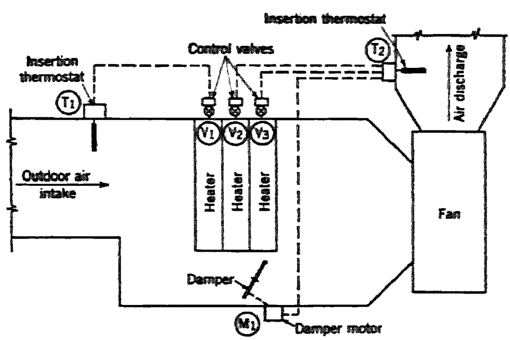


FIG. 2. CONTROL OF A CENTRAL SYSTEM FOR VENTILATION

In central fan systems, air washers are sometimes used and in such cases, due to the effect of temperatures on humidity, additional control is required. The heating coils are then divided, one or two at the inlet and usually two at the outlet, generally called preheaters and reheaters. There should be no by-pass under the former, because of the danger of a stratum of cold air freezing the water. To maintain relative humidity at a constant point a *dew-point* thermostat is inserted into the air stream between the two sets of coils, to control the preheaters. Cold air control of preheaters cannot be used because at temperatures just below freezing a standard heating coil, which will protect an air washer against freezing in zero weather, will give a too high dew-point temperature. Therefore, the one or two preheater coils must be controlled from the dew-point thermostat. This is preferably placed at the discharge side of the washer and set for about 40 or 45 F. As there is some cooling effect from the water, this provides a slightly higher temperature leaving the coils. In such cases, the throttled steam must be fairly uniformly distributed across the face of the coil, to prevent a stratum of cold air that would freeze water in the coil or in the washer.

Heating Cycle

Similar fan systems are used for heating, as well as for ventilating occupied spaces, by increasing the number of coils to four or five. Where they are all installed together the control remains the same as shown in Fig. 2, and the additional coils are controlled directly from a room thermostat which also causes T_2 to turn on full heat while the room is cool, but to function as described previously while the room is warm. This facilitates rapid heating of the room, after a vacant period.

An alternate plan is the use of a fan discharge thermostat whose control point can be automatically varied, and a room thermostat to reset it. Thus when the room is cold, air is delivered at a maximum temperature designed for rapid heating, and when the room is too warm, the air is kept as cool as can be safely introduced, or as the weather permits. The discharge temperature varies between these two extremes at the command of the room thermostat, until it finds the proper point for the existing conditions. This makes it unnecessary to vary the fan discharge thermostat manually to prevent overheating in moderate weather, or chilling in cold weather.

The heating coils are often separated into two groups, one at the suction side of the fan, controlled as shown in Fig. 2, and the other on the down stream side of T_2, controlled from the room. Control T_2 is then called the *tempered air* thermostat.

In all types of fan heating systems it is desirable to have the tempered air thermostat in the fan discharge where stratification has been broken up by the fan.

Where a fan system supplies heat to several rooms or zones that require separate treatment, the tempered air control can remain as in Fig. 2, and the variation can be supplied in any of the following ways:

1. By installing a separate duct to each zone, and using individual heating coils, each under control of its respective room thermostat.

2. By installing double chambers at the fan discharge, only one of which is supplied with additional heating coils. Individual room or zone ducts have mixing dampers which allow air to be taken from the warm air chamber, the tempered air chamber, or both, as demanded by their respective room thermostats. With this arrangement, precautions must be taken to prevent the warm air from being churned back and into the tempered air while a number of the mixing dampers are calling for the latter. If the warm air is controlled at a constant temperature under all conditions, the coils should be placed not less than 8 ft from the fan discharge and the dividing plate extended back several feet toward the fan. A good solution for the problem is to use an automatically adjusted thermostat in the warm air chamber, controlled by an outdoor thermostat so as to carry maximum warm air temperatures in the coldest weather and minimum in moderate weather.

3. By using a trunk duct, and varying the amounts of air delivered, by dampers at individual outlets or for the various zones. If a minimum amount of air is required for ventilation, the dampers must not close entirely. Thus to prevent over-heating, the trunk duct temperature must be varied according to the weather, as previously described. Also static pressure control may be needed. See a subsequent sub-head for a more detailed discussion of this subject.

Case 1 is illustrated in Fig. 3. Thermostat T in the fan discharge controls outside and return air through damper motors D_1 and D_2, face and by-pass dampers through damper motor D_3 and the steam supply to a heating coil through valve V_1. By having the face damper closed, and

the by-pass open, before steam is throttled, there can be no danger of freezing the coil. Thus the D_3 operation is completed before V_1 starts. However, the relationship between D_1 and D_2 controlling the amount of recirculation, and D_3 regulating the amount of steam heat added, depends on the design of the ventilation system. If the maximum amount of outside air is desired for ventilation, and the return air is used only during the heating-up period, D_1 and D_2 complete their operation to bring in all outside air before D_3 starts. On the other hand if greatest heating economy is desired and full outside air is to be used only to prevent overheating, D_3 completes its motion to close the face damper, before D_1 and D_2 start. Any relationship can be attained between these two extremes. Relay R prevents T from closing the outside damper completely, when a minimum of outdoor air is required during operation. Valves V_2 and V_3 control the steam supplied to booster coils for two zones, in accordance

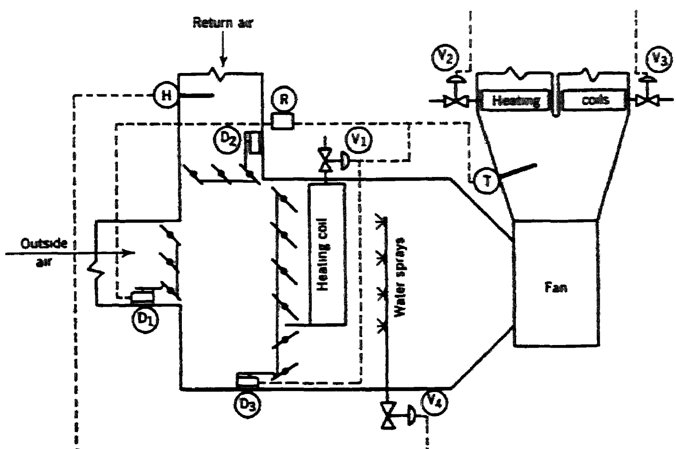


FIG. 3. CONTROL OF A CENTRAL SYSTEM FOR HEATING AND HUMIDIFICATION

with the requirements of room type zone thermostats, not shown in the diagram. Humidistat H, in the return air, regulates the amount of water supplied through V_4 to the spray heads.

Humidification

Humidification with air washers has been mentioned in connection with control of ventilating systems. Where ample room air change is provided, it is generally assumed that the dew-point of the conditioned space will soon equal that of the delivered air. This is due to exterior walls, especially glass, being less pervious to vapor flow than to heat transmission. With partial recirculation, dew-point control prevents over- as well as under-humidification, for ordinary installations.

Where air washers are not used, humidification may be accomplished by water sprays, preferably with heated water; by steam heated water in pans; or by steam jets, if their odors are not objectionable. Sub-atmospheric steam cannot be used, of course, for jets, and is not of much more value in coil heated pans. In all these cases the control is obtained by

humidistats, usually placed in rooms or in return air ducts. In ventilating systems the controlling instruments may be put in the fan discharge for results comparable with dew-point control. However, as hygroscopic elements are actuated by relative humidity they cannot be used with discharge temperatures that have been raised to supply heating effect.

Humidity control in cold weather is complicated by the danger of causing condensation or frost on windows and exterior walls, when otherwise desirable relative humidities are obtained. For satisfactory results in buildings that are not specially designed to prevent cold interior wall and glass surfaces, it is necessary to maintain lower humidities in very cold weather. This is done automatically either by an auxiliary humidistat mounted at a window to prevent condensation at that point, or by using a type of room or duct humidistat that is reset by an outdoor thermostat to maintain gradually drier conditions as the weather gets colder.

Cooling Cycle

Although central systems are occasionally used for cooling and dehumidifying only, the control features are essentially the same as for complete air conditioning systems. Where control of room conditions is obtained by varying the quantity of cooled air, as in a trunk duct system, individual room or zone thermostats operate volume dampers. It is customary to have these installed with a stop to prevent shutting off the air supply entirely, the reduction being from 40 to 60 per cent of the maximum delivery, depending on the design of the system. Later described control of the temperature of the air prevents over-cooling with the minimum supply, and the damper variation is normally sufficient to handle the distribution of the cooling effect throughout the area supplied by the fan or trunk duct.

In cases where systems and outlets are designed for particular velocities for proper room diffusion, volume dampers tend to produce undesirable results, by changing these velocities. Partially closed dampers reduce volumes in their ducts and increase volumes elsewhere. Trouble from too little air can be reduced by having the dampers close off, in one way or another, a part of the grille openings, thus maintaining approximately the same velocity through a smaller grille area. Trouble from increase of static pressure, due to reducing air volumes delivered, can be corrected by static pressure control, as described under a separate subheading in this chapter.

In installations where constant volumes of air are desired, and individual ducts are run to each room or zone, as shown in Figs. 3 and 5, of Chapter 20, air temperatures are varied as for the heating cycle, by room thermostats operating (1) mixing dampers which take air from either or both of two chambers, one of which has been cooled to the minimum temperature ever required; (2) booster cooling coils, one for each duct, in which the refrigerating medium can be turned on or off, or modulated; (3) individual by-pass dampers around booster cooling coils which are kept at a constant temperature; or (4) reheating coils which in times of light cooling load add heat to air that has been cooled to the minimum temperature required.

All these arrangements apply where one fan supplies more than one

room, or zone, and consequently the temperature-varying devices are downstream from the fan. The remainder of the control for the system is concerned with maintenance of conditions at the fan and is similar to what is used where a fan system is treated as a single zone.

Air washer cooling and dehumidification is commonly controlled by pumping the spray water through, or around a water cooler, with a dew-point thermostat operating a mixing valve which regulates the amount of water by-passing the cooler. An alternate scheme is to put cooling coils in the air washer spray or the pan, and to control the temperature of the coil. In both cases control of relative humidity is obtained by maintaining a constant dew-point temperature and thus a constant amount of water vapor per cubic foot of air handled. On account of this humidity factor, air leaving the washer must be reheated. As explained in Chapter 20, this is done, (1) by passing uncooled air around the washer[1] with thermostatic control of the proportion of uncooled air; (2) by adding heat by means of an automatically controlled coil; or (3) by allowing the room air to provide the heat by diffusion, in which case, still assuming a constant volume of air, the only means of dry-bulb control is the raising and lowering of the dew-point temperature, and hence the relative humidity.

Heat transfer surface coils, now more frequently used for cooling and dehumidification, are of either the direct expansion or cold water type. The former may be controlled by starting and stopping or unloading the compressor, by opening and closing a valve in the liquid line, by throttling the expansion valve, by throttling the suction line, or by raising and lowering the coil pressure, and temperature, through operation of a back pressure valve. The cold water type coils are controlled by valves to regulate the flow of water. They may throttle the flow, or they may be of the three-way type that allows a uniform flow but by-passes any necessary amount around the coil. Where well water pumps are operated only for cooling coils, control is added to stop them while cooling is not needed, but if they serve other purposes, they continue to run and the water is controlled by throttling valves.

The control with all types of coils may include a damper in an air by-pass[2] around the coils, with or without one over the face of the coils. If the installation is large enough to justify the use of two or more coils, side by side, the special air by-pass may be omitted and similar results obtained by closing the coils in sequence. The controlling instrument in all these cases is a thermostat in the room, return air, or fan discharge, whether the system serves one zone or several. In the latter case, a thermostat in the return air or in the fan discharge serves as a primary control, and the final control of room conditions is obtained with the zone thermostats.

Room or zone control in the cooling cycle is commonly provided by thermostats which operate at varying points depending on the weather. This takes care of the difference in optimum temperatures for the heating and cooling seasons and also of the objection to maintaining too high a differential between indoor and outdoor temperatures in hot weather.

[1]Patents exist covering the by-pass method.

[2]Loc. Cit. Note 1.

A thermostat sensing outdoor conditions is used to reset inside temperatures, raising them gradually to a point from 5 to 15 F below the highest outside temperature. This differential depends on the type of occupancy. Temperatures should be maintained so as to avoid too great a change for anyone entering or leaving. In large buildings, gradually lower temperatures at increasing distances from entrances and exits can be arranged. See Chapter 2 for general remarks about proper temperatures.

Except in the case of dehydrating systems, independent humidistatic control of dehumidification is seldom provided. Air washer systems as already described, are provided with dew-point thermostats. Cooling coils may be designed for proper proportion of sensible and latent heat removal so as to give satisfactory relative humidity when only the temperature is controlled. For a small installation, without by-pass or other reheat, a room thermostat and humidistat are sometimes arranged to provide cooling until both the temperature and humidity requirements

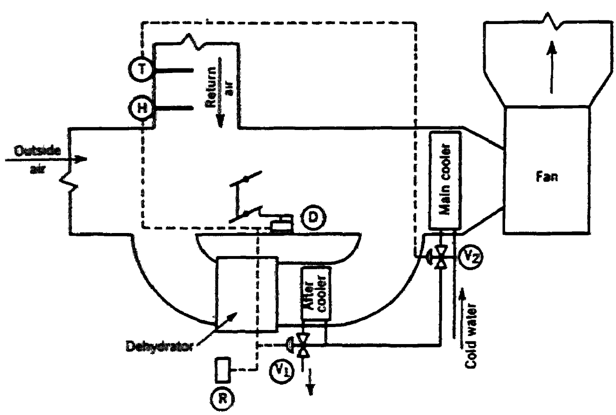


FIG. 4. CONTROL FOR A DEHYDRATING AND COOLING SYSTEM

have been satisfied, and a second thermostat is used to prevent excessive cooling by the humidistat. The cooling may be regulated by a combination of temperature and humidity that approaches *effective* temperature, by causing the relative humidity to gradually readjust the temperature control point, higher for dry air.

Control of Refrigeration. Room or duct conditions may start, stop and *unload* the refrigerant compressors directly, or may operate only at the evaporators. In either case other controlling instruments are used for the refrigeration, as described under the general heading, Control of Refrigeration Equipment.

Control of Direct Dehumidifiers—Dehydrators. Absorbent and adsorbent types of conditioning systems have the dehumidification controlled by room or return air humidistats. Since these processes are capable of producing relative humidities much below the desired point, only a portion of the air may be treated and a by-pass damper, controlled by the humidistat, used to vary this portion. The control of water cooling coils is similar to that previously described, except that the additional coil, used to extract

the sensible heat transformed from latent by the process, can use cooling water leaving another coil. That is, water leaves the main cooling coils at a low enough temperature to do the requisite cooling for the high temperature air. In order to have water available at both coils, the control valves at each are of the three-way type. As this allows free flow of water at all times, a normally closed valve can be installed in the water line and controlled by a thermostat varying the flow to maintain a suitable temperature. By connection to the fan motor circuit the valve can be kept closed while the system is not in use.

Some of these features are shown in Fig. 4. Humidistat H, on rising humidity, simultaneously starts the dehydrator and its fan through relay R, positions three-way valve V_1 to permit water to flow through the *aftercooler*, and closes damper D to increase the resistance in the main duct so as to reduce any tendency of the dehydrated air to short-circuit. Outside air and return air dampers, commonly used, are not shown. Their operation is as described for Fig. 3, except that for the summer cycle, the outside air is fully opened before V_2 turns on the main cooling

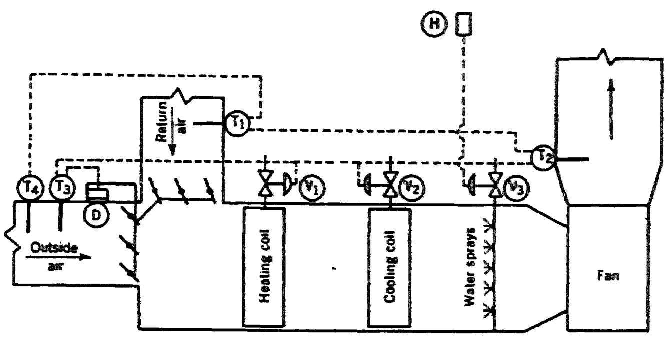


FIG. 5. CONTROL FOR AN ALL YEAR CENTRAL SYSTEM

coil, and a wet-bulb or similar thermostat in the intake cuts the outdoor air to a minimum when its wet-bulb temperature is greater than that of the return air.

All Year Systems

All year systems combine the features described for heating and cooling cycles, and have provisions for spring and fall conditions. Complete automatic control of all year systems incorporates an automatic change-over between the cooling and heating cycles. If the installation necessitates operation of a manual switch or other change-over device between the heating and cooling cycles, then the control system is semi-automatic. The full automatic change-over between cycles becomes particularly desirable in the early and late portions of the cooling and heating seasons, when heating and cooling may be required alternately.

For all year systems, a single thermostat may be used for both heating and cooling cycles, as shown in Fig. 5. In this diagram, T_2 regulates the amount of recirculation through damper motor D, the amount of steam by valve V_1 and the amount of chilled water by V_2. As the temperature rises, V_1 first operates completely to close off the steam; next, outdoor air

quantities are increased from a minimum, if the outdoor temperature as sensed by T_3 is low enough to provide cooling; and finally chilled water valve V_2 opens. Control T_2, however, operates, not at a constant temperature, but at a point varying from the minimum required in warm weather to the maximum required for heating. The variation is effected by T_1 in the return air, which raises and lowers the control point of T_2 until the proper return air temperature is obtained. During the heating and intermediate seasons, T_1 operates at a constant point, but in the cooling season it is readjusted by outdoor air thermostat T_4 to provide higher indoor temperatures. Room humidistat H opens valve V_3 on falling humidity to turn on the water sprays. As inside humidities in summer are normally higher than required in winter, the sprays are automatically kept closed.

Five diagrams of large central systems are shown in Chapter 20. The arrangement of coil and sprays diagrammed in Fig. 1 requires control as just described, except that if the cooling coil is of the direct expansion type, the refrigeration is controlled as explained in this chapter under

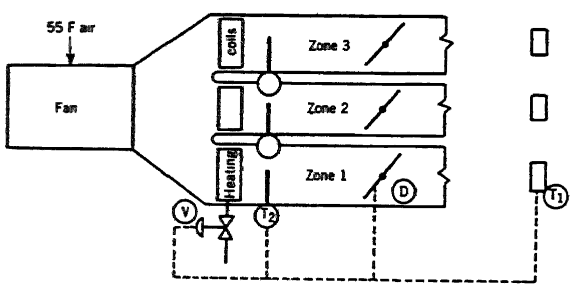


Fig. 6. All Year Zone Control with Booster Heating Coils and Volume Dampers

Cooling Cycle. In Fig. 2, a dew-point thermostat near the eliminator plates, on a rising temperature first turns off the preheater and then turns on the water cooler. A return air or fan discharge thermostat controls the reheater coil, and the return air and by-pass dampers. Assuming that the coil is not heated in summer, the by-pass damper is opened and the return air damper closed to provide reheat. In winter, provision must be made to keep the by-pass damper closed, or to reverse its operation to prevent by-passing the coil when heat is required.

The temperature at the primary fan in Fig. 4 is maintained 10 F or more below desired zone temperatures throughout the year, to allow correction of overheating in winter. In summer this setting is further reduced, to cut down the amount of outside air required for cooling and to provide sufficient dehumidification. The zone thermostats thus call for more return air for reheat.

Where the internal cooling load is great, an arrangement as shown in Fig. 6, of this chapter, has some advantages. Air entering the fan is controlled at about 55 F by operation of a steam valve and outside and return air dampers, so long as weather permits. In summer the refrigeration is turned on at a somewhat higher temperature, as required. Booster heating coils, low limit thermostats, volume dampers and room,

or return air, thermostats are installed for all zones, as shown. Volume dampers are adjusted with a minimum position that will supply sufficient air quantities for heating. While a zone is too cold, T_1 holds D in minimum position and steam valve V wide open. On rising zone temperature, the steam is first gradually turned off, and if internal heat sources cause the temperature to build up, D gradually opens to increase the amount of cool air delivered. Control T_2 is set for the minimum temperature at which air can be introduced into its zone.

If heating as well as cooling is supplied only by the fan system, and zone control is by volume dampers, special instruments known as *summer-winter* thermostats are required to open the dampers on falling temperature in winter and on rising temperature in summer. Such instruments are also used similarly to operate valves which supply hot water in winter and chilled water in summer.

Economizer Controls. Although the saving of fuel or power is one of the reasons for using any automatic control equipment, there are some applications where this is the sole reason. For example, central fan systems are usually designed to use all outdoor air, or as much as required, while it has suitable characteristics, for economical operation. Except for cases such as chemical laboratories where return air cannot be economically used, dampers are placed in both the return air and outdoor air ducts to regulate the amount of air used from each. These may or may not be mechanically interconnected but are arranged so that as one opens the other closes. Where a minimum amount of outdoor air is needed for ventilation requirements, the control of dampers may include a relay to prevent closing the outdoor damper beyond a certain adjustable point or this damper may be divided into two sections, only one of which operates with the return air damper. Another relay connected to the fan motor circuit operates the minimum outdoor opening, in either case, as the fan is started and stopped. Usually it also places the remainder of the dampers in recirculating position when the fan stops and leaves them under control of their thermostats while the fan is running.

The control of recirculation is from thermostats. Since the outdoor air, in excess of the minimum required for ventilation, is used for cooling, the dampers are commonly interconnected with other cooling devices, so as to gradually increase the amount of outdoor air, and no refrigeration is turned on until the possibilities of natural cooling are exhausted. So long as the wet-bulb temperature outside is lower than that inside the more outdoor air used during the cooling cycle, the lower the operating cost. However, as soon as the wet-bulb temperature of the outdoor air exceeds that inside, its use should be reduced to the minimum. This is done automatically by the conditions of the outdoor air as sensed by:

1. A wet-bulb thermostat.
2. A dry-bulb thermostat readjusted by a humidistat to produce operation approaching wet-bulb control.
3. A dry-bulb thermostat and a humidistat working together, either one of which may throw the dampers to recirculating position.
4. A dry-bulb thermostat, alone.

These items are listed in order of importance, from a theoretical standpoint, but practical considerations reverse the order. A wet-bulb ther-

mostat must be removed, or protected from damage, in sub-freezing weather. A dry-bulb thermostat is the most dependable under all conditions, and is generally sufficient for small installations. However, the considerably greater economy of wet-bulb or similar control justifies its use for the larger installations.

Limit Controls. There are certain limiting devices which are not concerned primarily with final room conditions but are necessary safety features. High and low limits for refrigeration pressures are described under Control of Refrigeration Equipment. Limit temperature controls are often used at heating coils exposed to sub-freezing air, to prevent freezing the condensate. Where there is danger of lack of steam pressure, a thermostat should be placed in the system to stop the fan or close the outdoor air damper when heat is not available.

The tempered air thermostat described under the Heating Cycle, serves as a low limit for air introduced in winter. When cold outside air is used for cooling, this same thermostat is used to restrict the amount, while inside conditions call for full cooling. A low limit fan discharge thermostat is sometimes operated in conjunction with the refrigerating cycle, although this is usually unnecessary.

A thermostat can also be placed in the fan discharge to stop the fan in case of fire. The maximum temperature setting is often determined by local regulations, but the most protection comes from the lowest feasible setting and a point is recommended only a few degrees higher than the highest temperature of normal operation. Safety measures to prevent gravity as well as forced flow of air, in case of fire, often require various dampers throughout the fan distribution system to be closed by fusible links or by thermostats.

Static Pressure Control

As described and illustrated in Chapter 30, the discharge of air through outlets must be carefully studied for proper results. Control systems that depend upon varying the air quantities are apt to upset the design conditions. Even where the dampers are located so as to maintain proper outlet velocities, as well as possible, by closing off portions of the grilles, there is a general increase in static pressures when most of the dampers reach their minimum positions. This tends to defeat the operation of the damper and magnify the danger of noise.

Air filters, commonly used in central fan systems, vary the operating static pressure in two ways. Reduction of air quantity tends to reduce the static drop through them as through all other resistances to air flow, but accumulation of dust increases this static drop. Thus filters add to the need for static pressure control.

This control consists generally of a device operating one or more dampers. If filters are not used and the only function of the controller is to reduce high pressures caused by reduction in amounts of air delivered, the dampers may be in the side of the main duct downstream from the fan, and arranged for opening enough to relieve the excess pressure. In this case, if the controller is of the differential type, affected by ambient pressures, care must be used to prevent distortion due to slight building up of pressure in the room outside the duct.

Whether or not filters are used, dampers may be installed across the area of the duct on either side of the fan. One type is of special design for attachment to the fan intake. Closing such dampers reduces the pressure in the distribution ducts. When filters are used, the systems may be designed for operation with the dampers partially closed while the filters are clean, so the pressure controller can automatically open them to correct for the gradually increasing resistance caused by dust accumulation.

Air distribution systems designed for high velocities and consequent high pressure drops are not entirely corrected for action of volume dampers by static pressure control at the fan, because varying pressure drops through the ducts follow changes in quantities of air delivered. Therefore, where relatively constant pressures are important it may be necessary to use controllers at several carefully selected points.

Back pressure dampers, commonly used to prevent down drafts through vent flues, may be employed to relieve objectionable pressures in rooms or other spaces, under certain conditions.

UNIT SYSTEMS

A unit system provides for the same functions as a central fan system except that the actual conditioning is usually done within the space being conditioned instead of at some central location outside of the space. The automatic control problems, therefore, become exactly the same as for central fan conditioning systems except that compactness, ease of installation and control cost often assume somewhat more importance.

Because of the usual segregated location of unit equipment throughout a building and its consequent lack of competent supervision, complete automatic control is essential to its satisfactory operation.

Unit Heaters

In its simplest form, unit heater control consists of a room thermostat to start the unit heater motor when heat is required and shut it off when the demand is satisfied. With this limited control, it is possible in some instances that, with no steam available at the heater, the operation of the fan would cause objectionable drafts. To avoid this, limit controls are available which will prevent the operation of the fan at the command of the room thermostat except when steam is available, as determined by the temperature of the steam or return pipe or the pressure of the steam supply.

Where several unit heaters serve a limited area, they may be grouped for purposes of automatic control, and several heaters placed in operation at the command of one thermostat. By properly grouping the units which will operate together, the benefit of zone control can often be obtained with a minimum of control equipment. Where such group operation is utilized, the thermostat and limit control usually function through a relay, as the combined load of the several motors may exceed the current capacity of the thermostatic control device.

In some cases where cold drafts will not result, it is desirable to operate the unit heaters continuously for circulation of air. In such instances the

room thermostat regulates the supply of steam to the unit through a control valve in the steam supply line and the unit heater motor operation is manually controlled.

Unit heaters equipped with dampers arranged for by-passing air around the heating coils are controlled by room thermostats operating modulating damper motors attached to these dampers so that as the temperatures rise, a decreasing amount of air is heated. When the by-pass is wide open the heating effect is so much reduced that control of the steam supplied to the coil is not generally important. If valve control is added, the throttling of the steam may be concurrent with, or subsequent to, the opening of the by-pass.

Cooling Units

The recommended form of temperature control for a cooling unit contemplates the continuous operation of the fan, with automatic regulation of the compressor or cooling coil, or both, as determined by a thermostat in the room, or in the return air to the cooling unit. Such operation insures continuous circulation of air in the room, and in addition to providing the cooling effect of moving air, overcomes the tendency of the air to stratify. As the temperature begins to rise, the controller opens the valve to a cold water cooling coil, or for direct expansion coils, opens a valve in the refrigerant line, closes a by-pass around the coil or starts a compressor.

Cooling units may also be controlled by arranging the room thermostats to start and stop the fan motors or by a combination of motor and refrigerant control.

Unit Ventilators

There are various types of unit ventilators available but in general all types are designed to draw air from the outside or to mix outside and recirculated air, heat it and introduce it into the room under control of a thermostat.

The design of unit ventilators has to an extent been based on the requirements for automatic temperature control and the cycles of control have been developed to include other heating devices in the rooms with unit ventilators. Unit ventilators are frequently used in schools and other types of buildings where many states have laws or regulations governing the minimum amount of ventilation to be provided. The control of the amount of outdoor and recirculated air is designed to conform to the various laws. Usually the device circulates a constant amount of air and the amount automatically taken in from outdoors is controlled in one of these ways:

1. Full recirculation until the room temperature reaches a certain point, generally two degrees, below the desired room temperature; then a minimum amount of outdoor air for ventilation while the temperature is maintained by throttling steam; and if the room temperature rises with all steam shut off, the gradual increase in amount of outdoor air up to 100 per cent.

2. Full recirculation until the room reaches a set point below room temperature, after which all air is taken from outside.

3. Gravity recirculation while the fan motor is not running, with full outside air as soon as the fan starts, obtained by a relay in the motor circuit.

4. Full recirculation or all outdoor air as determined by a manual switch which can be operated at any time whether or not the fan is running. All the unit ventilators in a single building may be operated by one or many switches.

With arrangements 1 or 2, it is desirable to include a relay to prevent the intake dampers from opening while the fan is not running, regardless of room temperatures. With a dual system of control this is essential to prevent the thermostat keeping the outside damper open until the temperature falls to the reduced setting.

The intake and recirculated air quantities are determined by a single damper or by a pair of dampers working together, and operated by a damper motor. Although this affects the temperature of the air delivered, the main heat control comes from the throttling of the steam supplied to the heating coil, with or without by-pass damper control. To prevent air being delivered at too low a temperature, a low limit thermostat is commonly installed in the air stream and set at some point between 55 and 70 F. The lower settings may cause discomfort, the higher ones overheating, depending on circumstances. The air stream thermostat can be used to turn on steam, reduce the amount of outside air, or both.

Rooms with unit ventilators frequently have auxiliary heating devices, such as direct radiators, convectors or unit heaters, all under control of a single room thermostat. A common control cycle for such rooms is composed of the following functions, assuming that 72 F is the desired temperature:

1. Below 70 F the unit ventilator intake damper is in full recirculating position and all heat is turned on.

2. At 70 F the intake damper moves to a position that will admit a predetermined minimum amount of air from outdoors.

3. At 71 F the auxiliary heating devices are shut off.

4. From 71 to 72.5 F, the heating effect of the unit ventilator is throttled.

5. From 72.5 to 74 F, the intake damper is gradually moved to increase the amount of outside air from the set minimum to 100 per cent.

6. If the room thermostat calls for too much cooling, the air stream thermostat holds the delivery temperature at a proper minimum.

Other similar cycles may be used. One additional feature is the use of an air stream thermostat that has its control point reset by the room thermostat. Then as the room temperature rises, the delivery temperature is gradually reduced from a maximum to a minimum.

All Year Conditioning Units

It is desirable to provide for automatic change-over between the heating and cooling cycles in the control system for all year conditioning units because of the probable necessity of a change several times a year. In the fall season a period requiring cooling often follows one requiring heating, and the reverse is true in the spring. The automatic change-over is especially valuable where a large number of units is used.

A control system for an all year conditioning unit providing for the automatic change-over is shown in Fig. 7. Operation of the control equipment is as follows:

1. *During the Heating Cycle.* Combination controller T_1 measures the temperature in the space being conditioned and opens control valve V_2

so as to admit steam to the heating coil whenever heat is required so as to maintain a fixed temperature in the space. Combination controller T_1 also measures the relative humidity in the conditioned space and opens control valve V_1 so as to admit water to the sprays whenever moisture is required in the space.

2. *During the Cooling Cycle.* Combination controller T_2 measures the temperature and humidity in the conditioned space and opens refrigerant control valve V_3, thereby admitting refrigerant to the cooling coil whenever cooling is required to maintain the temperature or relative humidity within predetermined maximum limits.

The temperature control point of controller T_1 must be set at a lower point than that of controller T_2 in order to provide for the automatic change-over between the cooling and heating cycles. As an example, controller T_1 might be set at 72 F and 35 per cent and T_2 at 76 F and 60 per cent. As the room conditions rise above the settings of T_1, the

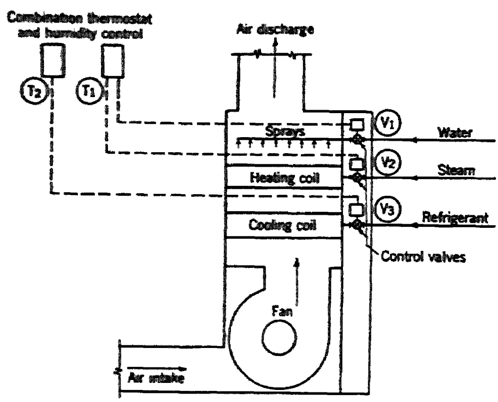


FIG. 7. ALL YEAR AIR CONDITIONING UNIT WITH COMPLETE AUTOMATIC CONTROL

heat and humidification are shut off and when they rise above the settings of T_2 the cooling and dehumidification are turned on.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic controls be used with oil burners, gas burners, and stokers in order to maintain even temperatures and provide safe and economical operation of the heating plant. There are many types of burners and many types of automatic control, and it is essential that the proper type of control equipment be selected to fulfill the requirements of the burner equipment and its application.

Combustion regulation equipment should be used on the larger commercial and industrial applications to control the secondary air supply and thereby provide for economical operation. This type of control will usually consist of a pressure regulator which measures and controls the pressure over the fire and which thereby indirectly regulates the carbon dioxide percentage in the flue gas.

On all automatically-fired steam boilers it is advisable to provide

control equipment which will stop the supply of fuel in case the boiler water line falls below a predetermined level of safety.

For hot water and warm air systems, control devices can be arranged to vary the water and air temperatures from outdoor thermostats. As the weather moderates, lower temperatures are maintained. Inside thermostats are usually installed to correct any improper results from the outside controls.

Thermostats used to control automatic fuel appliances may be supplied with clock mechanisms which will automatically shut off the heat or maintain lower temperatures during night hours for economy of fuel. For buildings that are not used every day of the week, clocks may be supplied to provide *night* conditions from Saturday noon or night to Monday morning.

Oil Burner Controls

In the normal oil burner installation as encountered in residential and small commercial installations, the burner operation is frequently regulated by electric controls and primarily governed by a room thermostat. It is essential that a limiting control be incorporated in the control system to prevent the temperature of the heating medium from exceeding any predetermined safe maximum. The type of limit control selected will depend on the type of the heating system. In a warm air furnace installation, a limit control would be used, reacting to the temperature of the heated air in the bonnet of the furnace; in a hot water system a control reacting to the temperature of the water in the boiler; and in a steam system a control reacting to the pressure of the steam in the boiler.

In addition to the normal control of the burner from the room thermostat and limit control, it is necessary that a combustion safety device be used to prevent operation of the burner under hazardous conditions. The oil fire is automatically ignited by means of gas, electric spark or incandescent element and the combustion safety control acting through a sequence device permits the burner operation only when the fire is properly established as the burner starts up. A further function of the combustion safety control is to react to any major disturbance in the flame during the running operation, shutting down the burner and preventing the discharge of unburned fuel if for any reason the flame is extinguished.

Gas Burner Controls

In the case of the domestic burner, full automatic operation is the normal requirement and the burner is started and stopped at the command of a room thermostat which, in turn, opens and closes a control valve in the gas supply line. Modulating controls and controls providing a high and low fire are also available for gas burners. For purposes of preventing abnormally high temperatures in the bonnet of gas-fired furnaces or in the temperature of the water in gas-fired hot water heating boilers or excessive pressures in gas-fired steam boilers, temperature and pressure limit controls are used. Ignition is normally secured through the use of a gas pilot flame and a safety device is provided, utilizing the heat of the pilot flame in such a manner that if the pilot light is extinguished for any reason, the main gas valve cannot be opened. For satisfactory

and economical operation, all automatically-fired gas burners should be equipped with pressure regulators on the gas supply line.

Stoker Controls

Domestic stokers are normally placed under command of a room thermostat for primary operation subject also to the command of a limit control to prevent their operation when conditions in the boiler or furnace exceed predetermined safe maximums. Utilizing coal as fuel, automatic ignition is not provided and the stokers, once ignited, maintain their fire, merely changing the rate of combustion by changing the draft and the rate at which the coal is fed. Thus, at the command of the room thermostat the stoker motor is started, driving a forced draft fan and fuel feeding mechanism. The rate of combustion is thus increased and this operation continues until the thermostat has been satisfied when the motor is stopped and the fuel in the combustion chamber continues to burn at a slow rate with reduced draft.

At certain seasons of the year, the operation of the stoker under the requirements of the thermostat may be so infrequent that there is a possibility of the fuel in the combustion chamber burning out or the fire going out between operations. To prevent this occurrence, automatic controls may be utilized to operate the stoker independently of thermostat requirements, sufficiently to sustain the fire either through a timing device functioning for short periods at predetermined intervals or through a temperature control device reacting to minimum stack or boiler temperatures. Control may also be utilized to prevent stoker operation and the delivery of coal into the combustion chamber in the event that the fire has gone completely out. This control is governed normally by the stack temperature and shuts down the stoker after a predetermined minimum stack temperature is reached.

RESIDENTIAL CONTROL SYSTEMS

The control installation in a residence may vary from the simple regulation of a coal-fired heating plant to the completely automatic all year air conditioning system. Residential installations with automatic fuel burning appliances, such as oil burners, gas burners or stokers, are normally equipped with single room thermostat, limit and safety controls as outlined previously under Control of Automatic Fuel Appliances.

Coal-Fired Heating Plant

Control in the normal coal-fired domestic heating plant consists of regulating the combustion rate in accordance with requirements. This function is accomplished by a spring or electric-driven damper motor which, under the command of a room thermostat and through chain linkage, operates the draft and check dampers of a boiler or warm air furnace. Such installation should be protected against excessive temperature or pressure by means of a limit control serving to check the fire when conditions at the boiler or furnace reach a predetermined maximum.

All Year Domestic Hot Water Supply

Hot water or steam heating boilers with automatic fuel burning appliances can be used for all year heating of domestic water supply. The

fuel burning appliance in this case is controlled from the temperature of water or pressure of steam in the boiler to maintain uniform boiler conditions and domestic hot water is heated by means of an indirect heater. The heating of the residence is normally governed by means of a thermostat which operates a control valve in the flow line of a gravity hot water or a steam system.

Air Conditioning Systems

Residential air conditioning systems normally include a heating source and a motor-driven fan for circulating air. In addition, such installations may involve spray-head equipment to supply humidity. Such installations distribute suitably heated and humidified air during the heating cycle, and during the summer or cooling cycle may be used effectively as conditioners if equipped with refrigeration means.

Regulation of the humidity during the heating cycle is normally accomplished by opening and closing a solenoid water valve supplying water to the spray-heads, the solenoid valve being under control of a room type humidity control. In the average installation the fan is permitted to run only during such intervals as the thermostat is calling for heat or at the command of a limit control to prevent the overheating of the bonnet of a warm air furnace. The limit control should also prevent the operation of the fan at the command of the thermostat until the circulating air temperature has increased to a predetermined point.

For the cooling equipment provided in such installations, control during the cooling cycle will be an adaptation of the control principles described for central fan systems selected for the type of cooling equipment utilized.

The selection of automatic control equipment for residential air conditioning systems is just as important as for commercial installations. Fewer controls are generally used and systems are usually less complicated except in the case of a very large residence installation when the control system may become as complete as the commercial installation.

CONTROL OF REFRIGERATION EQUIPMENT

The most common means of providing cooling for air conditioning may be divided into four general classifications as follows:

Compressor Type Refrigeration

Refrigeration compressors may furnish refrigerant to direct expansion cooling coils through which air is being passed, or to coils in cooling tanks through which water is passed which is then pumped to air washers or cooling coils through which the air is passed.

In either case the compressor motor may be started and stopped in order to meet the demand for refrigeration or a pressure controller may be used to regulate the low side or suction pressure of the compressor. When the latter method is used, the flow of refrigerant to cooling coils may be regulated by the opening and closing of a solenoid refrigerant valve at the command of a temperature controller or thermostat.

A high pressure cutout as an individual unit or in combination with

either a temperature or pressure controller provides a safety feature against excessive pressures on the high side of the compressor.

Many compressors may be *unloaded* by instruments sensing room or duct conditions, or by refrigerant pressures, thus reducing the frequency of starting and stopping. If two or more compressors are used for a single cooling system, *step controllers* are used to start them in sequence at intervals of a few seconds to avoid the large momentary electric input that simultaneous starting would demand.

When condensers are water cooled, thermostatic control to vary the quantity of water is needed for economical operation. Mechanical air condensers may be started and stopped with temperature demands.

Chilled water may be stored in tanks at temperatures slightly lower than required for air cooling coils. The control of temperature for the water distribution system is as described for Ice Cooling.

Ice Cooling

When ice is used for the cooling or dehumidification of air, it is usually placed in bunkers and water is sprayed over it. This water, after being cooled, may be used in air washers or surface cooling coils and is usually returned to the bunker for additional cooling after being used.

Control of the water temperature leaving the cold water tank may be maintained by a temperature controller, which measures the temperature of the water in the tank and modulates a control valve in a by-pass which permits a portion of the return water to return directly to the tank instead of passing through the sprays.

Vacuum Refrigeration

A vacuum refrigerating system consists of an evaporator, compressor, condenser and auxiliaries. The refrigerant used is water, and water vapor (steam) is the power medium.

Water which has been passed through an air washer or cooling coil is sprayed directly into the evaporator or water cooler where it is cooled by its own evaporation. A condenser is attached directly to the compressor discharge and its function is to recondense the water vapor drawn from the evaporator, plus the steam which supplies the energy for compression.

The temperature of the cold water leaving the flash chamber should be measured by a temperature controller which will in turn operate a two-position or positive-control valve installed in the steam line to the jet so as to permit steam to flow only when cooling is required. If city water is used in the condenser, the amount of water should be modulated according to the demand as measured at the condenser outlet by means of a temperature controller and control valve.

Refrigeration by Well Water

When well water is available in sufficient quantities at low temperatures during the cooling season, it may be pumped directly to air washers or cooling coils. Control is usually effected through control valves on the water supply to the cooling unit actuated by temperature or humidity controllers, or both, located either at the outlet of the conditioner or in the conditioned space.

Chapter 34

INSTRUMENTS AND TEST METHODS

Temperature Measurement, Pressure Measurement, Measurement of Air Movement, Air Change Measurements, Measurement of Relative Humidity, Dust Determination, Heat Transfer Through Building Materials, Measurement of Heat Exchange for Comfort Conditions, Combustion Analysis, Smoke Density Measurements, Carbon Monoxide Measurements

IN previous chapters, data from many tests and from much research on various divisions of heating, ventilating and air conditioning have been given. References have also been cited to a number of test codes adopted by the Society for the testing and rating of various types of equipment. This chapter presents a description of many test instruments, and discusses their use.

TEMPERATURE MEASUREMENT

Changes in the intensity of heat may be determined by several methods such as measuring the change in volume of a liquid, the change in internal pressure of a confined gas, the current set-up between dissimilar metals joined in a circuit, or the change in resistance of an electrical circuit.

Thermometers

The most common method used is the change in volume of a liquid such as mercury or alcohol enclosed in glass. Mercurial thermometers may be used for measuring temperatures from -40 F to approximately 1000 F. The lower limit is set by the freezing point of mercury. Since the boiling point of mercury is only about 675 F, the space above the mercury in thermometers designed for higher temperatures must be filled with an inert gas under pressure. Alcohol thermometers may be used for temperatures from -94 F to $+248$ F.

The more accurate thermometers are individually calibrated and have divisions etched on the stem. The two most common reference points are the freezing and boiling points of water. On the Fahrenheit scale, which is most commonly used in engineering work, there are 180 divisions between these points. On the Centigrade scale which is used by chemists and physicists, there are 100 divisions in this range. The temperature in degrees Fahrenheit equals $^9/_5$ of the temperature in degrees Centigrade, plus 32.

For permanent installations, glass thermometers are often protected by metal jackets and equipped with metal scales. Due to the heat

capacity and heat conductance of the jacket, it is more difficult to obtain the true temperature at a point with these than with the exposed etched stem type. The latter is usually preferred for test purposes. Where used to measure temperatures in a duct, it may be inserted through a cork or rubber plug. Care must be taken to locate the bulb at the point where the temperature is desired and in many cases several must be used to get a correct average.

Most mercury thermometers are calibrated for complete stem immersion. When incompletely immersed, a stem correction should be made for the most accurate determination. At ordinary atmospheric temperatures the correction is negligibly small, but it usually is important when measuring high temperatures such as those of steam and flue gas. The emergent stem correction may be calculated by the following equation:

$$K = 0.00009\,D\,(t_1 - t_2) \tag{1}$$

where

K = correction to be added, degrees Fahrenheit.
D = number of degrees on the thermometer scale which are not immersed.
t_1 = temperature indicated on the thermometer, degrees Fahrenheit.
t_2 = temperature of the non-immersed mercury column, degrees Fahrenheit.
0.00009 = difference in the coefficient of expansion of the mercury and glass.

In some cases, thermometers are calibrated for a certain depth of immersion indicated by an etched mark on the stem. Should such a thermometer be used for full immersion, a negative stem correction would be in order. In selecting a set of thermometers for a test, it is well to compare the group by immersion in a common bath and note variations. The more accurate ones can thus be selected for the more important positions. The interchanging of thermometers at inlet and outlet tends to cancel variations and therefore may result in greater accuracy. In extreme cases of small temperature differences involving large quantities of heat, it may be advisable to use thermometers graduated in tenths of degrees and mount magnifying glasses on them for accurate reading.

Since the bulb has considerable area, radiant energy may affect temperature readings[1]. In measuring room temperatures, care must be taken to locate thermometers away from hot surfaces such as radiators or cold surfaces such as walls or windows. Where this is impractical, shields should be used to screen the bulb from the radiant energy.

Thermocouple

When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends upon the metals used and the temperature difference of the two junctions. Often the cold junction is kept at 32 F by immersion in an ice bath. In other instances, a higher temperature such as that of the atmosphere is used for this junction. By proper selection of metals, any temperature up to 2900 F may be meas-

[1]Errors in the Measurement of the Temperature of Flue Gases, by P. Nicholls and W. E. Rice (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 473).

ured. Readings are obtained by means of a potentiometer or sensitive galvanometer which may be calibrated directly in degrees. A potentiometer balances the electromotive force against a known electromotive force with no current flowing, hence this method is independent of length and variations in resistance of leads. Calibration of thermocouples for high temperatures may be made against known melting points of metals. Radiation effects may be minimized by using the smallest size of wires consistent with mechanical strength. The use of small wires also makes the thermocouple sensitive to minute fluctuations in temperature.

Other advantages of thermocouples are: they are readable at remote points, they may easily and accurately be made recording, and an average temperature may be obtained readily by connecting many couples together in series.

Resistance thermometers depend for their operation upon the change of resistance of wire with change in temperature. Their use largely parallels that of thermocouples. Various metals may be used and the range is about the same as for thermocouples.

For measuring high temperatures, such as in furnaces, *pyrometers* are often used. Radiation pyrometers concentrate the radiant energy on a thermopile, and the reading is obtained on a galvanometer or potentiometer. Optical pyrometers match a narrow spectral band, usually red, emitted by the object with that from a standard electric lamp supplied with electric current.

PRESSURE MEASUREMENT

Barometer

The most accurate barometer for determining the atmospheric pressure is the mercurial type, consisting of a tube over 30 in. long closed at the top and standing in a mercury well. The barometric pressure is expressed as the height of the mercury column above the level of the mercury in the well. Such barometers are equipped with an adjustment to compensate for change in level of mercury in the well. The reading at the tube meniscus is obtained on a vernier scale. When extreme accuracy is required, as in determining the thermodynamic properties of vapors at very low absolute pressures, corrections for the variation of density of the mercury column with temperature should be made. Standard density of mercury is taken at 32 F and the conversion factor from inches of mercury to pounds per square inch is 0.491.

The following equation may be used to make corrections for temperature variations from 32 F for mercury columns:

$$h = h_1 [1 - 0.000101 (t_1 - 32)] \tag{2}$$

where

h = corrected column at 32 F, inches mercury.
h_1 = measured height of the column, inches mercury.
t_1 = observed temperature of the column, degrees Fahrenheit.

Standard atmospheric pressure at sea level is 29.921 in. mercury. Since normal atmospheric pressure decreases about 0.01 in. mercury for each 10 ft increase in elevation, it is important to make a correction if the

elevation of the barometer is not that of the test apparatus. In many cases the barometric reading may be obtained from a nearby weather bureau station. Inquiry should be made as to whether the value is as observed or corrected to sea level.

Atmospheric pressure may also be measured by an aneroid barometer which is easily portable. In this type, variations in atmospheric pressure bend the thin surface of a box or tube which contains a reduced pressure. The aneroid type is not as accurate as the mercurial and needs frequent calibration against one of the latter type.

Most of the pressure gages used in engineering work indicate the difference between the pressure being measured and the atmospheric pressure. Pressures as measured are called gage pressures. Absolute pressure may be obtained by adding barometer pressure and gage pressure algebraically.

Pressure Gages

The Bourdon type gage is a widely used device for measuring pressures. The Bourdon tube is elliptical in cross-section and circular in form, and is connected by suitable linkage to a hand which moves over a dial. An increase in pressure tends to straighten the tube and a decrease has the opposite effect. When used with high temperature steam, the tube must be protected by a water seal. When used with ammonia it must be made of steel or other material not attacked by this substance. When used for sub-atmospheric pressure, the gage is known as a vacuum gage, and is usually graduated in inches of mercury. For pressures above atmospheric, it is termed a pressure gage and is graduated in pounds per square inch. Some are made to read in both directions and are termed compound gages. Calibration is usually made with a dead weight tester, consisting of a platform and weights resting on a piston floating on oil. From the area of the piston and the total weight resting on the oil, the pressure at all points in the fluid is determined. Adjustments are provided in the gage linkage to make necessary corrections. A correction chart may also be made and used for accurate work.

For comparatively low gage pressures above and below atmospheric, the vertical *U* tube is a simple and accurate gage and is often used for test work with various fluids such as mercury, water, kerosene, or alcohol. Readings may be in inches of any of these fluids.

For measuring pressures within a few inches of water of atmospheric pressure, *U* gages are often made sloping for greater magnification of scale. In commercial gages of this type, commonly termed *draft gages*, only one tube of small bore is used and the other leg is replaced by a reservoir. Although the scale is calibrated to read in inches of water, a fluid having the density and characteristics of kerosene is often used. It is important, of course, to use a fluid having the same gravity as that for which the gage was originally calibrated, or to use a correction chart with some other fluid. Such gages may be checked one against another to detect errors in gravity of fluid. For more accurate calibration the gage may be checked against a calibrating device working on the *U* gage principle which uses hook gages and a micrometer screw. It is not considered desirable to use a slope of less than 1 to 10 in the design of these

gages. The accuracy of a draft gage is very dependent on the slope which is usually fixed by a built-in spirit level. If one side of a *U* gage is open to the atmosphere, the gage indicates pressure above or below atmospheric pressure. If both sides are connected, it indicates the difference in pressure existing between the two points of connection.

For measuring extremely low pressures accurately, very sensitive *micromanometers* of several types are available, such as the Chatelier, the Illinois or Wahlen and the Emswiler[2,3]. Calibration of these by a hook gage is impossible, and recourse must be made to fundamental calculations involving gravity of fluids and the principles involved. When proved accurate, a micromanometer is very useful for calibrating draft or slant gages.

MEASUREMENT OF AIR MOVEMENT

The problem of measuring air movement may be divided into three main parts: when confined in ducts, when circulating in free spaces, and when entering or leaving such space through openings such as grilles. Other gases might be measured by the same methods, but emphasis here will be on air measurements.

For determining the velocity, and therefore the volume of air flowing in a duct, such as in the test of a fan or a complete ventilating system, the Pitot tube as described in the A.S.H.V.E. Code[4] is probably most often used. The tube is a double tube $\frac{5}{16}$ in. outside diameter with a rounded end up-stream. The inner tube is $\frac{1}{8}$ in. inside diameter at the up-stream end, and the pressure in it is the sum of the velocity pressure and static pressure at its location in the duct. The outer tube, otherwise sealed, has 8 holes 0.04 in. in diameter and equally spaced around the circumference, and located eight diameters down-stream. A connection to this tube gives the static pressure. If both tubes are connected to opposite ends of a *U* gage, the gage indicates velocity pressure. At low velocities the resulting pressure head is so low that it becomes difficult to get accurate gage readings. The velocities used in many ducts are below the lower limit of determination with gages available. The relation between velocity and velocity pressure may be used to determine the range of gage required.

$$V = 1096.2 \sqrt{\frac{h_V}{d}} \tag{3}$$

where

V = velocity, feet per minute.
h_V = velocity pressure, inches of water.
d = density of air, pounds per cubic foot.

Air flow in a round duct is seldom uniform. In general, the velocity is lowest near the edges, and maximum at or near the center. In order

[2]Illinois Micromanometer (*University of Illinois, Engineering Experiment Station Bulletin* No. 120, p. 91).

[3]The Weathertightness of Rolled Steel Windows, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 527).

[4]Standard Test Code for Centrifugal and Axial Fans, Edition of 1938. See also Standard Code for the Testing of Centrifugal and Disc Fans (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407; Vol. 37, 1931, p. 363).

to obtain higher velocities and more uniform flow across the measuring section, it is sometimes possible to reduce the duct to a smaller cross section at the Pitot station by use of a long transition piece. In any case, a large number of readings in two traverses should be taken, with 20 being quite desirable. These should be taken at the centers of equal areas for correct determination of volumes. For round pipe, these would be located from the center by multiplying the radius by the following factors: 0.316, 0.548, 0.707, 0.837 and 0.961. A fundamentally correct method of measurement is obtained with a Pitot tube and therefore it can be used without calibration. Pulsating or disturbed flow will give erroneous results and every effort should be made to remove disturbances in the Pitot tube section.

Many forms of Pitot tubes other than the one described have been used and calibrated[5]. A double-ended tube, one end pointing down-stream, and one up-stream, is sometimes used for low velocities, but it should be carefully calibrated for accurate results. A special form of this tube design consists of two straight 1/8 in. tubes soldered together, closed at the end, and with a 0.04 in. hole in each tube opposite the line of contact. This tube is useful in exploring velocities on exhaust inlets, such as on hoods placed around grinding wheels.

The rounded approach orifice or nozzle of the general type described in the A.S.H.V.E. Unit Heater[6] and Unit Ventilator[7] Codes is a very accurate air measuring device. When it is well made, the coefficient closely approaches unity. The velocity at the mouth is increased over that in the duct, and the resulting increased velocity pressures may be measured more accurately. The discharge from such a nozzle is very uniform and provides a good location for calibration of air velocity instruments[8].

The Venturi meter is somewhat like the nozzle except for the addition of a down-stream transition section that reduces the friction drop through the measuring apparatus. However, since a good one is expensive to make, the Venturi meter is seldom used with gases, although it is often used to measure liquids.

The thin-plate square-edged orifice has a decided advantage over the rounded approach orifice in cost. Its coefficient is approximately 0.60. The exact value depends on the location of the connections, the pressure drop, the diameter ratio of orifice to pipe, and the sharpness of the edge[9].

Another method of air measurement uses the thermal electric principle where by means of a measured amount of current, heat is put into the air stream. The temperature rise is measured, and with the specific heat of the air mixture known, the weight of air flowing may be calculated. Heat should be applied uniformly to the mass of air passing, and the small temperature difference must be determined accurately.

---

[5]Technical Notes No. 546, (*National Advisory Committee for Aeronautics*, November, 1935).

[6]Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165).

[7]Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25).

[8]Discharge Coefficients of Square Edged Orifices for Measuring the Flow of Air, by H. S. Bean, E. Buckingham and P. S. Murphy (*Bureau of Standards Journal Research*, Vol. 2, 1929, p. 561).

[9]The Flow of Air Through Circular Orifices in Thin Plates, by J. A. Polson and J. G. Lowther (*University of Illinois, Engineering Experiment Station Bulletin* No. 240).

Air Currents in Free Spaces

One of the instruments useful in determining the velocity of air currents in free spaces is the Kata-thermometer. It is essentially an alcohol thermometer with a large bulb. The stem has two marks, one corresponding to 95 F, and the other 100 F. The instrument is heated above 100 F, and the time in seconds required for it to cool from 100 to 93 when placed in the air current gives a measure of the non-directional velocity. The usual way of heating the bulb is in a water bath, and it is important to wipe the Kata-thermometer dry before taking the reading. A thermostatically controlled water bath is very convenient to use along with two instruments so one may be heating while the other is in use. For high atmospheric temperatures the high temperature Kata with a range of 130 to 135 F may be used. Usually several readings are taken in a given location and the average used. Each Kata has its own factor etched on the stem, and this factor must be used with the cooling formula or chart for obtaining the velocity. The Kata-thermometer is useful in exploring ventilated spaces to determine whether the proper air movement and distribution is being maintained. The Kata-thermometer also finds use in determining the cooling power of the atmosphere, since it loses heat by radiation and convection when dry, and by radiation, convection, and evaporation when the bulb is equipped with a wetted cloth covering[10].

Another instrument for measuring low velocity air currents is the heated thermometer anemometer[11]. This consists of an ordinary mercurial glass thermometer with a resistance winding on the bulb. Current is supplied from an external source in a measured amount. The temperature rise shown on this heated thermometer over that shown by an ordinary thermometer at the same location, and the current supplied, make it possible to calculate the non-directional velocity of the air stream. Since a smaller bulb is used than that on the Kata-thermometer, it is less affected by radiant heat sources.

Another instrument is the hot wire anemometer which is available in several patterns. In general, a measured current is supplied to raise the temperature of a fine bare wire above the temperature of the surrounding air. With the use of a very fine wire, minute fluctuations in velocity may be measured, and the area exposed to radiant exchange with heated or cooled surfaces is at a minimum. This instrument is easily made remote reading or recording. A group of them may be connected together to give the average velocity in a space, or the velocity at individual points within a test space, by suitable switching arrangements[12,13].

Deflecting Vane Anemometer

The deflecting vane anemometer consists of a pivoted vane enclosed in a case, against which air exerts a pressure as it passes through the instrument from an up-stream to a down-stream opening. The move-

---

[10]Temperature, Humidity and Air Motion Effects in Ventilation, by O. W. Armspach and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 103).

[11]The Heated Thermometer Anemometer, by C. P. Yaglou (*Journal Industrial Hygiene and Toxicology*, Vol. 20, October 1938, No. 8).

[12]Development of Testing Apparatus for Thermostats, by D. D. Wile (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 349).

[13]Linear Hot Wire Anemometer, Its Application to Technical Physics, by L. V. King (*Journal Franklin Institute*, 1916).

ment of the vane is resisted by a hair spring and a damping magnet. The instrument gives instantaneous readings of directional velocities on an indicating scale. When used in fluctuating velocities, it is necessary to average visually the swings of the needle to obtain average velocities. This instrument is very useful in locating and measuring peak velocities that may be objectionable in air conditioned spaces. Various attachments are available, such as a double tube arrangement for obtaining velocities in ducts, and a device to measure static pressures. Another attachment will be mentioned later under the measuring of velocities at inlets and outlets. Each instrument and the attachments for it must receive individual calibration.

Propeller or Revolving Vane Anemometer

The propeller or revolving vane anemometer consists of a light revolving wheel connected through a gear train to a set of recording dials that read the linear feet of air passing in a measured length of time. It is made in various sizes, 3 in., 4 in., and 6 in. being most common. Each instrument requires individual calibration. At low velocities the friction drag of the mechanism is considerable. In order to compensate for this, a gear train that overspeeds is commonly used. For this reason the correction is often additive at the lower range and subtractive at the upper range with the least correction in the middle range of velocities. Most of these are not sensitive enough for use below 200 fpm; therefore, they are not commercially available for the low velocity range met with in air conditioned spaces. Further discussion follows under air measurement at inlets and outlets.

Measurement of Velocities at Inlets and Outlets of Ducts

In the field it is often advisable to make volume measurements at the face of the supply openings. Often it is hard to get into the duct system, or it is difficult to find sections where the flow would be sufficiently uniform. The many types of approaches and grilles used make a high degree of accuracy almost impossible. For the best accuracy the instrument and its application should be checked on a similar approach and grille in the laboratory before use in the field. Where extreme accuracy is not required, such as in balancing a system, various instruments may be used at the face of the grille.

Tests have shown that the propeller type anemometer can be used successfully on the heavier type of supply grilles, such as square mesh of the cast, or pressed pattern[14]. The core area is divided into equal squares, and the anemometer is held against the face of the grille for the same length of time in each. To get the air volume in cubic feet per minute, the average corrected velocity in feet per minute thus obtained is multiplied by the average of the gross and net free area of the grille in square feet and by a correction coefficient determined as 0.97 at velocities from 150 to 600 fpm and as 1.00 at higher velocities.

On exhaust grilles, the anemometer traverse is made as described previously. The air volume may be determined by multiplying the corrected velocity in feet per minute by the gross or core area of the

[14]A.S.H.V.E. RESEARCH REPORT No. 857, 911 and 966—Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 201, Vol. 37, 1931, p. 619, and Vol. 39, 1933, p. 373).

grille in square feet and by a coefficient for average conditions of 0.80. This coefficient takes care of the interference of the bars of the grille and the effect on the anemometer of the air entering an exhaust grille through 180 deg[15].

When a propeller type anemometer is held in a stream of varying velocities, it tends to indicate higher than the true average, that is, the speed of the propeller is nearer to the top velocity in its area than it is to the minimum velocity. This is the main reason for the large difference in ratings of unit ventilators by the anemometer method and by air volume measurements in a duct approach to the inlet[16].

Any of the other anemometers described can be used within their range at the face of supply grilles when properly applied. In principle it is a case of finding the velocity at many points and using the average thus found with the correct discharge area at that cross-section. The deflecting vane anemometer equipped with a jet on the end of a rubber tube has been found especially convenient and accurate on supply grilles[17]. On modern air conditioning grilles the core area is used without a correction coefficient when the jet is held one inch away from the face of the grille. At this distance the constriction due to the thin bars has disappeared since the small air jets have reunited, and the air stream has not yet spread beyond the core dimensions. With deflecting grilles the exploring jet should be turned to the angle giving a maximum reading. This method of using this instrument is only applicable to supply grilles and cannot be used on exhaust grilles because of static pressure differences at the location of the jet and the instrument case.

While hardly a quantitative instrument, smoke is very useful in studying air streams and currents. The application of a more accurate instrument is often made more exact by a preliminary exploration with smoke. A mixture of potassium chlorate and powdered sugar in equal portions gives a very satisfactory non-irritating smoke. It is fired by a match, and since considerable heat is evolved, it should be placed in a pan away from inflammable objects.

AIR CHANGE MEASUREMENTS

Atmospheric air contains a certain amount of carbon dioxide. Its concentration is increased within enclosures by the carbon dioxide given off by occupants. The air changes through all means: open windows, infiltration, and mechanical ventilation, may be measured by the carbon dioxide concentration[18]. The Petterson-Palmquist apparatus has been accepted as the standard device for the determination of carbon dioxide in air. The principle used is absorption by caustic potash solution of the carbon dioxide in a known volume of air, and a remeasurement of the volume in a finely graduated capillary tube. Since the concentrations are in the order of 3 to 10 parts in 10,000, extreme care must be used to

[15]A.S.H.V.E. RESEARCH REPORT No. 1092—The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr., and M. H. Dean (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 387).

[16]A.S.H.V.E. RESEARCH REPORT No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 463).

[17]A.S.H.V.E. RESEARCH REPORT No. 1076—Air Distribution From Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 77).

[18]A.S.H.V.E. RESEARCH REPORT No. 959—Indices of Air Changes and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

obtain accurate determinations. Since occupants also give off moisture, the increase in humidity may also be used as an index of ventilation within a space. Humidity determinations are much simpler to make, but the accuracy may be affected slightly by absorption of moisture by hygroscopic materials such as fabrics and wood within the space. Measured amounts of either carbon dioxide or water vapor may be added for test purposes. However, neither method is used at the present time, and more direct methods of measuring air supply and air distribution are in favor.

MEASUREMENT OF RELATIVE HUMIDITY

Wet- and dry-bulb mercurial thermometers are usually used to determine relative humidity. The sling psychrometer is a common mounting of the thermometers to permit swinging. The wet-bulb wick and water for wetting it must be clean, and the temperature of the water should preferably be slightly above the wet-bulb temperature. An air stream velocity of 900 fpm is recommended, although velocities from 300 fpm to 1000 fpm have been found satisfactory for passage over the wet-bulb wick. The velocity may be obtained by whirling the thermometer or by aspirating air over the wet-bulb. In ducts, the air flow itself gives the proper evaporating conditions. Several observations should be made until the minimum temperature is reached. Relative humidity may be obtained from tables or psychrometric charts[19]. Although it is common practice to use the charts which are based on a barometric pressure of 29.92 in. mercury, a correction for barometric pressure is necessary for extreme accuracy. This correction is made by multiplying the relative humidity as determined from the chart by the ratio of the observed barometric pressure and the standard barometric pressure.

For temperatures below 32 F, the water on the wick is allowed to freeze, during which time the temperature will drop below the true wet-bulb. A thin film of ice is more desirable than a thick one, and it is satisfactory to remove the wick and freeze a thin film directly on the bulb. Care must be taken to read the temperatures accurately due to the slight wet-bulb depressions. Tables for ice conditions must be used[20].

The dew-point apparatus for humidity measurements consists of a polished plated container cooled by the evaporation of a volatile liquid within. The temperature at which the first slight water vapor forms is the dew-point. If the temperature is below 32 F, the deposit will appear as frost. Another method of determining humidity is by chemical means in which the water vapor is removed by a drying agent and weighed on a chemical balance. A thermal conductivity method is available for temperatures above 212 F or for extremely low humidities[21].

DUST DETERMINATION

The measurement of dust is complicated by the many kinds involved. Some of the collecting methods are impingement on viscous surfaces,

[19]Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew-Point, (*U. S. Department of Agriculture*, Weather Bureau, Washington, D. C.).

[20]A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems, by W. H. Carrier and C. O. Mackey (*A.S.M.E. Transactions*, January, 1937, p. 33; Discussion, *A.S.M.E. Transactions*, August, 1937, p. 528).

[21]Gas Analysis by Measurement of Thermal Conductivity, by H. A. Daynes (*Cambridge Press*, 1933).

impingement at high velocity under water, collection on porous crucibles through which air passes, and electric precipitation. Determination may be by direct weighing of samples or by microscopic counting. The most commonly used methods are the modified Hill dust counter using microscopic count, the Smith-Greenburg impinger which collects samples in water and which are counted under a microscope in a Sedgwick cell[22], and the Lewis sampling tube with the analytical determination of the increase in weight of a porous crucible. All reports should state the method of sampling and counting. The A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work specifies the porous crucible method[23].

HEAT TRANSFER THROUGH BUILDING MATERIALS

The A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls specifies the use of the guarded hot plate for tests on homogeneous materials. It further states that the mean temperature shall be at 60 F and the materials dried.

This code describes the construction and use of the guarded hot box for determining overall heat transmission coefficients of built-up sections. The standard temperature range through the test section is specified as 80 F and the mean temperature of the wall as 40 F.

The Nicholls heat meter is very useful for determining the heat flow through walls of buildings[24].

MEASUREMENT OF HEAT EXCHANGE FOR COMFORT CONDITIONS

Several instruments have been devised to measure the effect of various factors as they relate to the comfort of the body. The principle ones are the Kata-thermometer, Dufton's eupatheoscope, Vernon's globe thermometer, Winslow and Greenburg's thermo-integrator, and Yaglou's heated globe[25,26]. These instruments are attempts to stimulate and measure the heat exchanges between the human body and its environment. In order to stimulate conditions of hard physical labor, the entire surface of the device is covered with a wet cloth. At present special attention is being given the thermo-integrator as a means of measuring radiant effects of environmental conditions.

COMBUSTION ANALYSIS

The analysis of flue gases to determine completeness and efficiency of combustion is usually made chemically with the Orsat apparatus. This consists of a measuring burette, a leveling bottle, and three pipettes. Carbon dioxide is absorbed in the first pipette by potassium hydroxide,

---

[22]Public Health Bulletin, No. 144, 1925, (*U. S. Public Health Service*).

[23]Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by S. R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

[24]A.S.H.V.E. RESEARCH REPORT No. 685—Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 65).

[25]Instruments and Methods for Recording Thermal Factors Affecting Human Comfort, by C. P. Yaglou, A. P. Kratz and C.-E. A. Winslow (Year Book, *American Journal Public Health*, 36-37).

[26]The Thermo Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 149).

oxygen in the second by potassium pyrogallate, and carbon monoxide in the third by cuprous chloride. A known volume of gas is drawn in, and after each of the three absorptions the reduced volume is again measured in the burette. Pressure and temperature of the gas sample are kept constant while measuring. Several passes are made through each pipette which contains tubes or glass beads to increase the wetted surface. It is essential that each reaction be completed before the next reaction is started. Since the life of the reagents is limited, it is well to keep a record of the number of samples tested. Care is needed in operation to prevent the pulling of reagents out of the pipettes into the capillary tubing and burette. Many recording gas analyzers are available and are usually found in the larger plants.

SMOKE DENSITY MEASUREMENTS

A common method of determining the relative density of smoke issuing from chimneys is by visual comparison with the Ringelmann Smoke Charts. A sheet of four ruled charts with varying weights of black lines is used. The sheet is 12 by 26 in. over all on which are four charts, each consisting of 294 squares, 14 wide and 21 high. The width of line and spacings is given in Table 1.

TABLE 1. RINGELMANN SMOKE CHART SPACINGS

Number of Card	Thickness of Lines, mm	Distance in Clear Between Lines, mm
1	1.0	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5

The charts are placed 50 ft from the observer and in line with the stack to be observed. At this distance the lines disappear and the charts appear as varying shades of gray. At times a white chart is added as No. 0 to the left of the four charts 1 to 4, and a black chart to the right as No. 5.

Apparatus using the photo-electric cell has been devised for recording smoke densities in large plants.

CARBON MONOXIDE MEASUREMENT

In garages and vehicular tunnels carbon monoxide is a constant potential danger. In small amounts it causes headaches and inefficiency, and in larger concentrations it causes collapse and death in rather short periods of exposure. A method of analyzing for carbon monoxide concentrations completes the oxidation of the carbon monoxide in a known volume of sample, in the presence of a catalyst. The heat resulting is measured by a thermocouple calibrated in parts per 10,000 of carbon monoxide[27].

[27]A Carbon Monoxide Recorder, by S. H. Katz, D. A. Reynolds, H. W. Frevert and J. J. Bloomfield (*U. S. Bureau of Mines, Technical Paper* No. 355, 1926).

Chapter 35

MOTORS AND MOTOR CONTROLS

Direct Current Motors, Alternating Current Motors for Single Phase and Polyphase, Special Applications, Classification of Motors, Manual Control, Automatic Control, Pilot Controls, Direct Current Motor Control, Squirrel Cage Motor Control, Multispeed Motor Control, Slip Ring Motor Control, Single Phase Motor Control

THE electric motor, available in many different types suitable for various services, is now the most widely used form of prime mover. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors used for heating, ventilating and air conditioning applications may be divided into two general classifications as follows:

1. For use with direct current.
2. For use with alternating current.

DIRECT CURRENT MOTORS

There are three types of direct current motors available:

1. Shunt Wound.
2. Compound Wound.
3. Series Wound.

Shunt Wound motors, being suitable for application to fans, centrifugal pumps, or similar equipment where the amount of starting torque required is relatively small, are used for the majority of applications in the field of heating, ventilating and air conditioning. They may be used on reciprocating pumps and compressors, if started under unloaded conditions.

Compound Wound motors are required for application to compressors, stokers, reciprocating pumps when started under loaded conditions, and also when applied to similar equipment where high starting torque is required. Whenever frequent starting makes high starting and accelerating torque desirable, or where sudden changes of load are encountered, compound wound motors are used.

Series Wound motors find only limited application in a few special cases and are available in only a limited range of sizes.

Speed Characteristics

Direct current motors are available with speed characteristics of four types:

1. Constant speed.
2. Adjustable speed.
3. Adjustable varying speed.
4. Varying speed.

Constant Speed motors may be shunt wound or compound wound. Shunt wound motors have a nearly flat speed-load characteristic, with a regulation of 15 per cent for up to ¾ hp, 12 per cent for one to 5 hp and 10 per cent for 7½ hp and larger, based on full load speed.

Compound wound motors have a speed regulation over the range from full load to no load of not more than 25 per cent, based on full load speed.

Adjustable Speed motors are usually shunt wound since it is impractical to maintain the proper relation between the shunt and series fields of compound wound motors when wide variations of the field strength are required to obtain the speed adjustment.

Adjustment of the speed of shunt wound motors is obtained by field control on motors rated at ¾ hp and larger, with the minimum or base speed at full field strength and higher speeds at reduced field strength (obtained by adding resistance in the field circuit). The speed regulation from no load to full load will not exceed 22 per cent for 2 to 5 hp; nor 15 per cent for 7½ hp and larger. Below 2 hp, the regulation may exceed 22 per cent. If closer speed regulation is required, specifically wound motors must be obtained.

Practically constant horsepower output is obtained at all speeds up to a ratio of 2 to 1. For higher speed ratios, the horsepower rating at the minimum speed is less than at the maximum speed, this difference varying with the speed ratio. High efficiency is maintained over the entire speed range. Most listed constant speed motors are suitable for operation up to a speed ratio of 2 to 1 by the use of proper control equipment.

Adjustable Varying Speed motors may be either shunt or compound wound and speed adjustment is obtained by adding resistance in series with the armature. The speed thus obtained is always below the rated full-field speed. Any standard shunt or compound wound constant speed motor may be used in conjunction with the proper armature resistor. The usual range of speed reduction is 50 per cent. The speed obtained for any setting of the resistor depends on the load of the motor and will vary with this load.

The speed regulation at high speed is comparable to a constant speed motor, but becomes poorer as the speed is decreased.

When operating at reduced speed, an increased torque requirement which the motor could easily handle at rated speed is easily sufficient to stall the motor; for example, a motor operating at two-thirds speed would be stalled by a torque about 50 per cent in excess of the normal requirement.

The efficiency of the motor is reduced as the speed is reduced, since the

loss in the resistor is greater at lower speeds. Speed reduction by armature control is usually selected where:

1. A wide speed range is not required.
2. Close speed regulation is not necessary.
3. Operating time at reduced speed is short.
4. Operating load at reduced speed is small so that the reduced efficiency can be ignored.
5. The rating is less than 1 hp.

Varying Speed motors are series wound and the speed varies with the load on the motor. They should be used where:

1. The load is practically constant or increases with speed.
2. The motor can easily be controlled by hand.

They should not be used where there is a possibility of operation without load or at a reduced load, as the speed of the motor may become dangerously high.

For shunt wound motors with full field strength, the starting torque varies almost directly with the starting current, which is dependent on the resistance in the armature circuit. With varying positions of the starting rheostat, it is possible to obtain a wide range of starting torque, within the limits of starting current permitted by the power company.

A compound wound motor requires somewhat less current for the same starting torque. The maximum torque of shunt, series, and compound wound motors is limited by commutation.

ALTERNATING CURRENT MOTORS

Alternating current motors may be divided into two main groups, namely, (1) those operating on single phase current, and (2) those operating on polyphase current.

1. Single phase motors are available in four common types:
   a. Capacitor motors.
      1. Full capacitor.
      2. Capacitor start induction run.
   b. Repulsion induction motors.
   c. Repulsion start, induction run motors.
   d. Split phase motors.

2. Polyphase (2 or 3 phase) motors are available in four common types:
   a. Squirrel cage induction motor.
   b. Automatic start induction motor.
   c. Slip ring, wound rotor induction motor.
   d. Synchronous motor.

Where the public utility supplying the current determines that a particular installation should be served with polyphase current, it is generally understood that the major portion of the motors will be for polyphase current, although it is commonly acceptable for the smaller motors to be single phase. This will limit the use of single phase current to the smaller motor ratings and the polyphase to the larger motors. Domestic and semi-commercial installations will invariably be single phase.

Single Phase Motors

Capacitor type motors are available in ratings up to 10 or 15 hp for general purposes. These motors are recommended for pumps, compressors and fan duty including housed centrifugal fans and propeller fans. The general purpose motor is commonly known as a high torque capacitor motor having approximately 300 per cent starting torque with normal current and having a different value of capacitance for starting and running which is automatically changed over by a mechanical or electrical means.

Capacitor motors for *fan duty* are usually divided into the open high torque type for belted fans and the totally inclosed non-ventilated low torque type for propeller fans mounted directly on the motor shaft. The open low torque capacitor motor may be used with small centrifugal fans mounted on the motor shaft.

Although the motors for *belted fans* are called high torque, the available starting torque is somewhat less than the torque of the general purpose motor and the slip at full load is approximately 8 per cent. With this larger amount of slip, adjustable speed down to 60 or 70 per cent of rated speed may be obtained by line voltage variation. Motors for *propeller fan* drive may be supplied with sleeve bearings to obtain greater quietness in the smaller sizes where the fan thrust does not exceed approximately 25 lb. For larger fans, thrust ball bearing motors should be used. Low torque capacitor motors have approximately 50 per cent starting torque and do not change the value of capacitance from start to run.

Capacitor motors with *high slip* may have taps brought out from the main winding which, when connected to the line, give a second speed of from 65 to 70 per cent of the normal speed. This type of motor must be specially designed for the individual fan, otherwise the correct low speed will not be obtained. Care should be exercised in applying it to centrifugal fans where restriction to the air flow through the use of adjustable dampers changes the motor load and consequently the speed. This same effect is also found in transformer speed controllers, however, a series of transformer taps allows for a selection which partially overcomes the effect of change in motor load.

Capacitor start-induction run motors are usually confined to the smaller horsepower ratings and differ from the capacitor motors by having no running capacitor. The value of starting capacitance used may vary with the different types of applications involved. These motors may be used for practically any of the applications met in air conditioning. However, consideration should be given to the fact that they are not as quiet as a capacitor motor.

Repulsion induction motors start as repulsion motors and operate under full speed as combined repulsion and induction motors through the inherent characteristics of the motor which has, in addition to the wire winding with commutator, a buried squirrel cage winding. No additional switching devices are required to change over from start to run. This and the repulsion motor described later may be used for constant speed drives where high starting torque is required and where commutator and brush noise is not a factor.

The *repulsion start-induction run* motor starts as a repulsion motor, has a switching means for transferring from start to run which short circuits the commutator and permits operation under full speed as a wound induction motor. This motor is suitable for applications similar to those for which the repulsion induction motor is used.

The *split phase* motor has a high resistance auxiliary winding in the circuit during starting which is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions, it operates as a single phase induction motor with one winding in the circuit. These units are available for the lower horsepower ratings and when equipped with a high slip rotor may be used for adjustable varying speed through line voltage control.

Polyphase Motors

Squirrel cage induction motors are available in three types and a full range of sizes:

1. The normal torque, normal starting current squirrel cage motor has close speed regulation, high efficiency, high power factor, medium starting torque, high pull-out torque, and is suitable for general purpose applications. This motor has a large current inrush and a low starting current power factor. It operates with these characteristics only when started directly across the line on full voltage. When central stations require current limiting starting equipment on such motors, the starting torque is less. Current limiting hand operated starters are standard equipment.

2. The normal torque, low starting current squirrel cage motor has approximately the same torque as the normal current motor, but the starting current is about 20 per cent less than the normal torque motor on full voltage and ordinarily within the *Edison Electric Institute* locked rotor current limits on sizes up to 30 hp.

   This motor lends itself to automatic or remote control because no current limiting starting equipment is necessary up to and including 30 hp. A magnetic starter with low voltage and thermal relay overload protection gives the most satisfactory service.

3. The high torque, low current squirrel cage motor has a starting torque approximately 25 to 50 per cent greater than the normal torque motor on full voltage with starting current approximately 10 per cent less than the normal torque motor started on full voltage, but within the required limits on 30 hp sizes and smaller. These motors are also started directly across the line on full voltage through a magnetic starter or other approved starting device.

These three types of motors are also available in two, three, or four speed designs with variable torque, or constant torque characteristics. Two speed motors may be either single, or two winding; three speed motors are single, two, or three winding; and four speed motors are two, three, or four winding. When a motor is wound with a winding for each speed, better operating characteristics may be obtained because no sacrifice is made for the other speed and operating characteristics approaching single winding motors may be expected.

Frequently, multispeed motors lend flexibility to an installation that cannot be obtained in any other way.

Multispeed motors are started directly across the line through magnetic starting equipment with overload and low voltage protection and compelling relays to insure starting on low speed regardless of the ultimate running speed. Starting on low speed limits the starting current to the starting current of the low speed winding and consequently lowers the maximum demand.

TABLE 1. CLASSIFICATION OF MOTORS

Current	Type	Speed Characteristics	Full Voltage		Hp Range	Type of Application See Footnote*
			Starting Torque	Starting Current		
		Constant Speed Drives				
	1. Shunt	Constant	Medium	Medium	All	(*a*) Fans and (*c*) Centrifugal Pumps
Direct	2. Compound	Constant or Variable	High	Medium	All	(*b*) (*c*) (*e*) Reciprocating Pumps and frequent or hand starting
	3. Series	Variable	High	Medium	Small	(*d*) Fans direct connected
	4. Squirrel Cage General Purpose	Constant	Normal	High 6-8 Times	All	(*a*) Fans and (*c*) Centrifugal Pumps
	5. Squirrel Cage Medium Torque	Constant	Normal	Medium 5-6 Times	Medium Small	(*a*) Fans and Centrifugal Pumps
	6. Squirrel Cage High Torque	Constant	High	Medium 5-6 Times	Medium Small	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
Poly-Phase	7. Automatic Start High Torque	Constant	High	Low 3 Times	Medium	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
	8. Slip Ring Wound Rotor	Constant	High	Low 1-3 Times with secondary control	All	(*a*) and Hoists (*b*) Reciprocating Pumps (*c*) and Frequent (*e*) or Hand Start
	9. Synchronous High Speed	Constant	Medium	Medium 5-7 Times	Medium Large	(*a*) Fans and Centrifugal Pumps
	10. Synchronous Low Speed	Constant	Low	Low 3-4 Times	Medium Large	(*a*) Reciprocating Compressors starting unloaded
Single Phase	11. Capacitor	Constant	High	Normal	Medium Small	(*b*) Pumps and Compressors

*Applications:

a. Drives having medium or low starting torque and inertia (WR^2) such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded.

b. Drives having high starting torques, such as reciprocating pumps and compressors started loaded.

c. Similar to (*a*) except where frequent or hand starting (large WR^2) requires a higher starting and accelerating torque.

d. Fans direct connected.

e. Stoker drives.

TABLE 1. CLASSIFICATION OF MOTORS—(Continued)

Current	Type	Speed Characteristics	Full Voltage Starting Torque	Full Voltage Starting Current	Hp Range	Type of Application See Footnote*
Single phase	12. Capacitor Fan	Constant	High	Medium	Medium Small	(a) Fans—belted
	13. Capacitor Fan	Constant	Low	Medium	Medium Small	(d) Fans—direct
	14. Capacitor Start Induction Run	Constant	Any	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	15. Repulsion Induction	Constant	High	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	16. Repulsion Start Induction Run	Constant	High	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	17. Split Phase	Constant and Adjustable	Medium	Medium	Fractional	(a) Fans (b) Pumps and Compressors
		Adjustable Speed Drives				
Direct	18. Shunt Field Adjustment	Constant	Medium	Medium	All	(a) Fans and (e) Centrifugal Pumps
	19. Shunt Armature Resistor	Variable	Medium	Medium	All	(a) Fans and (e) Centrifugal Pumps
Polyphase	20. Squirrel Cage High Slip, Tapped Winding	Variable	Medium	Medium	Medium Small	(a) Fans
	21. Squirrel Cage High Slip, Transformer Adjustment	Variable	Medium	Medium	Medium Small	(a) Fans
	22. Squirrel Cage Separate Winding or Regrouped Poles	Constant Multi-Speed	Medium or High	Low	All	(a) Fans (b) Pumps and (c) Compressors

TABLE 1. CLASSIFICATION OF MOTORS—(Concluded)

Current	Type	Speed Characteristics	Full Voltage Starting Torque	Full Voltage Starting Current	Hp Range	Type of Application See Footnote*
POLY-PHASE	23. Wound Rotor, Slip, Ring, External Secondary Resistance	Variable	High	Low	All	(a) Fans and (b) Centrifugal Pumps
SINGLE PHASE	24. Capacitor High Torque Tapped Winding	Variable	High	Normal	Medium Low	(a) Fans, belt
	25. Capacitor Low Torque Tapped Winding	Variable	Low	Medium	Medium Low	(d) Fans, direct
	26. Capacitor High Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	27. Capacitor Low Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	28. Split Phase Regrouped Poles	Constant	Normal	Normal	Fractional	(d) Fans

Often where the central station requires current limiting starting equipment for the normal torque, normal starting current motor, it is advisable to use the normal torque low starting current multispeed motor.

High slip polyphase motors may be used for adjustable varying speed drives in a manner similar to that described for capacitor motors, with either a transformer speed regulator or tapped motor windings.

It is apparent from these motor characteristics that a squirrel cage motor may be selected for operating any air conditioning and allied equipment.

Automatic start induction motors are constructed with two windings on the rotor, one of which is a high resistance, squirrel cage winding used in starting and gives a high starting torque approximately the same as the high torque, squirrel cage. A centrifugal mechanism within the motor switches to the second low resistance winding when the motor comes up to speed, thus obtaining running characteristics equal to the normal torque, normal current squirrel cage motor. The power factor of the starting current is high.

Slip ring wound rotor motors are built for two classes of service, constant speed and adjustable variable speed. The motors are identical in each case and use the same primary control, the only difference being in the secondary control.

Slip ring motors for constant speed service are used where high starting torque with low starting current is required for bringing heavy loads up to speed. The resistance is in the secondary or rotor circuit, only when starting, and is short circuited when the motor is up to speed.

For adjustable varying speed service, part or all of the secondary controller resistance is in the circuit whenever the motor is operating below full speed. The speed obtained with a given resistance in the secondary circuit is dependent on, and changes with the load on the motor. The horsepower developed by the motor is approximately proportional to the speed, whereas the power required by the motor is practically the same at reduced speed as at full speed, hence the efficiency at reduced speeds is much lower than at full speed.

Synchronous motors are ordinarily used only where there is a need for, or advantage in, obtaining power factor correction. It is necessary to consider each application as a special case which must be individually engineered, since for satisfactory operation, the combined moment of inertia of the compressor fly wheel and motor rotor must be correctly established.

The general classification of motors used for heating, ventilation and air conditioning is shown in Table 1.

SPECIAL APPLICATIONS

A few applications of motors may require special constructions such as splash proof, explosion proof, fully enclosed, and self-ventilated to meet hazardous or special duty conditions. These requirements are frequently encountered in certain industrial applications, in which cases it is necessary to select the motors from the viewpoint of service conditions, as well as the required operating characteristics to meet the demands of the machines being driven.

CONTROL EQUIPMENT FOR MOTORS

In selecting control for alternating and direct current motors it is necessary to determine whether the installation is to be operated by manual or automatic control. The available controls and the function of each group of apparatus may be outlined as follows:

1. Manual Control:
   a. To establish current.
      (1) Snap switch.
      (2) Knife switch.
      (3) Manually operated contactor.
      (4) Drum switch.
   b. Establish current and add overload protective device.
      (1) Snap switch with overload element.
      (2) Knife switch with fuse or thermal cutout.
      (3) Manual contactor with overload protective device; also reduced voltage starting compensator.
      (4) Drum switch with overload protection.
   c. Establish current and add overload and low voltage protective devices.
      (1) Not used.
      (2) Not used.

(3) Manual contactor or reduced voltage compensator with overload and low voltage release.
(4) Drum switch equipped with latch coil to give low voltage release.

2. Automatic Control:
a. To start on full voltage.
(1) Without overload device.
(2) With overload device.
(3) With combination overload device and knife switch.
b. Reduced voltage starting.
(1) Primary resistance type starter.
(2) Auto compensator type.
(3) Reactance type.

PILOT CONTROLS

In selecting pilot control devices to operate in conjunction with either manual or automatic motor control, it is necessary that they be classified as follows:

1. *Two Wire Control.* Most thermostats, float switches, and pressure regulators provide two wire control which gives low voltage release. A three position pilot switch can be used in connection with this method and thus provide manual control. With a low voltage (12 or 20 volt) control circuit it is desirable to use a low voltage thermostat. When this type of thermostat is used it will be found that a saving in the wiring cost results. When using the low voltage thermostat on a control circuit a relay and transformer panel should be used instead of the low voltage coil on the starter.

2. *Three Wire Control.* Momentary contact start and stop push button stations are usually furnished as standard accessories with automatic starters, which give low voltage protection. This control cannot be used in combination with two wire pilot devices.

In selecting manual control for an alternating or a direct current motor, the common practice is to locate the control near the motor. When the control is installed at the motor, an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Frequently manual control is employed only as a device to give overload protection and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air conditioning system. In other cases manual control in the form of drums, when used with multispeed motors, is only used as a speed setting device with the starting and stopping functions operated automatically through thermostats, and pressure switches.

Because of the increasing complexity of air conditioning systems, heating, ventilating and air conditioning equipment is being operated on automatic control with less dependence on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a control device located elsewhere. In the majority of air conditioning installations, requiring motors 1 hp and larger, two or three phase alternating current is usually supplied.

DIRECT CURRENT MOTOR CONTROLS

Air conditioning installations using direct current power are now only used where alternating current is not available. Direct current motors are always started through starters, which are devices using a resistance to be put in series with the armature circuit during starting only, the resistance being gradually cut out as the motor comes up to speed. The starting current is held within safe limits by the use of the resistance.

The speed of a direct current motor may be regulated by the following methods:

1. Speed regulation by field control—by using a device with resistance to be put in series with the field winding. After the motor has been started to be used to increase the speed of the motor above full field speed.
2. Speed regulation by armature control—by using devices with resistance to be put in series with the armature circuit to be used to reduce the speed of the motor below full field or normal speed.
3. Combinations of field and armature control, so that the starting, field control, or armature control may be combined in a single unit.

Field control is usually preferred, depending on the size of the installation. For example, if a direct current motor were required with speed regulation between 1200 and 600 rpm, a choice of supplying a 1200 rpm motor with armature control or a 600 rpm motor with field control, both giving the same speed variation would be possible. While the 1200 rpm motor with armature control is lower in first cost than the 600 rpm motor with field control, the cost of operating the 600 rpm motor with field control is less and will save the difference in first cost over a period of time depending on the size of installation. A wide speed variation can be easily obtained in a direct current motor by using a combination of field and armature control.

SQUIRREL CAGE MOTOR CONTROL

To meet the requirements of various drives of an air conditioning system, three types of squirrel cage, two or three phase motors may be used:

1. Normal torque, normal starting current.
2. Normal torque, low starting current.
3. High torque, low starting current.

Because of the large current inrush of the normal torque, normal starting current motor, central stations usually require current limiting starting equipment on such motors above 5 hp. To meet the starting current requirements, manual or automatic current limiting starting compensators are used. These compensators are equipped with 50, 65 and 80 per cent voltage taps, the 65 per cent tap being regularly furnished when the compensator leaves the factory. Motors 5 hp and smaller have starting currents within the requirements of central stations and manual or magnetic, full voltage control may be used.

The normal torque, low starting current motor has a starting current which is approximately 20 per cent less than the normal current motor on

full voltage and well within the required current limits on 30 hp sizes and smaller. This motor, therefore, lends itself to across-the-line control because no current limiting equipment is necessary. In selecting motors for fans, pumps, or blowers, it should be noted that while the cost of the normal starting torque, low starting current motor is higher, the cost of full voltage control is lower, so that the total cost of low starting current motors with across-the-line control is lower.

A magnetic starter with low voltage and thermal overload protection gives the most satisfactory service. These switches may be controlled by remote push button stations, thermostats, or pressure switches to meet the requirements of any particular installation.

The high torque, low starting current motor has a starting current approximately 10 per cent less than the normal torque, low starting current motor when started on full voltage. These motors, most commonly used on compressor drive, can be started directly across-the-line with manual or magnetic starters.

Adjustable varying speed motor control by terminal voltage regulation requires a tap-changing switch manually or magnetically operated. Such a control switch operates to alter the voltage applied to the motor by contacting different auto-transformer voltage-ratio taps or by changing the amount of resistance inserted in the primary or line circuit.

MULTISPEED MOTOR CONTROL

To make an installation more flexible, multispeed motors are available with two, three or four speed designs, with variable torque, constant torque or constant horsepower characteristics. Multispeed may be started by means of manual or magnetic starting equipment.

When using automatic magnetic control with two, three, and four speed separate winding or consequent pole motors, control is obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button, which is known as selective speed control. It is commonly used in the smaller theater installations where the fan and motor are located backstage and the speed control is located in the lobby.

Magnetic multispeed motor controllers may also be provided with a compelling relay which makes it necessary that the operator press the first speed button before regulating the motor to the desired speed. This assures the operator that the motor is always started at low speed before the motor is adjusted to one of the higher speeds. Starting on low speed limits the starting current to the starting current of the low speed winding, and therefore, permits the use of motors in sizes larger than ordinarily permitted by central stations for full voltage starting.

Timing relays, which provide for automatic acceleration, may be used for control. With the automatic acceleration feature, it is only necessary to press the button for the desired speed. The motor will always start in low speed and automatically step up to the desired speed.

Where the change of speeds does not occur at regular intervals, and where it is only necessary to change from one speed to another to take

care of seasonal requirements, a manual drum speed selector may be used. This drum is used to select the proper motor speed while an automatic starter is used to start and stop the motor.

The smaller size speed selector drums rated 10 hp at 220 volts and smaller may also be used as a motor starter to make and break the current, as well as serving as a speed selector device. Reversible or non-reversible drums may be supplied depending on the requirements of the installation.

In the large size drums, a separate contactor must be provided to make and break the current. The contactor may be any approved starter. Overload and low voltage protection may be accomplished by using a magnetic starter. No push button station is required, the handle switch on the drum having the same characteristics as a three wire push button station.

In selecting two speed motors for fan, pump, blower, or compressor drive it will be found that the two winding motors are more expensive than the single winding. The control for two speed, two winding motors is more economical and the combined price of the motor and contactor is only slightly higher. Because of the better performance of the two speed motor and the factor of safety in having two independent motor windings, the increased cost is considered worth the difference.

SLIP RING MOTOR CONTROL

When close speed regulation and low starting current are required slip ring or wound rotor motors are used. Slip ring motors are built for two classes of service, constant speed and adjustable varying speed. The motors for the two classes of service are identical, the only difference being in the secondary control used with the motors. Control for both primary and secondary of a slip ring motor is required.

The primary control for a constant or adjustable speed is the same type as used with squirrel cage motors. Manual or magnetic starters, across-the-line type, may be used depending on the installation.

The starting current and starting torque of a slip ring motor are almost entirely dependent on the amount of resistance in the secondary control and in the manner in which the secondary control is operated. The *National Electric Manufacturers Association* has adopted service classifications which allow a selection of resistors permitting a starting current on the first contact of resistance varying from approximately 25 per cent of full load current to approximately 200 per cent of full load current or more, and permitting the resistor to remain in the secondary circuit of the motor for a period varying from not more than 15 seconds during an interval of operation from 4 minutes to continuous.

Speed regulation of a slip ring motor is obtained by inserting resistance in the secondary circuit and usually provides for a 50 per cent speed reduction when the motor takes its full rated current at normal speed. As resistors are supplied for both fan duty and constant torque duty, care should be taken in selecting the proper resistors.

Slip ring motors when used with centrifugal pumps and fans should have fan duty resistors. Because of the low current inrush of the fan and pump

load a starting resistor *NEMA* classification No. 15 may be used. For speed regulation resistor, classification No. 93 should be selected. On a compressor drive using an unloader, a constant torque resistor classification No. 15 should be used. If the compressor is started under load, *NEMA* classification No. 56 or 76 is used. For constant torque speed regulation, resistor No. 95 is used.

SINGLE PHASE MOTOR CONTROL

Where three phase current is not available or where single phase operation is preferred, then single phase repulsion induction, capacitor type or multispeed single phase motors may be used. Since the starting currents of all single phase motors are required to be within the starting-current limits established by the local power supply company, a suitable type of starter may be chosen from the following selection:

1. Enclosed two pole manually operated motor starters with thermal overload protection.
2. Enclosed two pole automatic motor starter operated by a push button, thermostat or similar device, with thermal overload relay and low voltage protection.
3. A manual or magnetic resistance type starter with low voltage protection.
4. A manual or magnetic control for pole changing motors and for adjustable varying speed motors using an auto-transformer or resistance in the primary circuit to obtain line (or terminal) voltage drop.

In selecting across-the-line control for single phase capacitor type motors it is usually very desirable to use three pole across-the-line starters. Control for multispeed, single phase capacitor motors may be selected from tables on three phase rating when consideration is given to the increased current and the necessary switching of connections.

Chapter 36

AIR CONDITIONING IN THE TREATMENT OF DISEASE

Operating Rooms, Reducing Explosion Hazards, Nurseries for Premature Infants, Fever Therapy, Cold Therapy, High Temperature Hazards, Control of Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

IN the past few years air conditioning has made considerable progress as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, heat therapy, cold therapy, oxygen therapy, X-ray rooms, the control of allergic disorders, and for the physiological effects in industry.

OPERATING ROOMS

The widest application of air conditioning in hospitals is in operating rooms. Complete air conditioning of operating wards is important because winter humidification helps reduce the danger of anesthetic gases, summer cooling with some dehumidification is needed to eliminate excessive fatigue and to protect the patient and operating personnel, and finally, filtering for the removal of allergens from the operating room air.

Reducing Explosion Hazard

Explosion hazards in operating rooms began with the introduction of modern anesthetic gases and apparatus. Ether administered by the old drop method is still regarded as comparatively safe; but when mixed with pure oxygen or with nitrous oxide in certain concentrations the explosion hazard may be as great as with ethylene-oxygen, or cyclopropane-oxygen mixtures[1]. (See Table 1.)

During the course of ethylene anesthesia the mixture, usually 80 per cent ethylene and 20 per cent oxygen, is so rich that the danger of explosion is slight in the immediate vicinity of the face mask, but leakage of ethylene into the air may accumulate to any lower concentration, and thus introduce a serious hazard. The most dangerous period is at the end of the operation when the patients' lungs and the anesthesia apparatus are customarily *washed out* with oxygen with or without the addition of

[1]Safeguarding the Operating Room Against Explosions, by Victor B. Phillips (*Modern Hospital*, 46, April and May, 1936).

carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration. In either case the mixture would pass through the explosion range and extraordinary precaution is necessary for the safety of the patient and operating personnel.

Copious ventilation from 6 to 12 air changes per hour reduces to some extent the danger from the open drop method but is of little value in the closed system type of anesthetic machine now in common use. However, this abundant circulation reduces the concentration of anesthetic gases to below the physiologic threshold so that the surgeon and his personnel will not be affected.

The most important cause of accidents is probably static sparks which may result from accumulation of frictional charges on the rubber surfaces of the anesthesia apparatus, on woolen blankets, and on the bodies of the operators as they walk on insulated floors, when the humidity is low. Grounding the various parts of the anesthesia apparatus is not entirely

Table 1. Explosive Properties of Anesthetics[a]

Anesthetic	Formula	Density Air = 1	Limits of Inflammability			
			In Air		In Oxygen	
			Lower	Upper	Lower	Upper
Ethylene	C_2H_4	0.97	2.75	28.6	2.90	79.9
Propylene	C_3H_6	1.45	2.00	11.1	2.10	52.8
Cyclopropane	C_3H_6	1.45	2.40	10.3	2.45	63.1
Nitrous Oxide	N_2O	1.52	......	......	Not Inflammable	
Ethyl Chloride	C_2H_5Cl	2.23	4.00	14.8	......	......
Ether-divinyl	$(C_2H_3)_2O$	2.42	1.70	27.0	1.85	85.5
Ether-diethyl	$(C_2H_5)_2O$	2.56	1.85	36.5	2.10	82.0
Chloroform	$CHCl_3$	4.12	......	......	Not Inflammable	

[a]Explosion and Fire Hazards of Combustible Anesthetics (*U. S. Bureau of Mines*, Report of Investigations No. 3443, April, 1939).

effective, so long as rubber remains in use in the conventional equipment. Some form of protective grounding within the apparatus may be a partial solution.

A comprehensive study of the explosion problem and of the general causes and prevention of operating room hazards is being conducted by the *University of Pittsburgh*, the A.S.H.V.E. Research Laboratory, and the *U. S. Bureau of Mines*. The first result of this investigation has been a fruitful attempt to eliminate the explosive range of cyclopropane, one of the best but most difficult gases to handle. The use of helium as a diluent in the total gaseous mixture controls the oxygen concentration by replacement and since its flame quenching qualities are known it is the ideal gas for this purpose. In addition, a gaseous mixture containing helium is more difficult to ignite by electric discharges and this quality also increases the safety factor of anesthetic administration. A more general idea of the mixtures containing cyclopropane, oxygen and helium necessary to produce satisfactory anesthesia is given in Table 2. Clinically and with slight variation, the noninflammable mixtures of Table 2

have produced satisfactory results and samples of gas taken during operation show no tendencies to explosion.

In the absence of more understanding, no single safeguard can be given, but desirable precautions may be classed as follows: (1) to limit the region of the explosive gas mixtures; (2) to make all electric contacts explosion-proof; (3) to avoid building up static charges; (4) to ground those surfaces where charges may be built up; and (5) to discourage accumulation of static electrical charges by humidity control.

Operating Room Conditions

Little is known about optimum air conditions for maintaining normal body temperatures during anesthesia and the immediate post-operative period. An anesthetized patient displays dilatation of blood vessels in the skin resulting in profuse sweating and (it has been believed) inability to regulate body temperature. From this it was concluded that all anesthetized patients suffered considerable heat loss. In spite of this a recent paper[2] reports little more than 0.8 F variation in the rectal temperature during the course of the operation. The severe physiological effects, such as excessive sweating and rapid pulse, of high operating room temperatures on attendants and patients during the hot months signify the need for proper cooling. A comparison of surgeons' statements who operate in both air conditioned and non air conditioned rooms strongly indicates lesser fatigue; and the greater recuperative power of the patient is confirmed by the previously referred to study[3].

Table 2. Noninflammable Mixtures for Anesthetic Use[a]

Mixture No.	Composition, Per Cent by Volume		
	Cyclopropane	Oxygen	Helium
1	15	20	65
2	20	20	60
3	25	25	50
4	30	30	40

[a]Explosive Properties of Cyclopropane: Prevention of Explosions by Dilution with Inert Gases (*U. S. Bureau of Mines*, Report of Investigations No. 3511, May, 1940).

Although the comfortable air conditions for the operatives are not identical with those for the patient a compromise is as a rule not difficult; with a relative humidity of 55 to 60 per cent, temperatures from 72 to 80 F are used. The work just cited, reported that 68 to 70 deg effective temperature not only furnished comfort for the operating room workers but apparently prevented exhaustion of the patient as evidenced by rapid convalescence in the recovery ward. Additional heat may be furnished to the patient locally or by suitable covering according to body temperature in individual cases.

[2]A.S.H.V.E. Research Report No. 1111—Air Conditioning Requirements of an Operating Room and Recovery Ward, by F. C. Houghten and W. Leigh Cook, Jr. (A.S.H.V.E. Transactions, Vol. 45, 1939, p. 161).

[3]Loc. Cit. Note 2.

In an investigation recently conducted at the University of Pittsburgh, in a cooperative research program with the Society, comparative studies were made on the bacterial content of conditioned and non-conditioned operating rooms. From these studies[4] it was concluded that the bacterial content of conditioned operating rooms was considerably less than that of non-conditioned rooms. Although this difference may not be great it is sufficient to demonstrate that properly conditioned spaces with adequate filtration can definitely reduce the bacterial and other foreign substance content in an enclosure.

The increasing incidence of allergies or of their recognition is becoming a factor in the operating room. Operations may be postponed on allergic patients during asthmatic manifestations through fear of complications. The removal of the allergens, therefore, is in some cases an important function of the air conditioning system.

Central system air conditioning plants and unit air conditioners prove satisfactory in operating rooms when producing between 8 and 15 air changes per hour of filtered and properly conditioned air without recirculation during the course of anesthesia. A separate exhaust fan system is as a rule necessary to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and frosting on the glass in cold weather and to minimize drafts. The high air flow of 8 to 15 air changes in operating rooms is desirable for three reasons: (1) to reduce the concentration of the anesthetic to well below the physiologic threshold in the vicinity of the operating personnel, (2) to remove the great amounts of heat and sometimes moisture, from sterilizing equipment if inside the operating room, from the powerful surgical lights, from solar heat, and from the bodies of the operatives, and (3) to provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by careful insulation of sterlizing equipment and by thorough exhaust ventilation of sterilizing rooms adjoining the operating rooms.

A very common complication presumably traceable to operations is pneumonia. The difference in conditions between the operating room and the final hospital destination of the patient, including corridors and elevators, is conducive to post-operative pneumonia. A suggested remedy is a recovery ward where conditions closely approximate those of the operating room and in which the patients remain from one to four days. Satisfactory conditions in the recovery ward not only hasten convalescence, but dispel the fear frequently found in patients who must undergo operations during the hot seasons[5].

Sterilization of Air in Operating Room

Of considerable significance to operating rooms and contagious wards is the use of ultra-violet radiation for sterilizing the air[6]. Results reported[7]

[4]Report on Air Conditioning in Surgery, by W. Leigh Cook, Jr. (Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, 1940).

[5]Report of the Committee on Air Conditioning (*The American Hospital Association*, 1937, p. 2).

[6]Air-Borne Infection and Sanitary Air Control, by W. F. Wells (*Journal Industrial Hygiene*, 17:253, 1925).

[7]Sterilization of the Air in the Operating Room by Special Bactericidal Radiant Energy, by Deryl Hart (*Journal Thoracic Surgery*, 6:45, 1936).

would indicate that the post-operative temperature rise of patients during the first few days is in most instances caused more by bacterial contamination of the operative wound than by the absorption of blood and traumatized tissues. Operating room infections, which were quite frequent before the installation of special ultra-violet lamps, are apparently being reduced.

Direct ultra-violet radiation is distinctly advantageous in sterilizing not only the site of operation but also wounds to prevent the spread of infection. In infants' wards, contagious disease wards, and even in school rooms, the sterilizing effects are definitely known. Whether an air conditioning system with ultra-violet installations in the ducts is a feasible procedure is controversial; but it would appear that this indirect method is not as satisfactory as the direct in the light of present reported knowledge.

TABLE 3. NET MORTALITY OF PREMATURE INFANTS ACCORDING TO HUMIDITY[a]
Infants Hospital, Boston, Mass.

Cause of Death	Unconditioned Nurseries (1923-1925)	Conditioned Nurseries (1926-1929)	
	Natural Humidity	Relative Humidity	
		25-49 Per Cent	50-75 Per Cent
	Per Cent Mortality	Per Cent Mortality	Per Cent Mortality
Acute and chronic infections	26.5	9.7	0.0
Congenital deformities	1.2	0.0	0.7
Unclassified	1.2	4.8	0.0
All causes	28.9	14.5	0.7

[a]Excluding cases with multiple congenital anomalies incompatible with life, and also deaths occurring within 48 hours after admission to the hospital.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is because their heat regulating systems are not fully developed; the metabolism is low and the infants generally exhibit marked inability to maintain normal body temperatures. The resistance to infection is low and mortality rate high.

Air Conditioning Requirements

The optimum air conditions for the growth and development of these infants were determined by extensive research[8] at the Infants Hospital, Boston, Mass., using four valid criteria, namely, stability of body temperature, gain in weight, incidence of digestive syndromes, and mortality. Individual temperature requirements varied widely (from 72 to

[8]The Premature Infant: A Study of the Effects of Atmospheric Conditions on Growth and on Development, by K. D. Blackfan, C. P. Yaglou and K. McKenzie (*American Journal Diseases of Children*, 46: 1175, 1933).

100 F) according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 per cent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F and 65 per cent relative humidity was found to fulfill satisfactorily the requirements of the majority of premature infants. Additional heat for weak (or debilitated) infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery, and the temperature adjusted according to individual requirements. In this way multiplicity of chambers and of air conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Infants Hospital were kept at relative humidity between 25 and 50 per cent for two weeks or longer, the body temperature became unstable, gain in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 per cent relative humidity gave satisfactory results over a period of years. The effect of humidity on mortality is shown in Table 3. The initial physiologic loss of body weight (loss occurring within first four days of life) was found to vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 per cent of the birth weight; in the conditioned nurseries it was 8.9 per cent with 25 to 49 per cent relative humidity, and 6.0 per cent with 50 to 75 per cent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery and minimum in the conditioned nurseries under high humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 per cent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 per cent) in the same nurseries, and in the old nurseries prior to the installation of air conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea, persistent vomiting, diminishing gain or loss of body weight, and other symptoms, were generally from two to three times as high under low than under high humidity.

Summarizing, the best chances for life in premature infants are created by maintaining a relative humidity of 65 per cent in the nursery and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air Conditioning Equipment

Most of the installations now in use are of the central system type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A high ventilation rate, between 15 and 25 air changes, is desirable to remove odors and

maintain uniformity of temperatures in extremes of weather. Recirculation is not used extensively in these wards owing to odors and the possibility of infection.

FEVER THERAPY

Artificial production of fever in man is an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by destruction of the invading organism within the safe limit of human temperatures, or indirect in the case of heat resistant organisms, by general mobilization of the defensive mechanisms of the body, which retard or neutralize the activity of pathogenic bacteria and their toxins.

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 8 hours at a time. The total period of fever treatment varies with the type of the organism involved from a few hours to 50 or more.

The diseases which respond favorably to artificial fever therapy are gonorrhea and its complications, (which include arthritis, pelvic infections in women, and involvement of the eye), syphilis, chorea, infectious arthritis (non-gonorrheal), encephalitis, and some forms of asthma. There are other conditions which show promise under this treatment; but the most striking results are seen in gonorrhea and syphilis, since the causative organisms can be destroyed at temperatures compatible with human life[9].

Equipment for Production of Fever

Various means have been tried for producing artificial fever, including injections of various crystalloid or colloid substances, bacterial products of typhoid and malarial organisms; a number of physical methods, such as hot baths, radiant heat, diatherm, radiothermy, and in the last few years, air conditioned chambers. The relative advantages and disadvantages of various methods have been discussed in a recent paper[10]. The results by the use of air conditioned cabinets have not been fully explored, and it is therefore difficult to determine all the advantages and disadvantages of the value of air conditioning at this time.

In the earlier studies of the Society[11], temperatures were elevated more easily using saturated atmospheres. A fever therapy apparatus[12] using these same principles has proved efficient as a means of inducing and maintaining fever in a body with small likelihood of burns because of the comparatively low dry-bulb temperatures. This saturation factor is in great use today where fever is created by induction currents by

---

[9]Report of the First Year of Fever Therapy Research by the Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, 1938.

[10]Fever Therapy by Physical Means, by Frank H. Krusen and E. C. Elkins (*Journal American Medical Association*, 112: 1689-1696, April 29, 1939).

[11]A.S.H.V.E. RESEARCH REPORT No. 654—Some Physiological Reactions of High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 129).

[12]A.S.H.V.E RESEARCH REPORT No. 1054—Fever Therapy Induced by Conditioned Air, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 131). A.S.H.V.E. RESEARCH PAPER—Fever Therapy Locally Induced by Conditioned Air, by M. B. Ferderber, F. C. Houghten and Carl Gutberlet (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1940, p. 323).

placing the body in an electrical field. When the optimum body temperature has been reached by electrical induction, the atmosphere of the enclosure is kept at saturation to prevent heat loss, thus maintaining the patient's temperature at the desired point. Other apparatus[13] which use electric heaters, centrifugal fans, and a water container for humidification have been used in the past, but the more recent trend is toward saturation with a lower dry-bulb temperature.

When heat is necessary in treating legs or arms, such media as short or long wave diathermy, infra-red, water baths, etc. have been used extensively. A recent development, a saturated atmosphere heating unit, similar to one previously described[14] has proven satisfactory, because heat may be administered over longer periods which render deep heating possible without fear of burns or shocks[15]. Local heating has been somewhat satisfactory in relieving the painful symptoms of peripheral vascular disease.

The final criteria for the use of fever therapy may be changed because of the introduction of certain drugs which appear prominent in the experimental treatment of some diseases for which fever therapy has been efficacious.

COLD THERAPY

In contrast to fever therapy the use of cold as a means of treatment is being investigated. From the available literature[16] the chief virtues of cold therapy (cryotherapy) are the reduction of pain due to extensive cancer and the possibility that the process may be arrested. For a localized lesion, ice water between 36 to 48 F is circulated through tubing at the site of the disease for periods ranging from 4 to 48 hours. A later development was the principle of hibernation during which time the patient is kept in an air conditioned space with an environmental temperature between 50 to 60 F for five days. The body temperature is reduced below the critical level of 95 F to as low as 80 F. Most of the vital processes of life are at very low ebb and this period simulates the hibernation of the wild animals. Although relief of pain is reported it remains to be seen to what extent this form of treatment will be used.

HIGH TEMPERATURE HAZARDS

Heat disease is now classified as heat exhaustion, heat cramps, and heat stroke[17]. The last was formerly thought to be the result of high temperatures, but in the light of present knowledge is considered a neurologic defect. This inability to control temperature is most frequently seen in diseases of the nervous system, and heat stroke, therefore, is not so much a result of environmental temperatures as it is some intrinsic defect in the mechanism itself. The hazards of high temperatures are

[13]Artificial Fever Therapy of Syphilis, by W. M. Simpson (*Journal American Medical Association*, 105: 2132, 1935).

[14]Loc. Cit. Note 11.

[15]Saturated Atmospheres in the Treatment of Injuries, by M. B. Ferderber (*Industrial Medicine*, 8: 256-259, June, 1939).

[16]Temperature Factors in Cancer and Embryonal Cell Growth, by L. W. Smith, and Temple Fay (*Journal American Medical Association*, Vol. 113: 653-660, August, 1939).

[17]Heat Disease: Clinical and Laboratory Studies, by M. W. Heilman and E. S. Montgomery (*Journal of Industrial Hygiene and Toxicology*, 18: 651-666, November, 1936).

not easily understood. It is difficult to say whether a repeated rise of 1 or 2 deg of body temperature is dangerous or whether short exposures at high temperatures are more harmful than longer exposures at lower temperatures. A new concept is evident in finding an increase in leucocytes (white cells) of the blood in workers subjected to high temperatures. These leucocytes are defensive factors which are increased when infection invades a body. A rise in temperature and leucocyte count indicates body defense in the presence of disease. Since a recent study[18] showed that both temperature and cell count were increased, the question arises whether long exposures to very high temperatures might not cause exhaustion of these defense mechanisms.

ALLERGIC DISORDERS

Although there is some division of opinion over the ultimate cause of allergy, the prevailing belief is that it is due to an inherited or acquired hypersensitiveness to pollen or other foreign proteins in certain individuals who react abnormally to the offending substance. The reaction may be induced by inhalation, eating, or absorption (through the skin) of the allergens. Some of the clinical manifestations are hay fever, asthma, eczema, and contact dermatitis.

Symptoms of Hay Fever and Asthma

The respiratory tract is the site of probably the most usual allergic manifestations, the so-called hay fevers and asthma. In hay fevers, the nose and eyes are red and itchy, and there is considerable discharge. Nasal obstruction is the most common and most distressing symptom. The severity of the symptoms varies widely from day to day depending chiefly on the amount of pollen in the air.

Seasonal asthma comes in attacks. The most popular theory concerning the mechanism of action is that the offending substance irritates the nerve endings in mucous membranes of the respiratory tract, causing spasmodic contraction of the small bronchioles of the lungs, which interferes with breathing, particularly with expiration. Non-seasonal allergic disturbances are sometimes attributed to house or street dusts, fungi, odors, animal dander, irritating gases, and heat or cold, particularly sudden temperature changes. It is often stated in the literature that heat regulation in asthmatic individuals is likely unstable, with a tendency toward the subnormal. Many allergic cases who are apparently well, develop their attacks when cold weather appears, or upon changing from warm to cool outdoor air.

Air Conditioning Apparatus

In recent years considerable effort has been directed toward the elimination of the principal cause of allergy from the air of enclosures by filtration or other air conditioning processes capable of removing pollens, in the hope of providing relief to individuals who fail to respond to medical treatment (desensitization or immunization).

Paper or cloth filters, mounted in inexpensive window or floor units, prove quite satisfactory, but since dust and smoke frequently cause

[18]A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59).

asthmatic attacks, it is necessary that an air filter, to be of full value in the treatment of asthma, must remove all dusts and pollens regardless of size or amount. An electrostatic cleaner has proved extremely efficient in removing particles of 15 to 20 microns and smaller, besides dusts and smoke[19].

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification appears to be desirable. A comfortable temperature between 70 and 75 F and a relative humidity well below 50 per cent proved satisfactory[20]. Direct drafts, overcooling or overheating are apt to initiate or aggrevate the symptoms.

Limitations of Air Conditioning Methods

The results obtained with air filtration or other air conditioning processes in the control of allergic conditions are fairly comparable to those obtained by desensitization treatment so long as the patients remain in the pollen free atmosphere. But while specific desensitization is preventive and in a few instances curative, for all practical purposes filtration gives only temporary relief. With rare exceptions, the symptoms recur on exposure to pollen laden air. Moreover the usefulness of air conditioning methods is limited because all cases are not caused by air-borne substances. Cases of bacterial asthma do not respond at all to the treatment with filtered air.

Despite these limitations air conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions almost immediate relief. Pollen cases are usually relieved of most of their symptoms within 1 to 3 hours after exposure to properly filtered air.

A pollen-free atmosphere is especially valuable in cases where desensitization has given little or no relief, and where desensitization is not advisable owing to intercurrent illness. On the whole, conditioning methods are considered to be a valuable adjunct in medical diagnosis and treatment of allergic disorders.

OXYGEN THERAPY

Oxygen therapy is the principal measure employed for preventing and relieving the distressing symptoms of anoxemia, which is a deficiency in the oxygen content of the blood. Some of the more important conditions in which oxygen treatment is believed to be beneficial are pneumonias, anemia, heart affections, post-operative pulmonary disturbances, certain mental disturbances, asphyxia, asthma and atelectasis in new-born infants.

The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. The oxygen rich atmosphere in these enclo-

---

[19]Air Cleaning as an Aid in the Treatment of Hay Fever and Bronchial Asthma, by Leo H. Criep and M. A. Green (*Journal of Allergy*, 7: 120, January, 1936).

[20]The Effect of Low Relative Humidity and Constant Temperature on Pollen Asthma, by B. Z. Rappaport, T. Nelson and W. H. Welker (*Journal of Allergy*, 6: 111, 1935).

sures is therefore reconditioned in a closed circuit by removal of excess heat, moisture, and carbon dioxide given off from the occupants being treated.

Oxygen Tents

In oxygen tents the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means of a pressure reducing valve and flow meter. In an inadequately cooled tent, high temperatures and humidities are inevitable, increasing the discomfort of the patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good. An ice melting rate of approximately 10 lb per hour gives satisfactory results in patients with fever in a medium size oxygen tent.

Oxygen tents are somewhat confining to the patient; the restless type of person is difficult to control, and the delirious, impossible to control. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 per cent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. The direct advantages are portability and low cost.

Oxygen Chambers

The conventional oxygen chamber is an air-tight sheet metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service, to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from outside the chamber.

The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or air drying agents. The gravity system includes a bank of cooling coils controlled thermostatically, which dehumidify and cool the air. The cool air falls over trays of soda lime at the bottom of the coils, to remove the carbon dioxide given off by the occupants. A heater at the base of the opposite wall warms the air to the desired temperature. Ordinary industrial oxygen is introduced from storage tanks outside the chamber and the concentration is regulated according to the prescription of the physician. The only change of air in the chamber is that taking place by air leakage through the trap doors.

The chief objections to the gravity circulation system are stratification of cold air near the floor and accumulation of odors, which may require the use of activated charcoal, or an excess of oxygen for dilution of the air in the chamber.

The fan circulation systems include compact extended surface coolers, heaters, and sometimes air-drying beds installed outside the chamber for the removal of moisture.

The temperature and humidity requirement in oxygen therapy depend primarily upon the physical condition of the patient, and secondarily

upon the type of disease. In pneumonias, according to Bullowa[21], prescribed conditions should be an effective temperature of 66-68 F, humidity of 50 per cent, air movement of not less than 50 linear feet per minute, oxygen concentration of 50 per cent, and carbon dioxide of less than 1 per cent.

Oxygen chambers are more comfortable than oxygen tents. The patients receive unhampered medical and nursing care, and the oxygen concentration, the temperature and humidity can be adequately controlled at any desired level. The chief disadvantages are high initial and operating costs in comparison with oxygen tents or with the nasal catheter method of oxygen administration. The nasal catheter method is the simplest and most inexpensive of all but it may cause considerable discomfort to the patient and it is not satisfactory for continuous administration nor for restless or delirious patients. Moreover, oxygen concentrations greater than 40 per cent in the inspired air are difficult to maintain, although concentrations as high as 48 per cent have been obtained.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of a large hospital involves a capital investment and running expense which may not be justified. In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of usual heating in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; it will also help in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the window openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may preclude the cooling of the entire building, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed. Objectionable noise is an important drawback to the use of self-contained units, but the difficulty is gradually being overcome by improvements in design.

In the North and certain sections of the Pacific Coast, cooling is needed but a few days during summer, while in the South, it can be used to advantage from May to October, and in tropical climates almost continuously throughout the year.

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of fevers in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, and in a variety of other ailments which often accompany summer heat waves.

Considerable research is in progress on the influence of air conditioning upon a wide variety of diseases such as pneumonia, upper respiratory diseases, tuberculosis, arthritis, nervous instability, hyper-thyroidism, essential hypertension, skin diseases, and vascular disorders.

---

[21]The Management of Pneumonias, by J. G. M. Bullowa, 1937, p. 260.

Chapter 37

TRANSPORTATION AIR CONDITIONING

Railway Passenger Car Ventilation, Method of Air Distribution, Air Cleaning, Winter and Summer Air Conditioning, Humidity and Temperature Control, Summer Air Conditioning for Buses and Automobiles

THE principles of air conditioning used in connection with stationary applications such as stores, restaurants, hospitals, theaters, and homes are in general applicable to such mobile applications as railway passenger cars, passenger buses, automobiles, and ships. However, the equipment used for these mobile applications, with the possible exception of those on board ship, differs from that used for stationary purposes in that it must meet additional requirements. Especially important are the features of compactness with the retention of ready accessibility for quick inspection and servicing, and low weight. Freedom from vibration which could be transmitted to the supporting vehicle and thus to the passengers is essential.

RAILWAY PASSENGER CAR VENTILATION

In non air-conditioned cars, ventilation is accomplished by exhaust fans, roof ventilators, and open doors and windows. This practice provides an ample supply of outside air but does not prevent the entrance of smoke, cinders, and dirt.

An average passenger car contains approximately 5000 cu ft of air and may seat as many as 80 passengers. The occupants are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from their breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidification, but the other constituents can be successfully handled only by proper ventilation and air cleansing. In the average car from 2000 to 2500 cfm should be circulated by the air conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from the outside. The amount of outside air required depends upon the type of car, number of passengers, air temperature, humidity, odors, and whether or not occupants are smoking, and will vary from 15 to 90 per cent of the total air circulated.

Careful attention must be exercised in specifying the rate of outside air taken in so as to fit the type of service adequately and yet not to supply more ventilation than is necessary. Conditioning this outside air is a major factor in determining the size of both summer and winter conditioning equipment. With present average ventilation requirements, about 30 per cent of the cooling equipment and sometimes as high as 50

per cent of the heating equipment is necessary to handle only the outside air load.

For normal conditions, 10 cfm of outside air per passenger is sufficient. When smoking is permitted, at least 15 cfm should be admitted. In some of the dining cars and deluxe sleeping cars, outside air rates as high as 20 and 30 cfm per occupant are used.

Method of Air Distribution

The fact that the amount of space devoted to railway passengers may be as low as 60 cu ft per person (ranging as high as 190 cu ft per person), coupled with the high air flow rates made necessary by severe ventilation and sun loads, makes the problems of air distribution and air delivery in railway cars critical ones.

Various methods may be used to distribute the air delivered to the interior of the car by the circulating fan or blower. The methods commonly used are:

1. A duct lengthwise along the center of the car.
2. One or two side ducts built on the outside of monitor-roofed cars, or on the inside of turtle-backed or arched-roofed cars.
3. Free discharge at the end bulkheads, or by free discharge from a unit placed overhead in the center of the car, discharging toward the ends. This bulkhead delivery system while inexpensive, is apt to cause complaints due to drafts, and accordingly, is not being favored.

Delivery grilles and plaques are used, and are often designed to give considerable entrainment and mixing to avoid cool drafts.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near the ceiling can be broken up by having the incoming air directed along the ceiling in all directions at a velocity somewhat higher than that used for the rest of the car. The air should be exhausted from the room by a fan or through a grille to the washroom or lavatory, and then outside by a fan in a ventilator.

For compartments an adjustable supply duct outlet grille of suitable size and design should be provided and provisions made in the door or partition for the removal of the air to be recirculated.

Lower berths in sleeping cars and office cars should be provided with an adjustable air outlet which will discharge the amount of air desired at low velocity in any direction so that the occupant can regulate the ventilation to meet his own requirements.

In cars containing but one or two rooms or compartments, satisfactory results may be obtained by discharging the air directly from the conditioning unit into the upper part of the car. Care must be taken to have a proper discharge velocity. If the velocity is too low, the air will drop before reaching the end of the car and if too high it will discharge against the end bulkhead and be reflected back. Care must be exercised to secure proper circulation, otherwise objectionable drafts will be experienced.

The recirculating air grilles are usually of the straight flow type, and should be located so that objectionable drafts will not be created by the return air. The outside air intakes, located in the car vestibule, on the side of the car, or on the roof of the car, depending upon the location of the cooling coils, should be of ample size to permit the entrance of suf-

ficient outside air. On many of the recently air-conditioned cars, there are no dampers or shutters at the outside air intakes, the percentage of outside air being controlled by blocking the flow through the recirculating grille.

Air Cleaning

All of the air circulated by the blower is filtered before passing over the cooling coils. In some cars the outside and recirculated air are filtered separately before mixing, while on others the air from the two sources is mixed before passing through a common filter. Filters in use are made of metal, wool, cloth, spun glass, hemp, paper, hair, and wire screen. Most filters have a viscous coating of oil for greater cleaning efficiency. Some types may be cleaned, retreated, and returned to service while other types are discarded when dirty.

RAILWAY PASSENGER CAR WINTER AIR CONDITIONING

The majority of cars in service use steam from the locomotive or from a head-end, oil-fired boiler as a source of energy for winter heating. In some instances electrical energy from either a head-end generating set or motive power supply is utilized for resistance heating. In still other cases electrical energy and waste heat from individual car engine-generator sets is employed. The peak heating loads which depend largely upon the amount of insulation used in the car, the type of windows (whether single or double glazed), and the ventilation rate, may vary from 150,000 to 250,000 Btu per hour.

In order to temper the cold outside air, about 30 to 50 per cent of the total heat energy required is distributed by means of finned coils or resistance heaters located in the outside air duct. The remainder is usually transmitted to the car air by finned tubing located along the sides of the car near the floor, thus preventing cold convection currents falling from the car windows from reaching the feet of the passengers.

RAILWAY PASSENGER CAR SUMMER AIR CONDITIONING

Three general types of cooling or refrigerating equipment are being used in the 11,700 railway cars which are now air conditioned in the United States. Of these 3,900 are ice-activated, 1,900 use steam jet systems, and 5,900 employ mechanical compression schemes. These systems which functionally are identical with those used for stationary applications (see Chapter 24) are modified in design to meet the requirements of mobile service. Contrasted with stationary applications of summer conditioning equipment, the use of water as a final means of heat disposal from condensers cannot be resorted to because water in such quantities cannot be transported economically. Accordingly, air cooled or evaporative condensers are always used, with the result that mobile cooling equipments operate at higher temperature, pressure, and power requirement levels than stationary equipment.

The maximum cooling and dehumidifying load which depends largely upon the amount of insulation, the type of windows, the ventilation rate, the sun intensity, and the number of passengers may vary from 60,000 to 96,000 Btu per hour.

An average ice-activated system for such capacities uses about 500 pounds of ice and 1.2 kw per hour. The increase in car weight due to such a system is approximately 8500 lb.

The same service from a steam jet system is obtained with the expenditure of 230 lb of steam and 3.3 kw per hour, with an added weight per car of 11,000 lb.

The mechanical compression systems, all of which use dichlorodifluoromethane as a refrigerant, may be classified by several types depending on the method of driving the compressor. The source of power for driving the compressor (approximately 10 hp) is complicated by the necessity of obtaining this power at all times whether the car is in motion or standing still on the right-of-way or in a terminal where auxiliary power plug-ins are available. In those cases where compressors are driven from car axles, additional refinements in the drive are necessary in order that a nearly constant cooling capacity may be obtained from a variable speed power source. Numerous combinations of electrical generating schemes for generating sufficient electrical energy from the car axle for lighting, ventilation, and summer air conditioning are in use, and their operation is closely interlocked with compressor demands, need for pre-cooling, battery charging, etc. It is difficult therefore to state the additional weight imposed on a car because of such a compression air conditioning system, but it is probably in the vicinity of 6000 lb. These systems, depending mostly upon the locomotive for supplying power for operation, impose a load which may amount to 10 per cent of the capacity of the locomotive.

Several schemes for relieving the locomotive of this compression load are used. Some of the articulated trains, which run as unit equipment—the same cars always in the same train—employ a head-end, engine-generator combination for supplying power to compressor motors. In other cases engine-alternators on individual cars are used to supply alternating current power to compressor motors, as well as to supply all power for car lighting and auxiliaries. Engine-compressor combinations on individual cars provide attractive low weight equipment where continuous engine operation is permissible under all circumstances. Diesel engines and propane engines are used for these purposes, and such engine-driven units have the additional advantage of being able to use waste engine heat either for modulating refrigeration with a reheat cycle or for car heating purposes.

RAILWAY PASSENGER CAR HUMIDITY AND TEMPERATURE CONTROL

The temperature to be maintained in a car depends upon the outside temperature and the humidity desired inside the car. With a low humidity it is necessary to maintain a higher temperature to establish a desirable comfort condition. Little humidity control has been attempted on cars up to the present time. A certain degree of automatic humidity control is secured with cooling, but the relative humidity obtained depends largely upon the temperature of the evaporator, which should be below the dew-point temperature of the air. With certain outside atmospheric conditions it may not be possible to operate the conventional equipment with a sufficiently low evaporator temperature to reduce the humidity

without dropping the temperature too low. One method has been developed whereby the evaporator temperature is carried below the dew-point a sufficient amount to insure dehumidification and then the cold air is heated to the proper temperature by passing it over coils through which part of the high temperature liquid from the condenser is by-passed. Such a system is costly and has not been generally applied. The reheat cycle obtainable from waste engine heat may be used to good advantage in reducing the humidity without reducing the dry bulb temperature.

During the heating season humidification is desirable from a comfort standpoint, but unless properly controlled, condensation will appear on the windows. A steam or water spray controlled by a humidistat will provide the necessary moisture for humidification. There are several cars with this feature now in use.

Temperature control for the most part obtained by rugged thermostats and relays capable of withstanding vibrations attendant with mobile service is usual equipment.

Manual zone control for varying outdoor conditions, as well as controls which regulate the car temperature automatically in accordance with outdoor conditions, are employed.

Simplified controls from the standpoint of operation by train crews and especially from the servicing viewpoint are very desirable. The control of summer temperatures is accomplished mainly by cycling the complete cooling system; however, modulation is being effected by using multiple evaporators in which a fixed portion may be cut out of the system. In the engine-driven equipments, modulation is obtained by changing engine speed.

For further information on controls, see Chapter 33.

PASSENGER BUS SUMMER AIR CONDITIONING AND VENTILATION

The highways in the United States are now traveled by about 1000 summer air conditioned passenger buses. Many of the facts stressed in connection with the design and installation of summer conditioning equipment in railway cars are even more important in these newer vehicles. Weight and space limitations are more stringent, and the problem of circulating from 900 to 1200 cfm of air in coaches carrying from 25 to 40 passengers with about 35 cu ft of space per passenger without drafts is no easy one.

Some bulkhead delivery systems have been used, and while the overhead package racks have served to break up drafts to some extent, these installations are not gaining in popularity. Longitudinal ducts in the corners above the package racks are sometimes used to carry conditioned air to a series of outlet louvers along the top of the windows. Other designs provide for false spaces below the package racks which serve as ducts to distribute air to either entrainment grilles in the bottom of the racks or distributing slots at the edges of the package racks. Some coaches employ a false ceiling to provide a duct, with delivery taking place from numerous perforations in the ceiling.

Return air grilles and filters are usually located near the rear ceiling where the evaporator is placed. Outside air intakes and filters are located

preferably near the front of the vehicle so as not to contaminate this supply with exhaust fumes and road dust. Of the 30 cfm circulated per person, about 8 to 10 cfm are outside air and the remainder is recirculated. Power for the motor driving the centrifugal fans is obtained from the bus battery.

More recently a coach design has been brought out which provides for a number of return air outlets below the seats; these permit return air to enter a longitudinal duct below the floor. The filters and evaporator are located in this duct near the front of the vehicle. In this instance a central heating coil utilizing waste heat from the coach engine is also located in this duct. Conditioned air is delivered through a pair of vertical ducts to a package rack distribution scheme.

Summer conditioning systems for these vehicles range in cooling capacity from 36,000 to 48,000 Btu per hour. Mechanical compression systems using dichlorodifluoromethane are used, and are powered by water cooled, gasoline engines of approximately 14 hp.

Complete systems add from 800 to 1300 lb to the weight of a coach. Sometimes an auxiliary generator driven by the air conditioning engine is used which serves to help charge the bus battery and thus offsets the power drain imposed by the ventilating blower. Belted reciprocating compressors and direct driven V-type and rotary compressors are used, with engine speeds up to about 1800 rpm. Air cooled condensers for this service require about 5000 cfm of outdoor air, and this is provided by either centrifugal or propellor type fans belted or direct driven by the air conditioning engine. Preventing noise and vibration from affecting passengers is of vital importance. Installations must be made so that quick daily servicing of the engine is possible. In all cases fuel is obtained from the main bus tanks, and in some cases the main engine jacket water cooling system is used to cool the air conditioning engine.

In the more deluxe equipment after the driver has started the air conditioning engine by means of its own cranking motor, the engine speed is modulated automatically as the refrigeration demand is partially met, and if this demand is then fully met, the engine is stopped thermostatically. Restarting when the cooling thermostat is no longer satisfied is accomplished either automatically or manually. The various protective and automatic devices on the refrigerant and engine systems make some of the bus air conditioning control systems quite complicated.

AUTOMOBILE SUMMER AIR CONDITIONING

Recently summer air conditioning has been applied to automobiles. The average present day automobile with little insulation, large, single glazed window areas, and high infiltration and exfiltration losses requires about 15,000 Btu per hour of cooling capacity. One system utilizes a reciprocating compressor belted from the main engine fan shaft thus operating at varying speeds up to 3000 rpm. The resulting refrigeration capacity varies from about 6000 Btu per hour at idling speed to 24,000 Btu per hour at maximum car speed.

A dry air condenser is placed in front of the engine radiator, and the liquid and suction refrigerant lines run back under the car floor to the

evaporator which is located in back of the rear seat. Conditioned air is delivered into the car just above the shelf near the back of the rear seat. A return grille is provided under the rear seat, and the recirculated air is filtered. Outdoor air is provided by infiltration. Power for the air circulating blowers is obtained from the car storage battery. Equipment of this nature increases the car weight approximately 200 lb.

REFERENCES

RAILWAY PASSENGER CARS

Summary Report on Air Conditioning of Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, November 24, 1936.

Engineering Report on Air Conditioning of Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, April 15, 1937.

Report on Performance and Cost of Operation of 1937 Internal Combustion Engine Mechanical Compression Equipment for Air Conditioning Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, May 1, 1937.

Report on Relative Performance of Air Filters, by Mechanical Division, *Association of American Railroads*, January 15, 1938.

Air Conditioning of Railroad Passenger Cars, by L. W. Wallace and G. G. Early, Jr. (*A.S.M.E. Transactions*, November, 1937).

Improved Methods of Railroad Air Conditioning, by A. E. Voigt (*Refrigerating Engineering*, July, 1940, p. 16).

BUSES AND AUTOMOBILES

Iso-Thermal Manifold and the Conditioned-Air Unit Used on Nash Cars, by E. Monson (*Society of Automotive Engineers Journal*, January, 1938).

Summer and Winter Air Conditioning of Automobiles and Buses, by L. W. Child (*Heating and Ventilating*, March, 1938, p. 38).

Air Conditioning Takes to the Highways (*Heating, Piping and Air Conditioning*, May, 1938, p. 314).

Fleet of Air Conditioned Buses West Carries Demountable Units (*Heating and Ventilating*, May, 1938, p. 30).

Conditioned Air for Automobiles (*Heating and Ventilating*, May, 1938, p. 44).

Air Conditioning of Automobiles and Buses, by L. W. Child (*Society of Automotive Engineers Journal*, June, 1938).

Bus Air Conditioning, by Jerry Hicke (*Heating, Piping and Air Conditioning*, October, 1938, p. 639).

Bus Air Conditioning, by A. J. Mallinckrodt and Lars Hanson (*Refrigerating Engineering*, June, 1939, p. 388).

Problems in Air Conditioning Automobiles, by F. J. Linsenmeyer (*Society of Automotive Engineers Journal*, July, 1939).

Packard Offers Automobile Air Conditioning System (*Heating, Piping and Air Conditioning*, December, 1939, p. 756).

AIRPLANES

Flight in the Sub-stratosphere, by J. C. Edgerton (*Society of Automotive Engineers Journal*, 39:484-6 December, 1936).

Comfort Aloft (*Heating and Ventilating*, October, 1937, p. 40).

Portable Climate, American Buys 12 Plane Air Conditioning Units (*Aviation*, 36:59, September, 1937).

for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture and accurately regulating the final moisture content. Either conditioning or drying are frequently made continuous processes in which the material is conveyed through an elongated compartment by suitable means and subjected to controlled atmospheric conditions.

CONTROL OF RATE OF CHEMICAL REACTIONS

A typical example of the second general classification, that is the control of the rate of chemical reactions, occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that during this process close control of both temperature and relative humidity should be maintained. Temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution and gives a solution of known strength throughout the mercerizing period.

Another well-known example of this class is the *drying* of varnish which is an oxidizing process dependent upon temperature. High relative humidities have a retarding action on the rate of oxidization at the surface and allow the gases to escape as the chemical oxidizers *cure* the varnish film from the bottom. This produces a surface free from bubbles and a film homogeneous throughout.

Desirable temperatures for *drying* varnish vary with the quality. A relative humidity of 65 per cent is beneficial for obtaining the best processing results.

CONTROL OF RATE OF BIOCHEMICAL REACTIONS

In the field of biochemical control, industrial air conditioning has been applied to many different and well-known products. All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a temperature of 80 F. A relative humidity of 65 per cent is maintained so as to hold the surface of the dough open to allow the CO_2 gases formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

Another example of a similar process is found in the curing of macaroni. The flour and water mixture is fermented and dried. As it is necessary to have a definite amount of water present to carry on a fermentation process, the moisture must be removed in a relatively short period to stop fermentation and prevent souring and in such a manner as to avoid setting up internal strains in the mixture. Best results are obtained with the correct cycles of both temperature and humidity.

The curing of fruits, such as bananas and lemons, also comes under this classification. Bananas are treated somewhat differently and to accomplish the required results, a cycle of temperatures and relative humidities is used. The starches in the pulp of the fruit must be changed and the

ventilation a 2000 fpm exhaust air velocity at the slot face is advisab In addition, the duct should not be required to draw the air laterally f a distance of more than 18 in. and the level of the solution should be ke 6 to 8 in. below the top of the tanks.

Flexible Exhaust Systems

The flexible exhaust tube method may be advantageously used f removing dust or fumes. Flexible tubes having one end connected to a exhaust system and a slotted hood attached to the other end may shaped at will to fit in with industrial processes without affecting the ea of operation. Efficient dust or fume removal may be had with use relatively small exhaust volumes. This type of system may be used swing grinders, portable grinding wheels, soldering operations, sto cutting, rock drilling, etc.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain a even distribution of air flow through the opening and about the obje being sprayed. While in many instances spraying operations can l performed mechanically in wholly enclosed booths, the volatile vapo may reach injurious or explosive concentrations. At all times the co centrations of these vapors, and particularly those containing benz should be kept below 100 parts per million. Spray booth vapors a dangerous to the health of the worker and care should be taken to mir mize exposure to them.

It is recommended in the design of spray booths that the exhaust du be located in a horizontal position slightly below the object spraye Stagnant regions within the booth should be carefully avoided or shou be provided with exhaust. The air volume should be sufficient to mai tain a velocity of 150 to 200 fpm over the open area of the booth, and tl vapors may be discharged through a suitable stack to permit dilution, b it is better practice to pass the fumes or vapors through baffle ty washers or scrubbers designed for efficient spray fume removal[7].

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided wi sliding windows which permit positive control of the fumes and vapo evolved by the apparatus. Their design should offer easy access for tl installation of chemical equipment and should be well lighted. A velocities should exceed 50 fpm when the window is opened to its max mum height.

DUCT SYSTEM DESIGN

The duct system should be large enough to transport the fumes material without causing serious obstruction to the air flow. It is goc practice to proportion the ducts to obtain the desired velocities ar suction pressures at the hoods, although in many cases only an approx mation to an ideal design is possible. Many exhaust hoods, and pa

[7]For a discussion of spray booths, see Special Bulletin No. 16, Spray Painting in Pennsylvania, Depa ment of Labor and Industry, 1926, Harrisburg, Pa.

drier decreases as a transfer of heat to the material being dried takes place. Where part or all of the heat is supplied by steam coils or other means, within the drier itself, the drier is known as a *constant temperature drier*. Driers using little air for heating medium with a high temperature drop are difficult to hold at uniform temperatures; the more air used, the easier it is to secure accurate control of temperature and humidity. Driers may be classified as shown in Table 1.

MECHANISM OF DRYING

The modern theory of drying may be summed up as follows: Assuming uniform velocity and distribution of air at a constant temperature and humidity over the surface to be dried, the drying cycle will be divided into two distinct stages:

1. Constant rate period.
2. Falling rate period.

The *constant rate period* occurs while the material being dried is still very wet, and continues as long as the water in the material comes to the surface so rapidly that the surface remains thoroughly wet, and evaporation proceeds at a constant rate, precisely as from a free water surface. The material assumes a temperature corresponding to the wet-bulb temperature of the surrounding air, or slightly higher, due to radiation and conduction from dry surfaces adjoining the material. The constant rate period continues until a time when the moisture no longer comes to the surface as fast as it is evaporated. This point is called the *critical moisture content.*

As the drying proceeds, a period of *uniform falling rate* is entered. During this period, the surface of the material is gradually drying out, and the rate of drying falls as the remaining wet surface decreases in area. This period is also known as unsaturated surface drying.

As drying continues, the surface is completely dry and the water from the interior evaporates and comes through the surface as vapor. As the plane of water recedes, the diffusion of the vapor becomes more difficult and hence the period is known as *varying falling rate period*, or sub-surface drying.

As drying progresses another point called *equilibrium moisture content* is reached, where the vapor pressure of the moisture in the air and the vapor pressure of the moisture in the material are equal, and drying ceases. The drying of a slab of whiting is shown in Fig. 2 and illustrates the principles pointed out above. The factors affecting the variations of drying rates during the above periods are pointed out in Table 2.

Omissions in the Cycle

Many solids, such as lumber, are so dry at the beginning of the drying operation that the constant rate period of free surface evaporation does not occur. Frequently the surface of the material is dry enough so that no surface drying can take place, in which case only the final stage of sub-surface drying is involved. In other instances, the critical moisture content of a wet solid is sufficiently low that sub-surface drying starts almost immediately after the conclusion of the constant rate period. Thus the

In discontinuous driers, *e.g.*, compartment driers, the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S' \frac{dw}{d\Theta} \tag{2a}$$

In the continuous drier, the heat consumption per unit time is:

$$\frac{Q}{\Theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)\,(H_2 - H_1) + S(t'_2 - t'_1)\,(s' + w_1) + B \tag{3}$$

Equation 3 assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average

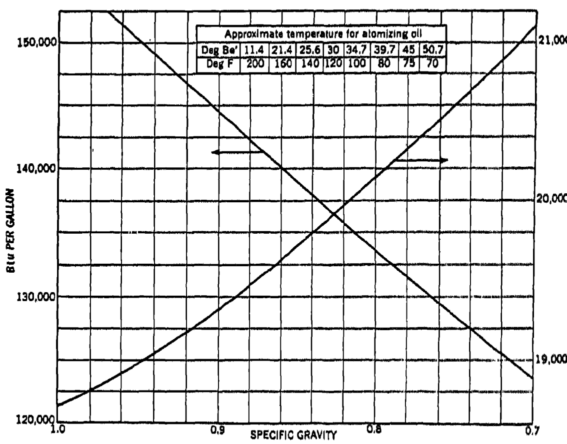


FIG. 7. HEATING VALUES OF FUEL OIL, BTU GROSS

values of t, t' and H may be employed provided the third term of the right hand member of the equation is modified to read:

$$S'\,(t''_1 - t''_1)\,(s' - w_1)$$

and in the second term t'_2 be replaced by

$$\frac{t'_1 + t''_2}{2}$$

Theoretically these periods should be very short and the equation integrated. Practically the error introduced by using a small number of long periods and employing average values of the variables over each, rarely introduces serious error. The evaluation of Equation 2a may be approximated in a similar manner.

The first term of the right hand member of Equation 3 represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air

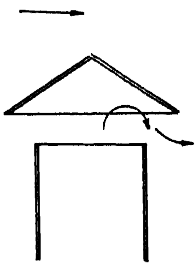


FIG. 2

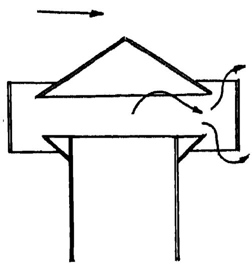


FIG. 3

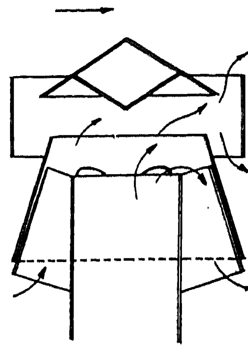


FIG. 4

TYPES OF STATIONARY VENTILATORS

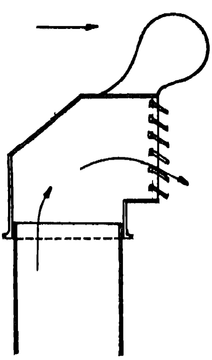


FIG. 5

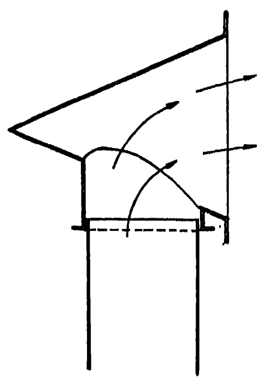


FIG. 6

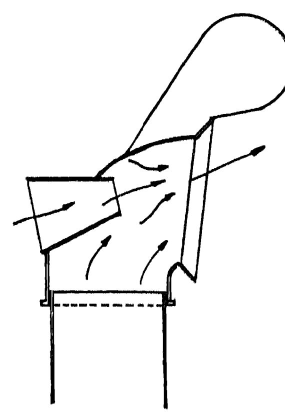


FIG. 7

OSCILLATING VENTILATORS

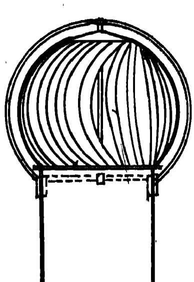
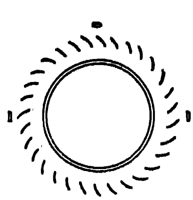


FIG. 8



Section

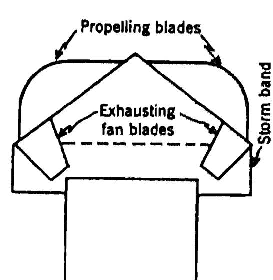


FIG. 9

ROTATING VENTILATORS

Table 3 can also be used for calculating the free convection rate of transmission for various commercial shapes such as pipes and ducts. These calculations are simplified by the use of the factors in Tables 4 and 5. Table 4 gives factors by which the values in Table 3 must be

TABLE 6. HEAT LOSSES FROM HORIZONTAL BARE STEEL PIPES

Expressed in Btu per hour per linear foot per degree Fahrenheit difference in temperature between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	299.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.455	0.495	0.546	0.584	0.612	0.706	0.760
¾	0.555	0.605	0.666	0.715	0.748	0.866	0.933
1	0.684	0.743	0.819	0.877	0.919	1.065	1.147
1¼	0.847	0.919	1.014	1.086	1.138	1.324	1.425
1½	0.958	1.041	1.148	1.230	1.288	1.492	1.633
2	1.180	1.281	1.412	1.512	1.578	1.840	1.987
2½	1.400	1.532	1.683	1.796	1.883	2.190	2.363
3	1.680	1.825	2.010	2.153	2.260	2.630	2.840
3½	1.900	2.064	2.221	2.433	2.552	2.974	3.215
4	2.118	2.302	2.534	2.717	2.850	3.320	3.590
5	2.580	2.804	3.084	3.303	3.470	4.050	4.385
6	3.036	3.294	3.626	3.886	4.074	4.765	5.160
8	3.880	4.215	4.638	4.960	5.210	6.100	6.610
10	4.760	5.180	5.680	6.090	6.410	7.490	8.115
12	5.590	6.070	6.670	7.145	7.500	8.800	9.530

TABLE 7. HEAT LOSS FROM HORIZONTAL BARE BRIGHT COPPER PIPE

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.180	0.210	0.218	0.229	0.299	0.338	0.355
¾	0.236	0.275	0.291	0.307	0.357	0.408	0.418
1	0.290	0.338	0.354	0.373	0.440	0.492	0.523
1¼	0.340	0.400	0.418	0.443	0.510	0.571	0.598
1½	0.390	0.463	0.473	0.507	0.598	0.671	0.710
2	0.490	0.525	0.600	0.628	0.719	0.813	0.851
2½	0.580	0.675	0.709	0.750	0.840	0.953	1.008
3	0.680	0.788	0.848	0.871	0.987	1.107	1.165
3½	0.760	0.888	0.946	1.000	1.114	1.235	1.307
4	0.940	1.000	1.045	1.107	1.210	1.361	1.456
4½	.....	.....	.....	.....	1.335	1.495	1.488
5	1.020	1.200	1.255	1.320	1.465	1.670	1.755
6	1.160	1.375	1.410	1.500	1.685	1.890	1.942
8	1.460	1.725	1.820	1.890	2.100	2.373	2.510

D = diameter of round ducts, feet.
V = velocity of air in the duct, feet per minute, at specified temperature.
d = density of air, pounds per cubic foot, at the specified temperature at which V is measured.
e = naperian base of logarithms = 2.718.

In using formulae 8, 9, and 10, one of the duct air temperatures will be unknown and will be solved for by substitution of the other known or assumed values. The assumed values dependent upon the mean duct air temperature can be determined exactly by the cut-and-try method.

TABLE 15. HEAT TRANSMISSION THROUGH DUCT WALLS INSULATED WITH MATERIALS OF VARYING CONDUCTIVITIES[a]

Values are expressed in Btu per hour per square foot of flat surface per degree Fahrenheit difference in temperature between air inside and still air outside at 90 F for cold air and 50 F for warm air in ducts

Conductivity of Insulation at 86 F Mean Temp	Thickness of Insulation (Inches)	Cold Air			Warm Air			
		40 F	60 F	80 F	90 F	120 F	150 F	180 F
		Temperature Difference						
		50 F	30 F	10 F	40 F	70 F	100 F	130 F
0.200	½	0.319	0.323	0.328	0.324	0.330	0.337	0.344
	1	0.175	0.177	0.180	0.178	0.181	0.184	0.188
	1½	0.121	0.122	0.124	.....	0.125	0.127	0.129
	2	0.092	0.093	0.095	.....	.....	.....	.....
0.250	½	0.382	0.387	0.392	0.390	0.397	0.404	0.412
	1	0.214	0.217	0.220	0.218	0.221	0.225	0.229
	1½	0.149	0.151	0.153	.....	0.154	0.156	0.159
	2	0.114	0.115	0.117	.....	.....	.....	.....
0.300	½	0.440	0.445	0.450	0.448	0.457	0.466	0.475
	1	0.252	0.255	0.258	0.256	0.260	0.264	0.268
	1½	0.176	0.178	0.180	.....	0.181	0.184	0.187
	2	0.135	0.137	0.139	.....	.....	.....	.....
0.350	½	0.494	0.499	0.505	0.502	0.511	0.521	0.530
	1	0.286	0.289	0.292	0.290	0.295	0.300	0.306
	1½	0.202	0.204	0.207	.....	0.208	0.211	0.215
	2	0.156	0.158	0.160	.....	.....	.....	.....
0.450	½	.....	0.596	0.602	0.599	0.610	0.621	0.633
	1	.....	0.356	0.360	0.358	0.364	0.370	0.376
	1½	.....	0.254	0.257	.....	0.259	0.263	0.267
	2	.....	0.198	0.200	.....	.....	.....	.....
0.550	½	.....	0.682	0.688	0.685	0.699	0.714	0.730
	1	.....	0.417	0.422	0.418	0.425	0.432	0.440
	1½	.....	0.302	0.305	.....	0.307	0.312	0.317
	2	.....	0.236	0.239	.....	.....	.....	.....

[a]For round ducts less than 30 in. diameter, increase heat transmission values by the following percentages:

Thickness of Insulation (Inches)	½	1	1½	2
21 to 30 in. Duct Diameter	1%	2%	3%	4%
12 to 21 in. Duct Diameter	3%	5%	7%	9%

Electric Resistor: A material used to produce heat by passing an electric current through it.

Electric Heating Element: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

RESISTORS AND HEATING ELEMENTS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by an alternating current of electricity passing through it. One of the more common resistors is nickel-chromium wire or ribbon which, in order to avoid oxidation, contains practically no iron.

Commercial electric heating elements are made in many types. Some have resistors exposed to the air being heated. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. This type of element is used extensively for operation at high temperatures when radiant heat is desired, also at low temperatures for convection and fan circulation heating, especially in large installations.

Some elements have metallic resistors embedded in a refractory insulating material, encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Elements are made in many forms, such as strips, rings, plates and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in some types of convection air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. Cloth fabrics woven from flexible resistor wires and asbestos thread, are used for many low temperature purposes such as heating pads and aviators' clothing.

Special incandescent lamps are used as heating elements in certain applications where radiant heat is desired. These use carbon or tungsten filaments as resistors, and are designed to produce maximum energy in the infra-red portion of the spectrum.

ELECTRIC HEATERS

Electric heaters may be divided into three groups: conduction, radiant and convection.

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such applications as aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and water heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant

body, heat is also lost through evaporation from both the body surface and the respiratory tract.

The rate of heat loss by convection depends upon the average temperature difference between the surface of the body and the surrounding air, the shape and size of the body, and the rate of air motion over the body.

The rate of heat loss by radiation depends upon the exposed surface area of the body, and upon the difference between the mean surface temperature of the body and the mean surface temperature of the surrounding walls or other objects. This latter temperature is called the mean radiant temperature (MRT).

Because these two types of heat loss supplement each other, a required rate of total heat loss can result either from a relatively low air temperature and a relatively high MRT, or vice versa. If the air temperature is reduced, the heat loss from the body by convection is increased, which can be compensated for by raising the MRT so as to decrease the heat-loss by radiation.

A heating installation should provide comfort for those individuals doing the least physical work, without causing undesirable changes either in the rate of heat generation, or in the body's heat regulating mechanism.

Rate of Heat Production

The normal rate of heat production in an average sized sedentary individual is about 400 Btu[1] per hour. When considering radiant heating, the evaporation, radiation and convection losses must be separately studied. The human body is of complicated shape, and radiation takes place freely, only from the exposed outer surface; there are considerable portions of the body such as the legs, arms, lower part of head, etc., which radiate most of their heat to other portions. It is necessary to determine the equivalent surface of the body from which heat is radiated and a similar value for convection. The total surface may be assumed as approximately 19.5 sq ft for convection and 15.5 sq ft for radiation, for an average sized individual.

The loss by evaporation and respiration depends on the temperature and area of the moist surfaces (outside and respiratory) of the body, the air temperature, air movement and humidity. In air at a temperature of 80 F, this loss for a sedentary individual of average size will be approximately 180 Btu per hour; at 70 F, about 90 Btu per hour; and at 60 F, about 60 Btu per hour. All of these values are relative, because the total will vary materially with change of position, bodily activity, age, sex, race, etc.

The balance of the heat generated in the average human body (approximately 300 Btu per hour at about 71 F room temperature) is the approximate amount of heat given off by radiation and convection. It is difficult to determine the exact proportions of these two; but it appears that if the body loses about 190 Btu per hour by radiation (or 12.25 Btu per hour per square foot of radiating body surface), the greatest comfort

[1]A.S.H.V.E. RESEARCH REPORT No. 830—Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1939, p. 245).

sectionalizing is for the purpose of avoiding excessive pressures on the fixtures in the lower stories of each system. This limits the consideration of water pipe sizes to horizontal mains and to risers not exceeding 20 stories in height or about 200 ft.

For the purpose of this chapter the following terms will be used and should be clearly distinguished from one another:

Maximum Possible Flow: The flow which would occur if the outlets on all fixtures were opened simultaneously. This condition is seldom, if ever, obtained in actual practice except in cases of gang showers controlled from one common valve, and similar conditions.

Maximum Probable Flow: The maximum flow which any pipe is likely to carry under the peak conditions. This is the most important amount to be considered in pipe sizing.

Average Probable Flow: The flow likely to be required through the line under normal conditions.

It is evident that any pipe adequate to take care of the *maximum*

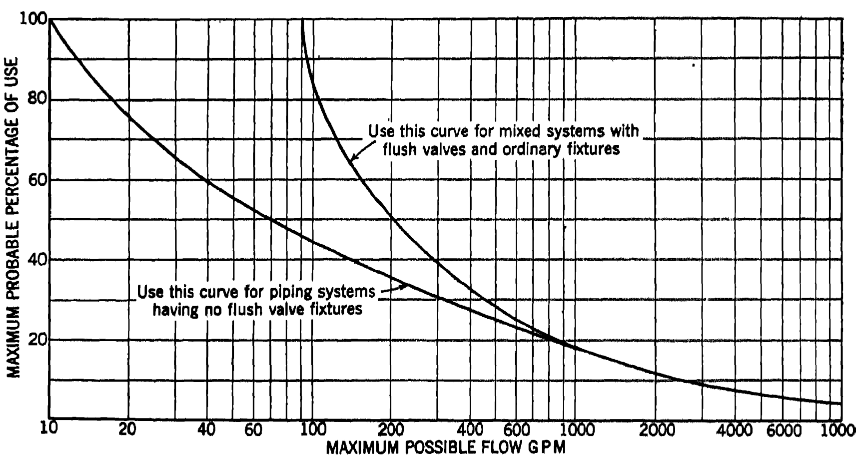


FIG. 1. CHART SHOWING RELATION BETWEEN MAXIMUM POSSIBLE FLOW AND MAXIMUM PROBABLE PERCENTAGE OF USE

probable flow will also be more than able to take care of the *average probable flow*, and hence the latter has no bearing on the pipe size.

MAXIMUM PROBABLE FLOW

There are two factors to be considered in calculating the maximum probable flow, namely, (1) the quantity of water that will flow from the outlets when they are open, and (2) the number of outlets likely to be open *at the same time*. Table 1 shows the maximum approximate rate of flow from each fixture when it is in use, and will serve as a guide in estimating maximum probable flow demands although there is considerable variation in different fixtures and valves. Probably the flow under normal water pressures, or with the pressure properly throttled, will not differ greatly from the values stated. With the aid of this table it is possible to calculate the maximum possible flow with all outlets open in both the hot and cold water lines.

TABLE 6. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(*Riser No. 2. Fig. 4*)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
1st	1 W. C.	45	45	45	100	45	30	1¼
2nd	2 W. C. 1 U. 1 Lav.	45 30 3	90 30 3 123	168	58	98	30	1½
3rd	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	417	31	130	30	2
4th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	666	24	160	30	2
5th	6 W. C. 4 Lav.	45 3	270 12 282	948	19	180	30	2
6th	6 W. C. 4 Lav.	45 3	270 12 282	1230	16	196	30	2½
7th	6 W. C. 4 Lav.	45 3	270 12 282	1512	14	211	30	2½
8th	6 W. C. 4 Lav.	45 3	270 12 282	1794	12	215	2	4

Then the allowable drop per 100 ft will be $\frac{25.6 \text{ lb} \times 100}{300} = 8.5$ lb and the sizes shown in Fig. 5 are based on this amount of drop. Of course the other risers will have the same maximum flows at the bottom as they formerly had at the top, namely 215 and 122 gal, respectively, for Risers Nos. 2 and 3. Combining these maximum flows in the same manner as pursued in the down-feed system it is seen that the maximum flow between Riser No. 2 and Riser No. 3 is 255 gpm, and between Riser No. 3 and the street main, 282 gpm which at a drop of 8.5 lb gives the main sizes indicated. It will be noted that in determining the maximum flow in an up-feed riser it is necessary to begin at the top floor and work down instead of beginning at the bottom floor and working up as was done in the down-feed sizing.

SIZING UP-FEED AND DOWN-FEED HOT WATER SYSTEMS

Hot water supply systems, when of the circulating type, have a few differences to be considered although the same general principles of sizing apply to these lines as to the cold water lines. Owing to the fact that there are no flush valves on the hot water piping and also because many plumbing fixtures have no hot water connections, the sizes of the hot water piping in general will be considerably less than the cold water piping in the same building. On the other hand it is almost invariably

the temperature rise most commonly assumed and required. On this basis it will be seen that the various conditions cited in Example 5 will require additional boiler capacity as follows:

Heating Capacity (Gph)	Additional Boiler Capacity (Sq Ft EDR)
833	3332
1200	4800
1500	6000

From this it is apparent that it is less costly to provide ample storage and to reduce boiler capacity than to diminish the storage and supply a greatly increased boiler capacity to compensate.

The boiler allowance value of 4 sq ft of equivalent steam radiation for each gallon of water heated through a temperature range of 100 F is based on an hourly heating rate. When reduced heating capacities are desired for economic reasons of boiler design and selection, engineers frequently recommend that the heating rate be extended over a period of two hours in which case the boiler allowance value would be reduced to 2 sq ft of equivalent steam radiation. Similarly, any other heating rate may be established and a corresponding value of boiler allowance determined.

Reliable information based upon the installations of several heaters in existing heating systems indicates varying arbitrary values of boiler allowances to be used. When these values are selected for usage, a careful analysis of the varying factors involved in determining these values should be considered so that the proper heating allowances may be provided.

ESTIMATING HOT WATER DEMAND BY FIXTURES

In buildings where the occupancy is doubtful and only the number of plumbing fixtures can serve as a basis for determining the probable hot water demand, the problem is not so simple owing to the fact that a fixture gives no information as to how heavy a service may be demanded from the fixture and this amount of service is really the governing factor in making an estimate of the probable hot water demand. Table 13 may prove of some value in this respect as it gives the maximum assumed quantity of hot water per hour which will be demanded of any fixture and then gives a percentage of this amount which may be assumed as probable in different types of buildings. Table 14 gives approximate hot water requirements in various types of buildings.

Example 6. Let it be assumed that an apartment house with 20 apartments has 20 baths, 20 lavatories, 20 kitchen sinks and 20 laundry trays; what is the probable maximum hourly demand for hot water?

20 Baths at 40 gal and 33 per cent	270 gal
20 Lavs. at 20 gal and 25 per cent	100 gal
20 Sinks at 30 gal and 33 per cent	200 gal
20 Trays at 50 gal and 60 per cent	600 gal
Total	1170 gal
Probable peak use at one time	35 per cent
Probable actual peak demand	409 gph

If three persons are assumed to an apartment the total daily use of hot water should approximate 20 × 3 × 40 gal = 2400 gal and if the peak hour is 10 per cent of this amount, the peak hour by this method shows a probable demand of one-tenth of 2400 gal, which indicates that the values in Table 13 are safe.

Micron .... μ (mu)
Miles per hour .... mph
Millimeter .... mm
Minute .... min
Molecular weight .... mol. wt
Mol .... mol
Ounce .... oz
Pound .... lb
Power, Horsepower. Work per unit time .... P
Pressure, Absolute pressure, Gage pressure, Force per unit area .... p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred .... Q
Quality of steam, Pounds of dry steam per pound of mixture x
Revolutions per minute .... rpm
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor .... *Subscript* f
Second .... sec
Specific gravity .... sp gr
Specific heat .... sp ht or c
Specific heat at constant pressure .... c_p
Specific heat at constant volume .... c_v
Specific volume, Volume per unit weight, Volume per unit mass v
Square foot .... sq ft
Square inch .... sq in.
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion) .... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature). .... T or Θ (*capital theta*)
Thermal conductance[3] (heat transferred per unit time per degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance (heat transferred per unit time per unit area per degree) .... C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

Thermal conductivity (heat transferred per unit time per unit area, and per degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer (heat transferred per unit time per unit area per degree) .... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area (heat transferred per unit time per unit area per degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

[3]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples: conductivity, resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples: conductance, transmittance. Terms ending *ion* designate rate of heat transfer. Examples: conduction, transmission.

In this Catalog Data Section of The Guide 188 manufacturers present detailed descriptions of modern heating, ventilating and air conditioning equipment—260 pages of valuable data, profusely illustrated.

Alphabetical arrangement of advertisers—on pages 811-816—permits ready reference to the products of a specific manufacturer.

For convenience in locating manufacturers' data on various types of apparatus and materials, the equipment shown in the Catalog Data Section has been grouped as follows:

A classified Index to Modern Equipment—on pages 1097-1120—contains complete listings of manufacturers whose products are described in the Catalog Data Section.

Parks-Cramer Company

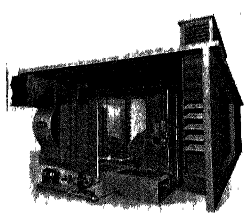
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

Complete Air Conditioning Systems including Heating, Cooling, Humidifying or De-humidifying, Air Changing, Refrigeration, Air Filtering, Air Washing

AUTOMATIC REGULATION

Merrill Process System of Hot Oil Circulation for Heating Industrial Materials

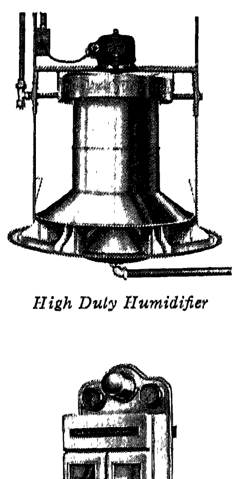


Central Station

Central Station Air Conditioning

Centrally located AIR WASHER. Proper moisture. Positive, pre-determined air removal or re-circulation. Heating coils and refrigeration optional. Helps such industries as Celluloid; Cement; Ceramics; Cereals; Cigars, Cigarettes and Tobacco; Clothing; Confectionery; Glassine; Leather; Paper and Envelopes; Printing and Lithographing; Shoes; Starch and Dextrine; Storage of Perishables; Textiles; Wood Products. Similar installations effective in Hospitals, Art Galleries, Auditoriums, Restaurants.

Air Washer or Central Station Units.
Nozzles for Central Station Air Washers.



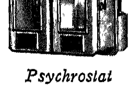
High Duty Humidifier

High Duty Humidifier

Water under pressure generates spray. Excess water returns to filter tank and re-circulates. Evaporation per unit high; two sizes of heads each with three sizes of nozzles give flexible capacity for varying conditions. Circulation increased by individual motor-driven fan. Spray thoroughly diffused and distributed over wide area.

Turbomatic Humidifier

(not illustrated)
Efficient humidifier of the atomizer type. For direct humidification, as humidity boosters for Central Station systems of all makes. Self-cleaning.



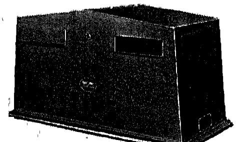
Psychrostat

Parks Automatic Airchanger

For use with High Duty or Turbomatic Humidifiers. Insures fixed humidity and maximum evaporative cooling.

Automatic Regulation

The Psychrostat for accuracy, durability, sensitivity. Employs the principle of the Sling Psychrometer, used in all U. S. Weather Bureau Stations. Hygrostat (not illustrated) where requirements are not so exacting. An Air Conditioning System is no better than its Regulation.



Pettifogger

The Pettifogger

A compact humidifier for offices, stores, storerooms, laboratories, or other isolated departments. Self-contained in lacquered copper casing. Permanently though flexibly connected to water and electrical supplies. Automatic control. Adjustable capacity. Reduces dust. Neutralizes drying effect of heating.

Delco Appliance Division

General Motors Sales Corporation

BACKED BY
BUILT AND GM GENERAL MOTORS

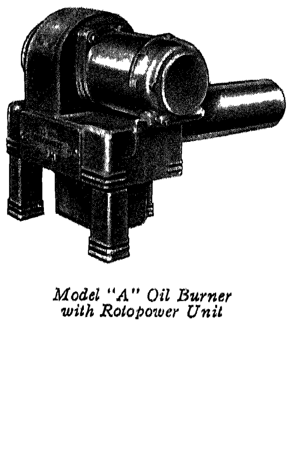
Rochester, N. Y.

Delco Offers a Complete Line of Automatic Heating Products, including: Oil burners, bituminous coal stokers, oil-fired boilers, oil and gas-fired winter air conditioners, automatic water heaters, thermostats and master controls.

Write to Delco Appliance Division, General Motors Sales Corporation, Rochester, N. Y., for latest information and detailed specifications, or consult your local Delco distributor whose address is listed in the classified section of your telephone directory.

DELCO AUTOMATIC HEAT

OIL BURNERS



Model "A" Oil Burner with Rotopower Unit

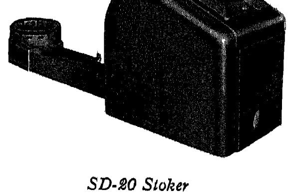
Delco Oil Burners employ the highly efficient pressure atomizing method of breaking the liquid fuel into fine particles for complete combustion. In the Delco Oil Burner with the Rotopower Unit, the motor, air blower, fuel pump, filter and fuel control valve are all contained within a single, easy-to-remove unit on an integrated shaft. The high-precision pump connecting directly to the shaft of the Rotopower Unit motor has only two moving parts, fitted together within 2/10,000ths of an inch.

Built into the Rotopower Unit is the Thin Mix Fuel Control which regulates the pressure of the fuel oil for proper mixing with air to assure the most economical flame. Delco Oil Burners are available in 5 sizes in standard voltage characteristics with combustion rates from 1 to 16.5 gal per hour or a capacity range from 440 to 6,550 sq ft of steam, EDR.

BITUMINOUS COAL STOKERS

Delco Stokers which are designed to provide automatic firing for coal-fired domestic heating plants are of the underfeed, screw type with intermittent coal feed. Two 20-pound, one 30-pound and one 50-pound stoker, burning bituminous coal, make up the Delco line. Features incorporated in deluxe models include: automatic controls, air control, Rhino-Hide lined hopper, smokeback eliminator, oversize feed worm, and sound insulation.

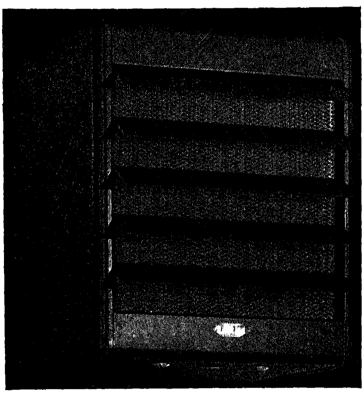
Fingertip transmission control permits ease of regulation of coal flow to suit the weather. Coal control consists of two speeds and neutral in 20-pound models; three speeds and neutral in 30-pound and 50-pound models. Using 12,000 Btu per pound coal, capacity of 20-pound models is 600 sq ft steam EDR, 30-pound models, 900 sq ft steam EDR, and 50-pound models, 1500 sq ft steam EDR.



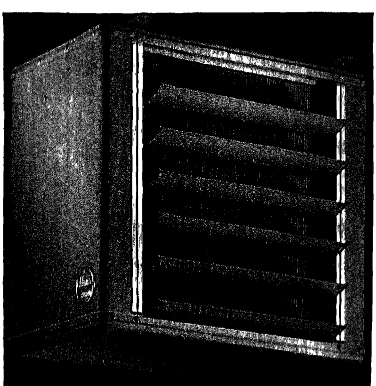
SD-20 Stoker

FEDDERS

MANUFACTURING COMPANY, INC.
85 TONAWANDA ST.
BUFFALO, NEW YORK



Unit Heaters



Unit Coolers

HEAT TRANSFER SPECIALISTS SINCE 1896

FEDDERS UNIT HEATERS

Streamline tubes, individual non-clogging fins, patented full-floating mounting eliminates expansion stresses between heating element and mounting, relief of differential expansion among tubes, rugged monopiece cabinets, quiet operation. Standard horizontal delivery unit heaters built in 25 basic sizes, all available with single, two or multi-speed and standard speed motors. Also available with slow speed motors and motors for odd frequencies.

FEDDERS VERTICAL HEATERS

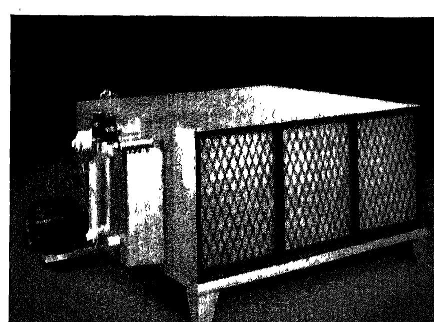
In handsome circular design . . . require minimum headroom . . . provide broad distribution of heated air . . . quiet operation . . . built in a wide range of sizes.

UNIT COOLERS

Built in a complete range of single and twin fan sizes for use with refrigerant or cold water. Finished in attractive polar green.

AIR CONDITIONING UNITS

Available in any combination for heating, cooling, humidifying and dehumidifying and air filtering. Seven basic models from 1 to 25 tons cooling capacities designed for floor or ceiling installations.



The Unit Heater and Cooler Division

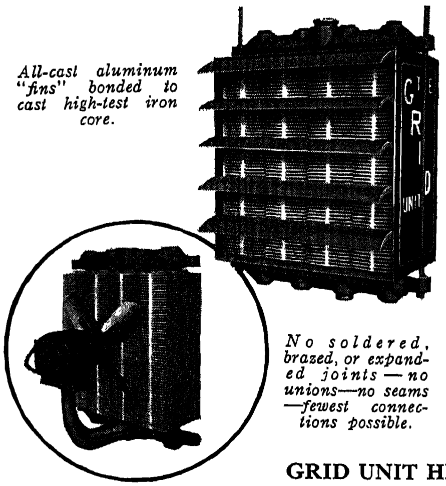
D. J. Murray Mfg. Co.

Wausau, Wisconsin

Offices in Principal Cities

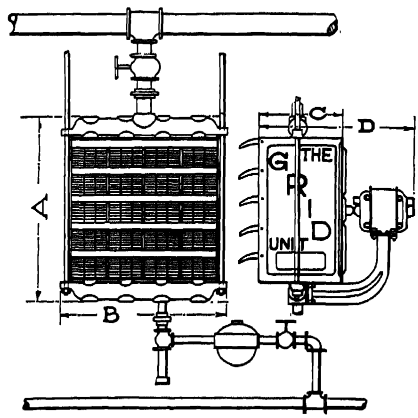
MANUFACTURERS OF THE GRID UNIT

(PATENTED)



All-cast aluminum "fins" bonded to cast high-test iron core.

No soldered, brazed, or expanded joints — no unions—no seams —fewest connections possible.



Typical Installation of Grid Unit Heater

GRID UNIT HEATER DATA

Model No.	Dimensions Inches				Face Area Sq Ft	Motor		Vol. at Fan	Capacities 5 Lb Steam 60° Air		Approx. Shipping Weight	Pipe Sizes	
	A	B	C	D*		Hp	Rpm		Btu	Final Temp.		Supply	Return
1000	16¼	11¾	9⅜	16	0.84	1/20	1700	578	29,400	107	100	1¼"	1¼"
1200	18½	13½	11½	17½	1.04	1/20	1700	711	46,000	119	120	1¼"	1¼"
515	23⅜	17½	11½	20	1.65	1/10	1750	1290	59,600	102	160	1½"	1¼"
1500	23⅜	17½	11½	20	1 67	1/10	1750	1450	77,500	109	210	1½"	1¼"
1520	27⅜	17½	11½	20	2.2	1/10	1750	1700	104,000	113	250	1½"	1¼"
520	28⅜	22⅛	11⅞	21½	2.9	1/6	1150	2500	102,300	97	250	2"	1¼"
2000	28⅜	22⅛	11⅞	21½	2.9	1/6	1150	2500	148,000	114	320	2"	1¼"
2025	33½	22⅛	11⅞	21½	3.6	1/6	1150	2875	177,000	115	370	2"	1¼"
525	35¼	27½	13	28	4.5	1/2	1150	4200	166,400	94	390	2"	1¼"
2504	35¼	27½	13	28	4.5	1/4	1150	3200	210,000	118	420	2"	1¼"
2500	35¼	27½	13	28	4.5	1/2	1150	4200	225,000	108	440	2"	1¼"
2530	37¼	27½	13	28	5.3	1/2	1150	4650	282,000	115	530	2"	1¼"
530	39⅛	32⅝	13	29	6.5	1/2	1150	5300	260,500	105	600	2½"	1¼"
3000	39⅛	32⅝	13	29	6.5	1/2	850	6350	341,000	109	690	2½"	1¼"
3000	39⅛	32⅝	13	29	6.5	1½	1150	8100	394,000	104	725	2½"	1¼"

*Varies with type of motor.

GRID UNIT HEATERS ARE NOT AFFECTED BY ELECTROLYTIC ACTION

No leaks—no breakdowns.
Low maintenance expense.
More air changes per hour.
Positive "directed" heat.

Lower outlet temperatures.
Larger air volume.
Reduced fuel cost.
Applicable to either low or high steam pressure lines.

Send for Bulletin on Units not listed above.

The Air-Maze Corporation

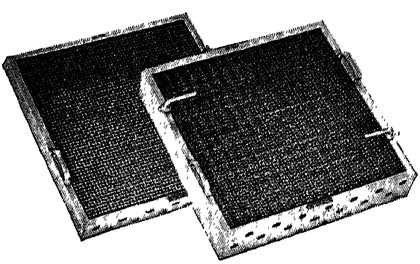
5202 Harvard Avenue, **Cleveland, Ohio**

ENGINEERS AND MANUFACTURERS OF AIR FILTERS EXCLUSIVELY

DIRECT FACTORY REPRESENTATIVES IN ALL INDUSTRIAL AREAS.

DISTRIBUTORS IN PRINCIPAL CITIES AND TOWNS THROUGHOUT THE UNITED STATES.

During more than a decade devoted exclusively to air filter engineering and manufacturing, a great deal about the control and elimination of dust, pollens and grit has been learned by AIR-MAZE engineers. Their design and development of a unique type of filter element construction, embodying distinctive advantages, has been considered a worthy contribution to the air filtering science and has resulted in wide acceptance of AIR-MAZE air filters in all fields of application.



2 in. Thick Panel *4 in. Thick Panel*

AIR-MAZE Permanent Cleanable Panel Filters

Note Advantages Made Possible by Air-Maze Scientific Construction:

Costs Little to Clean—The separating layers and exact spacing of baffles permit free washing action between and around all baffles. Thus, cleaning and charging operations may be easily and economically performed.

Great Dust Capacity—Unique design of the AIR-MAZE screen wire element provides a vast area of baffles on which collected material can become impinged; thus great capacity is assured.

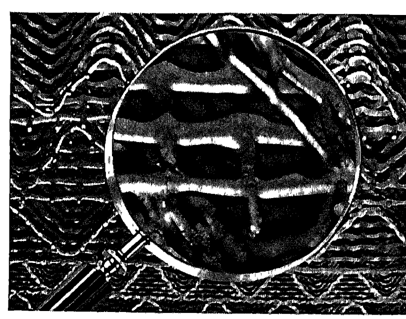
Vibration Proof—Vibrations in service cannot shake filter media out of position—the uniform density remains *permanently* perfect; **no replacements are necessary!**

AIR-MAZE Are Listed by Underwriters' Laboratories—When serviced according to the methods recommended by the Underwriters' Laboratories, AIR-MAZE panel filters are approved as fire retardant air filters.

Efficiency—Tests under varying conditions, both in laboratories and field operations, show air filtering efficiency of from 98.00 to 99.83 per cent with practical dust.

No Clogging—Because AIR-MAZE panel filters are easy to *completely* clean and since the exact density enables uniform deposit of dust, no clogging can occur.

Adaptibility—In addition to air conditioning and power equipment installations AIR-MAZE panel filters are effectively used in humidifiers, water eliminator units, paint spray-booths, oil separators, range canopies in kitchens, and other applications where specific problems and unusual requirements are easily handled by adaptations of the panels. AIR-MAZE panels will be made to fit frames of existing installations and can be furnished with locking handles and latches, snap catches, or with flanged edges and lift handles.



Magnified Section of "Loaded" AIR-MAZE Air Filter Element. Note that dust has been quite evenly impinged on the wires. No obstructed spaces can be seen. This feature accounts for the Low Pressure Drop and Non-clogging characteristics of AIR-MAZE

TECHNICAL INFORMATION

Sizes—All sizes and thicknesses are available; two and four inch thick panels are the accepted standard. Installations using large sizes of these permanent panels are surprisingly low in cost.

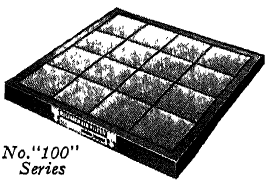
Capacity—Recommended air capacity is 1½ to 2½ cfm per square inch. Thus, the capacity of a 20 x 20 in. panel is 600 to 1000 cfm. Normally, 2 cfm per square inch should be used.

Research Products Corporation

Madison, Wisconsin

RESEARCH AIR FILTERS FOR HEATING AND AIR CONDITIONING

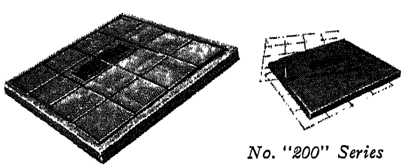
Research Filter With Cardboard Frame



No. "100" Series

Disposable filter, which when dirty, is replaced by entire new unit. For warm air furnaces, air conditioning units, filter banks.

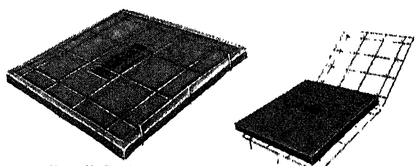
Research Self-Seal Re-Fil-Able Filter With Hooked Wire Grids



No. "200" Series

Filter pad, sandwiched between wire grids which hook together; pad is easily replaced. Used in furnaces, air conditioning units and filter banks.

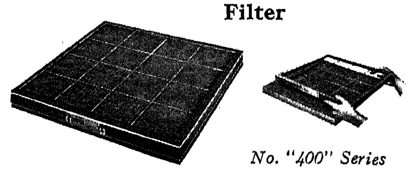
Research Self-Seal Re-Fil-Able Filter With Hinged Wire Grids



No. "300" Series

Easy-to-change Re-Fil-Able filter. Replaceable pad held snugly between hinged wire grids. Used extensively for filter banks.

Research Steel Frame Re-Fil-Able Filter



No. "400" Series

Re-Fil-Able filter with permanent steel frame, complete with wire grids to hold the removable pad in rigid position. New pad easily inserted.

TECHNICAL DATA ON RESEARCH AIR FILTERS

Capacity Rating

Research Air Filters are rated at 2 cfm of air per sq in. of gross nominal area. Recommended maximum air velocity is 400 fpm.

Resistance, Inches of Water	Air Velocity, FPM
.018	100
.065	200
.130	300
.200	400

When Filters are used in tandem the restriction is additive.

A Research Air Filter 20 x 20 x 2 in., when tested according to the test code of the American Society of Heating and Ventilating Engineers has an efficiency of 93 per cent, tested with standard code dust. The dust holding capacity with standard code dust is 150 grams per sq ft of filter area, the restriction at this dust load being .2 in. of water.

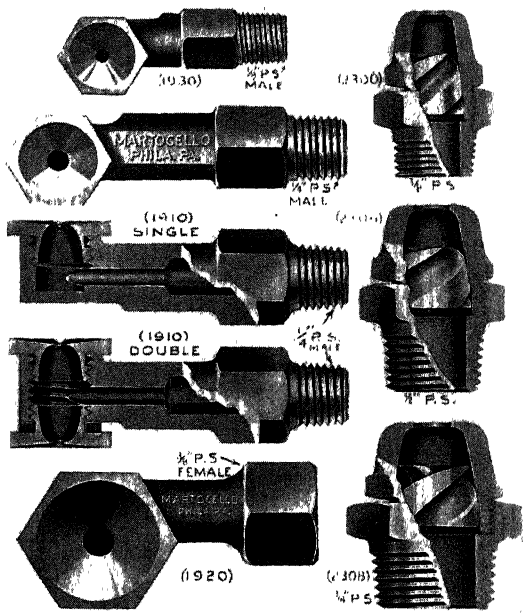
RESEARCH "100" AND "400" SERIES 2-INCH AIR FILTERS
Dimensions, Ratings and Manufacturing Tolerances

Nominal Sizes	Ratings	Actual Dimensions			
These dimensions are used by the trade to order filters and refer primarily to the size of holder into which the filter fits.	Volume of Air Cleaned at Velocity 300 F.P.M. C.F.M.	A Width	B Height	C Thickness	D Border
		Tolerances			
		Plus 0.00 In. Minus 1/8 In.		Plus 1/16 In. Minus 1/16 In.	
20 x 25 x 2	1000	19 5/8	24 9/16	1 13/16	3/4
20 x 20 x 2	800	19 5/8	19 5/8	1 13/16	3/4
16 x 25 x 2	800	15 13/16	24 13/16	1 13/16	3/4
16 x 20 x 2	640	15 13/16	19 5/8	1 13/16	3/4
15 x 20 x 2	600	14 15/16	19 15/16	1 13/16	3/4

Jos. A. Martocello & Company

229-31 North 13th Street, **Philadelphia, Pa.**

ATOMIZING SPRAY NOZZLES



Types of Martocello Spray Nozzles

Martocello Spray Nozzles are broadly used for all types of installations. Manufactured with precision and of a design which has been thoroughly tested for results and durability, they are guaranteed to give you satisfaction.

Successful-Efficient results depend largely upon selecting the proper number and type of Nozzle suitable for your installation. Consult with us.

Martocello Spray Nozzles produce a uniform fine wide spray with minimum friction and at lowest pressure requirements.

We appreciate your inquiries and offer our cooperation in assisting you to select a proper Nozzle for best results.

For maximum efficiency we recommend Nozzle orifices as indicated in table below. Any reasonable range of capacity for various pressures can be provided.

Sizes and Capacities

Pipe Size Inches	Part No.	Diam. Orifice Inches	Capacity, Gallons per Minute 5 lb	10 lb	15 lb	20 lb	25 lb	30 lb	35 lb	40 lb
1/8	1930	7/64	.22	.29	.34	.39	.44	.49	.54	.59
1/4	1910	13/64	.54	.77	.96	1.13	1.29	1.44	1.58	1.71
1/4	1910 Double	5/32	.86	1.18	1.48	1.76	2.02	2.24	2.44	2.63
3/8	1920	17/64	1.48	1.96	2.38	2.75	3.08	3.36	3.60	3.82
3/8	2300	7/32	1.98	2.63	3.15	3.62	4.05	4.44	4.80	5.13
1/2	2304	5/16	2.66	3.77	4.71	5.52	6.24	6.87	7.47	8.04
3/4	2308	11/32	3.59	4.87	5.92	6.83	7.62	8.33	8.98	9.60

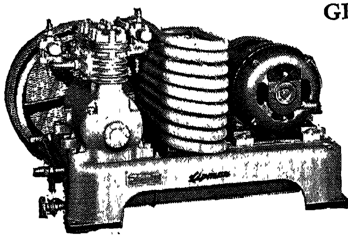
Nozzles illustrated above are made in Brass Forging and machined brass bar stock. Cast Red Brass Nozzles in 1 in. to 2 in. pipe sizes also available. All sizes carried in stock for prompt shipment.

Satisfaction Guaranteed

GENERAL REFRIGERATION DIVISION
YATES-AMERICAN MACHINE COMPANY

Dept. H.V.A.-41 Beloit, Wis., U.S.A.

Commercial Refrigeration
Cooling Units
Air Conditioning Units



Refrigerating units in Air or Water-Cooled types

GR-LIPMAN Equipment for air conditioning and commercial refrigeration includes apparatus for practically every type of service—a great variety of models in a wide range of capacities.

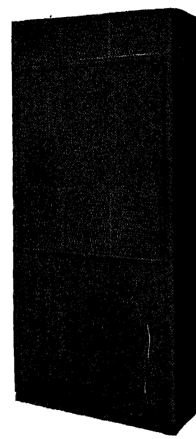
Refrigeration Units are available in air or water-cooled types, for use with accepted and popular refrigerants.

Types and Capacities

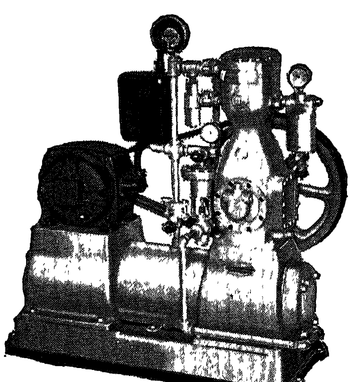
Methyl Chloride	¼ to 5 hp
Freon-12	¼ to 30 hp
Ammonia	1 to 40 hp

Small sizes are self-contained and fully automatic; larger sizes are available with manual control optional.

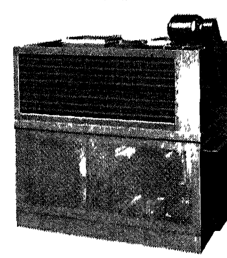
Air Conditioning Units are furnished in both floor and ceiling types for use in stores, restaurants and other commercial establishments. All units are self-contained and are shipped ready to install and operate—controls are adjusted before shipment.



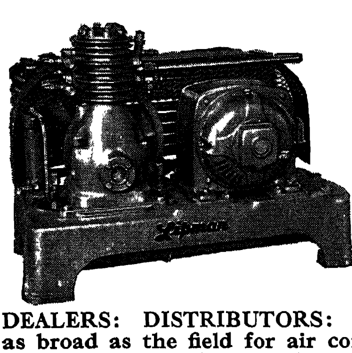
Store Coolers available in 3 popular sizes.

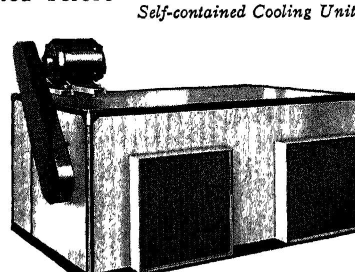


A size and type to meet every need. Ammonia. Methyl Chloride, Freon-12



Self-contained Cooling Unit.





Ceiling Type Cooling Unit.

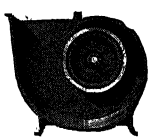
DEALERS: DISTRIBUTORS: The market for GR-LIPMAN Equipment is as broad as the field for air conditioning and commercial refrigeration. A complete line of dependable, economical equipment, backed by a thorough-going engineering service, and aggressive sales and advertising cooperation. Write for details on GR-LIPMAN Equipment.

Bayley Blower Company

1817 S. Sixty-Sixth Street **Branches in Principal Cities** **Milwaukee, Wis.**

Builders of Heating, Ventilating, Cooling, Purifying, Humidifying and Air Washing Equipment; Exhaust and Drying Apparatus, Mechanical Draft and Blast, Fans and Blowers of all Types

Bayley Plexiform Fan:



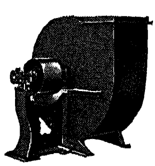
Is a multi-blade fan for supplying air for heating and ventilating systems, manufacturing processes, drying systems, forced and induced draft systems. It is suitable for handling high or low temperature gases at medium or low pressure. Will deliver maximum quantities requiring minimum space with great economy.

This is a distinct Bayley product, high class material and workmanship, properly designed to avoid excessive vibration and overstressing of parts. Inlets and outlets are properly sized for maximum delivery and maximum efficiency. Fans are furnished in single or double width of any required arrangement and with sleeve or anti-friction bearings.

Aeroplex Fan:

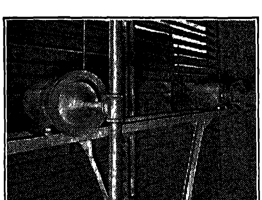
Is of high speed design with self limiting power characteristics. Application parallel to the Plexiform Fan. Highly efficient and quiet in operation.

Bayley Exhausters and Pressure Blowers:



Type "B" exhaust fan is for heavy duty, handling refuse from industrial and textile plants. Type "SE" is used in handling smoke, fumes and dust-laden gases. Type "H" for high-pressure work. These units are highly efficient and of high class design and workmanship.

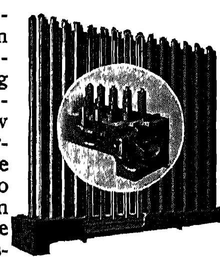
Bayley Turbo Air Washers, Humidifiers and De-Humidifiers:



The Bayley Turbo Air Washer Showing Turbo Atomizer and Eliminator

The Turbo Atomizer used in the Bayley Washer produces a steady, fine spray. Water at low pressure is delivered to the center of a rapidly revolving cone-shaped rotor provided with atomizing pins set in its periphery. This atomizer requires very little attention, and will operate successfully under low water pressure. The orifices are large and this atomizer, unlike high pressure nozzles, cannot clog.

Bayley Chinook Heating Sections:



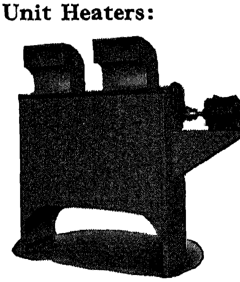
The Chinook section is used with blast heating, ventilating and drying systems, and is suitable for high or low pressure steam circulation. The base is divided into two chambers. Steam enters (see cut) the lower chamber, rising through ⅜-in. pipes located within the 1¼-in. pipes leading from the upper chamber. Condensation takes place in the larger pipes, the water falling into the upper chamber and draining away through the return outlet. The Chinook can be repaired in the middle of the bank without breaking steam connections or taking down a section.

Shipped assembled in smaller sizes, and knocked down in the larger units. May be installed in horizontal or vertical position.

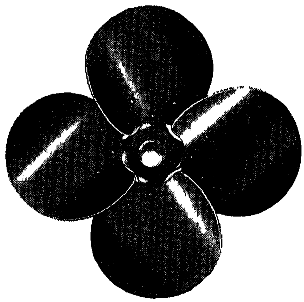
Bayley Chinookfin Heating Sections:

Are the same design as the Chinook Heaters, using heavy gauge copper fin tubes. As compared with Chinook it is much lighter and occupies less space.

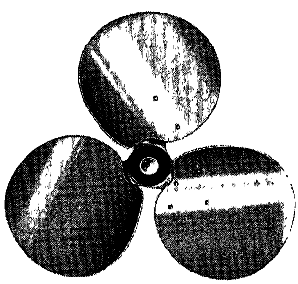
Bayley Plexfin Unit Heaters:



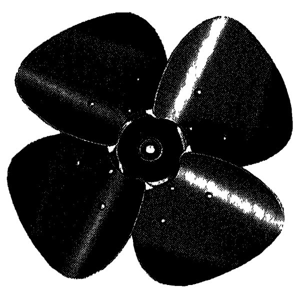
This unit incorporates Chinookfin radiation and Plexiform or Aeroplex fans. The fan assembly including top plate and motor is removable as a unit for maintenance and inspection. The heating element is a removable unit. Casing all welded extra heavy gauge. This is an exceptionally high grade unit at a moderate price.



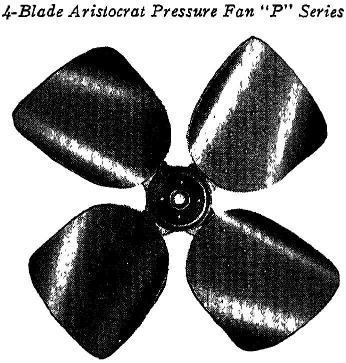
Airistocrat "Standard" Series Pats. 2,072,322 and 2,021,707



3-Blade Airistocrat "Y" Series



4-Blade Aristocrat Pressure Fan "P" Series



4-Blade Aristocrat Pressure Fan "U" Series (also made in two blade model)

AIRISTOCRAT Quiet Propeller Fan Blades are widely recognized for their all-around excellent performance. The unique, patented construction embodies entirely new principles in the art of fan design—produces a blade unsurpassed for quiet operation, rugged construction and attractive appearance. Every **Airistocrat** unit is carefully built and the blades are hand gauged for correct contour and alignment. Statically balanced, these blades deliver full air volume with a minimum of noise. Aluminum alloy blades and steel spiders are standard except where otherwise noted. Rotation is clockwise only (facing air delivery side).

Available in the following finishes: 1. Plain—no finish on blades, spiders or hubs. 2. Blades with no finish; spider and hub with cadmium plate or black lacquer. 3. All black lacquered, with or without center button. 4. Buff and lacquered blades, black lacquered spider and hub, with or without center button. Catalog gives detailed dimensions and guaranteed performance curves recorded under *NEMA* and *NAFM* code tests at various speeds for each of the **Airistocrat** models described below.

"Standard" Series—Has blades mounted on a steel spider. Sturdy, attractive steel or aluminum blades which have withstood extreme laboratory breakdown tests. Sizes 8 in., 10 in., 12 in., 14 in., 16 in., 18 in. and 20 in. diameters in a variety of pitches to meet every need.

Three Blade "Y" Series—The design of this blade is the result of two years of laboratory experiment to produce a better air circulator blade. At recommended speed these blades produce a high velocity air stream effecting deep penetration with unusual quietness. Sizes 10, 12, 14, 16, 18, 20, 24 in. and 30 in. diameters, steel or aluminum blades.

Pressure "P" Series—Similar in construction to "Standard" Series but with blades especially designed for higher pressures. Sizes 10 in., 12 in., 14 in., 16 in. and 18 in. diameters.

Pressure "U" Series—Two and four blade models of steel designed for pressure operation. Sizes 20 in., 22 in., 24 in., 26 in., 28 in. and 30 in. diameters. 24 in. and 30 in. sizes suitable for attic fans. Bulletin gives complete specifications and ratings.

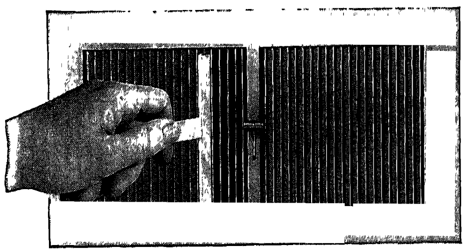
Hart & Cooley Manufacturing Co.

Established 1901

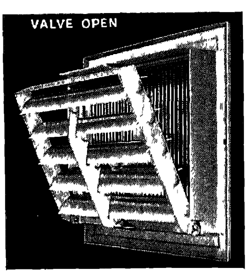
Air Conditioning Registers and Grilles - Warm Air Registers
Damper Regulators - Furnace Regulators - Pulleys - Chain

Holland, Mich.

NO. 75 DESIGN—FLEXIBLE FIN TYPE with TURNING BLADE VALVE to provide DOUBLE DEFLECTION. Also without Valve as Grille or Intake

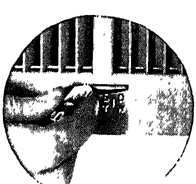


CONTROL OF AIR FLOW IN TWO PLANES



Instant Adjustment of Air Flow (Up, Straight or Down)

Is obtained by turning the regulator on the register face to the proper setting with a key furnished with each register. When the valve is opened, as shown at the left, the individual valve louvres automatically stop in position to provide the proper air flow—Up (Fig. 1) for cooling systems to avoid drafts; Straight (Fig. 2) for ventilating systems; Down (Fig. 3) for heating systems to prevent stratification. When the valve is closed, as shown at the left below, it completely stops the flow of air.



Air Flow Can be Quickly Adjusted Sideways

No. 75 Design has a flexible fin-type face. Each fin may be twisted individually with a tool furnished with each register or grille to provide any desired sideway deflection of the air flow.

Greatly Reduced Turbulence and Resistance

Figs. 1, 2, and 3 show the air flow with No. 75 Design; Fig. 4, with the conventional register. Compare the turbulence in the stackhead of the latter with the smooth flow obtained with No. 75 Design. So efficient is No. 75 Design that there is actually less resistance with this register, using a standard stackhead, than if no register at all were used.

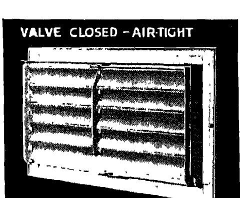


Fig. 1

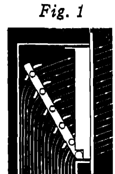


Fig. 2

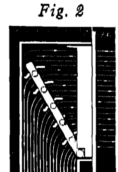


Fig. 3

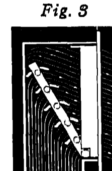


Fig. 4

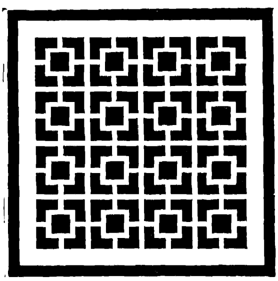
Wickwire Spencer Steel Company

500 Fifth Avenue, **New York City**

BUFFALO CHICAGO WORCESTER DETROIT PHILADELPHIA
CHATTANOOGA LOS ANGELES SEATTLE SAN FRANCISCO PORTLAND

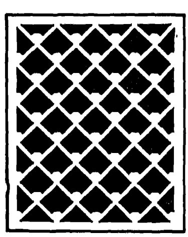
Agents in Principal Cities

WISSCO STAMPED GRILLES and PERFORATED METALS



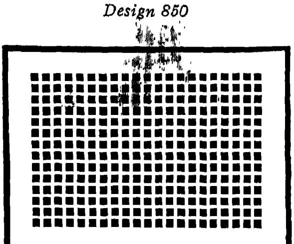
Design 820

The Wickwire Spencer Steel Company was the first perforator in America to produce stamped metal grilles. Over fifty years' experience in this field, modern equipment, complete warehouse stock, and competent workmen having an average service record of over nineteen years, assures excellent service and a superior product.



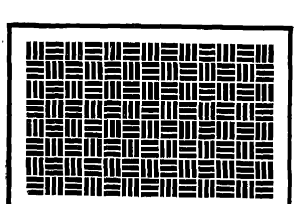
Design 850

Wickwire Spencer's modern machinery produces grilles as heavy as 5⁄16 inches in thickness, and in any size or shape up to 60 inches x 156 inches in one piece. Larger sizes in two or more pieces suitably joined so that the joint is virtually invisible on the front of the grille. All tools, both for standard design grilles and those of special design, are supplied by our own tool room.



Plain Lattice

Wissco Grilles are manufactured in over 100 different styles from standard dies, and in many special designs and curved shapes to customer specifications. Perforations are designed so that uninterrupted vertical and horizontal members give adequate rigidity and strength of grille structure with effective concealment and free air openings as high as 70 per cent of grille area.



Design 204

Wissco Grilles are fabricated from steel, bronze, brass, aluminum, stainless steel or monel. Any standard electroplated finish can be supplied on steel. Polished or oxidized standard finishes can be given bronze, and polished or alumilited finishes can be applied to aluminum. Special finishes to match adjoining metal or to harmonize with the surroundings can also be furnished.

Invisible doors, hinged grilles, angle frames, or other special features will be supplied as required. Other sheet metal designs and specialities produced as desired.

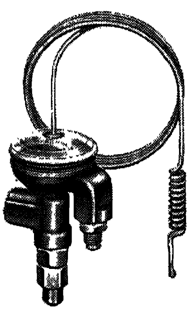
Our new catalog "Wissco Grilles" gives detailed analysis and specifications of grilles layouts. A copy will be mailed to you upon request.

Detroit Lubricator Company

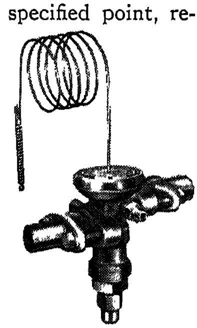
Detroit, Michigan, U. S. A.

NEW YORK, N. Y., 40 West 40th Street
CHICAGO, ILL., 816 S. Michigan Avenue LOS ANGELES, CALIF., 320 Crocker Street
Canadian Representative:
RAILWAY AND ENGINEERING SPECIALTIES LIMITED, Montreal, Toronto, Winnipeg

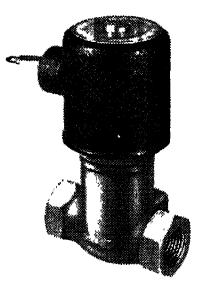
Dura-Fram Expansion Valves



"Detroit" Expansion Valves are used in the liquid lines to evaporator coils of refrigeration and air conditioning installations. They operate at variable back pressures to keep them completely refrigerated regardless of the variations in load. Power elements are charged with gas at a definite pressure instead of a liquid. By this means the valves remain tightly closed whenever the suction pressure rises above a specified point, regardless of the temperature of the thermostatic bulb. This prevents overloading the motor when starting up a warm system or when operating under excessive temperature conditions. Also provides instant action with no temperature lag. Capacities range from 1/3 ton to 30 tons with Freon.



Detroit Solenoid Valves

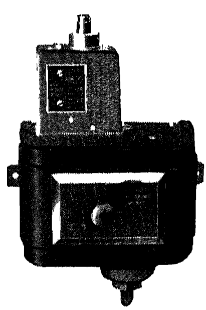


Detroit Solenoid Valves control refrigerants, water or gas. They are available in the following orifice sizes: 1/8 in., 3/16 in., 7/32 in., 3/8 in., 1/2 in., 5/8 in.—and in capacities up to 17 tons Freon at a 2 lb pressure drop.

Other Controls

This Company can also supply a full line of Boiler and Furnace Limit Controls—Room Thermostats for both heating and cooling — Fan controls — Humidity and Stoker Controls and Convector Valves for concealed radiators.

No. 450 Series Refrigeration Controls

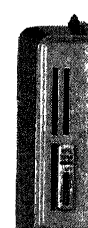


FIBA

Supplied in several models for various refrigeration and air conditioning requirements. Line voltage type. Model FB-3 controls refrigeration compressor from pressure changes in the suction line. Model FIBA has, in addition, a high pressure cutout, which stops the compressor if high side pressures become excessive. Other models control from temperature changes. Range of either −40 deg. to plus 25 deg. or −5 deg. to plus 60 deg. is available.

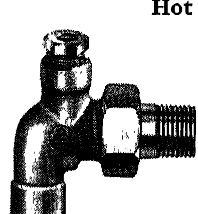
Other features include alarm circuit, external "cold control," and high pressure cut-in for meat box applications, which prevents long "off" cycles and guards against "slimy meat" in cold weather.

The New No. 411 Room Thermostat



The "Genuine Detroit" No. 411 is a very sensitive room thermostat for heating and is supplied with or without compensation (pre-heater). With compensation it provides straight line temperature control, eliminates "cold 70" and prevents costly overheating.

Balancing Fittings for Circulated Hot Water



With "Genuine Detroit" Balancing Elbows and Straight Way Fittings on each radiator return the entire system can be accurately balanced after it is all tightened up and in operation—it is done with the turn of a screw driver and without breaking a single joint.

ASHCROFT GAUGE DIVISION

AMERICAN SCHAEFFER & BUDENBERG INSTRUMENT DIVISION

Manning, Maxwell & Moore, Inc.

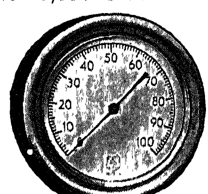
Bridgeport, Conn.—BRANCHES IN PRINCIPAL CITIES

Makers of AMERICAN INDUSTRIAL INSTRUMENTS—Since 1851

Manufacturers of Indicating and Recording Gauges; Gauge Testers; "U" Gauges; Draft Gauges; Indicating and Recording Thermometers; Tachometers; Dial Thermometers; Pressure and Temperature Controllers; Electric Temperature Controllers; Pop Safety and Water Relief Valves; Steam Traps; Absolute Pressure Gauges.

Also manufacturers of Bronze, Cast Steel and Forged Steel Valves, Engine Room Clocks; Barometers; Mercury Column Gauges; Gauge Boards.

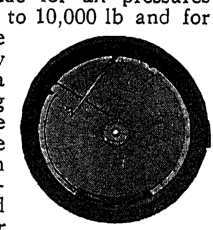
Ashcroft Gauges—Ashcroft Gauges are made in all sizes from 2½ to 12 in., for pressures from 8 oz to 25,000 lb and also for vacuum. Cases are cast-iron or cast brass. The movements are heavy duty and all bearings are Monel Metal. Write for Catalog No. A-59.



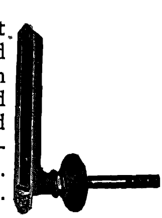
Also Duragauges—accurate to within ½ of 1 per cent. Stainless steel movement. In Phenol Cases in 4½ in., 6 in. and 8½ in. dial sizes.

For Mercury Pressure and Vacuum Gauges, "U" Gauges, Draft Gauges and Mercurial Barometers, write for Catalog B-59.

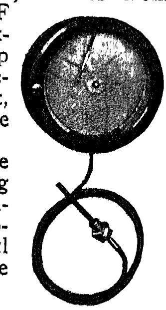
Recording Duragauges—Recording Duragauges are made for all pressures from 15 in. of water to 10,000 lb and for vacuum. They are made in one size only to accommodate a 10 in. chart, having an effective scale width of 3⅝ in. The case is die cast with a dull black hard-rubber finish and with either bottom or back connection. The pen-arm is made of non-corrosive monel metal and is of the inverted type. Operating instructions are lithographed on the chart plate so that they cannot be lost. Write for Catalog E-59.



American Air Duct Thermometer–Designed especially for both warm and cold air ducts. Fitted with chromium plated frame, glass front. Furnished with 9-in. or 12-in. scale graduated 0-160 F. Write for Catalog F-59.

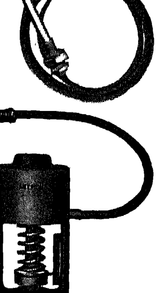


American Recording Thermometers—Made for recording temperatures from minus 40 to plus 1000 F or equivalent C. Very flexible connecting tubing up to 200 ft. One size to accommodate 10 in. chart, with an effective scale width of 3⅝ in.



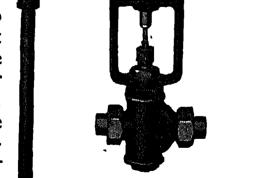
Same case as for the American Recording Gauge, so that all instruments are uniform in appearance when mounted on Gauge Boards. Write for Catalog H-59.

American Dial Thermometers—American Dial Thermometer (mercury-filled) has the accuracy of the standard glass tube thermometer and the reading convenience of a dial face. Entire working mechanism is made of steel, meaning long life.



Six sizes, ranging from 4½ in. to 12 in. diameter dials. Furnished with rigid connection or flexible capillary tubing up to 200 ft. For temperature ranges from minus 40 to plus 1000 F. Write for Catalog G-59.

American Precision Temperature Controllers—Self-operated. For regulating temperatures from 20 to 325 F. For hot water service tanks, water heaters, etc. Size of valve must be specified. Write for R-59 Bulletin.



GRINNELL COMPANY, INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence, R. I.**

Offices, Plants and Branches

Albany, N. Y.	Cleveland, Ohio (Branch)	New Orleans, La.
Atlanta, Ga. (Plant and Foundry)	Columbia, Penna. (Plant)	New York, N. Y.
Auburn, R. I. (Plant and Foundry)	Columbus, Ohio	Philadelphia, Penna. (Branch)
Baltimore, Md.	Dallas, Texas (Branch)	Pittsburgh, Penna.
Boston, Mass.	Detroit, Mich.	Providence R. I. (Plant and Foundry)
Buffalo, N. Y.	Kansas City, Mo.	Rochester, N. Y.
Charlotte, N. C. (Branch)	Long Island City, N. Y. (Branch)	St. Louis, Mo. (Branch)
Chicago, Ill. (Branch)	Milwaukee, Wis.	St. Paul, Minn. (Branch)
Cincinnati, Ohio	Minneapolis, Minn. (Branch)	Warren, Ohio (Plant and Foundry)
	Newark, N. J.	

GRINNELL COMPANY OF THE PACIFIC

Los Angeles, Cal. (Branch) Oakland, Cal. (Branch) San Francisco, Cal. (Branch) Seattle, Wash. (Branch)

GRINNELL COMPANY OF CANADA, LTD.

Montreal, Que. (Branch) Vancouver, B. C. (Branch) Toronto, Ont. (Plant and Foundry) Winnipeg, Man. Oshawa, Ont. (Foundry)

PRODUCTS AND SERVICES—

Complete Service on materials to Specification on Power Plant Piping, Industrial Piping, and Industrial Heating Systems; Prefabricated Piping including Pipe Cutting and Threading, Pipe Bends, Welded Headers, Welded and Welding Fittings, Lap Joints and the Grinnell line of products for Super Power.

Grinnell Equiflo Valves for forced hot water heating systems; Grinnell Adjustable Pipe Hangers and Supports; Grinnell Cast Iron and Malleable Iron Pipe Fittings; Grinnell Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoliers (Unit Heaters); Grinnell Thermofin (Convectors); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Constant Level Size Circulating Systems; Piping for acids and other special materials.

Malleable Iron, Brass, Bronze and other Castings; Brass, Cast Iron, Wrought Iron and Steel Pipe; Seamless Steel Tubing in Iron Pipe Sizes.

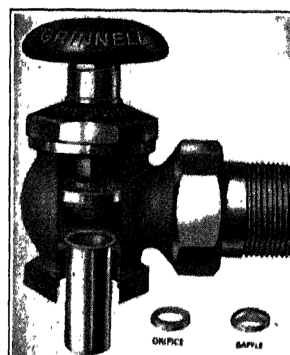
Valves: Check, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell "Junior" Automatic Sprinkler Systems for Basements and other hazardous areas of Dwellings, Small Apartment Buildings, Schools, Churches, Stores, etc.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

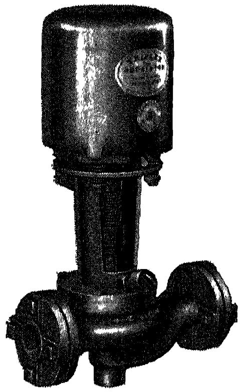
The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems. This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices and baffles capable of setting up any required frictional resistance. This method of establishing any desired resistance does away with elaborate calculation of pipe sizes. Grinnell guarantees perfectly balanced circulation to each and every radiator where these valves are installed throughout the system.

Equiflo Data Book sent to interested parties.

For Data On Thermoflex Traps and Heating Specialties, see page 1034

Triplex Heating Specialty Co., Inc.

242-268 Grant St. **Peru, Indiana**



Direct Drive

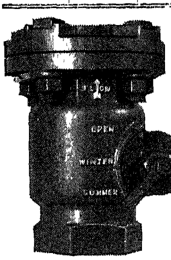
HOT WATER SYSTEMS AND SPECIALTIES

DIRECT DRIVE AQUALATOR

A most powerful Circulator. The design of the pump proper is the result of years of experiments. It will deliver water against an 11 ft head—about twice as powerful as direct driven pumps. The trouble-free coupler combined with the unique stuffing box assures lasting service. The Circulator is lubricated, protecting the bearings and stainless steel shaft. Every motor is protected with an automatic overload switch and meets all states, electrical codes.

SIZES AND CAPACITIES

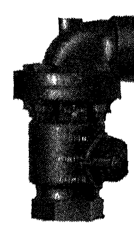
No.	Size	60 Cyc. 110V Motor Size	Circulator R. P. M.	Rad. Cap. Sq Ft	Storage Tank Cap.	Gals. Per Min	Max. Head	End to End	Ship. Wt.
22D	3/4″	1/8 H. P.	1750	300	500 gal.	20	138″	8¼″	55
23D	1″	1/8 H. P.	1750	500	1000 gal.	25	138″	8¼″	55
24D	1¼″	1/6 H. P.	1750	800	1500 gal.	35	138″	8¼″	60
25D	1½″	1/6 H. P.	1750	1200	2000 gal.	50	138″	8¼″	60
26D	2″	1/6 H. P.	1750	2000	2000 gal.	75	138″	8¼″	65



Straight Type Flow Control Valve

THERMOLATORS

Only one Flow Control Valve is required in a properly balanced Flow Control System. For best operation, the air should be eliminated from the boiler and carried directly to the expansion tank. The Anti Gurgle Fitting shown in the bottom of the Flow Control Valve positively accomplishes this. TRIPLEX Flow Control Valves have a convenient lever handle to set the valve for emergency or seasonal operation. The Anti Gurgle Fitting is not inlcuded with the price of the Thermolator.



Angle Type Flow Control Valve

No.	Size	Width	Height	Sq Ft Rad.	Ship. Wt.
123	1″	4″	4″	500	4
124	1¼″	5″	7½″	800	9
125	1½″	5½″	8½″	1200	11
126	2″	6″	9¾″	2000	15
127	3″	6¾″	9½″	5000	28

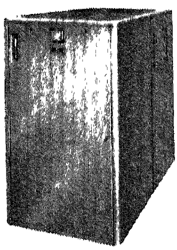
No. 127 is not an angle type. When ordering specify straight or angle type.

FLOW CONTROL SYSTEMS COMPLETE

SIZE INCHES	¾″ P. 1″ FCV	1″ P. 1″ FCV	1¼″ P. 1¼″ FCV	1½″ P. 1½″ FCV	2″ P. 2″ FCV	3″ P. 3″ FCV
No. of System	22-3-501	23-3-502	24-4-503	25-5-503	26-6-504	27-7-5044
Sq Ft Rad.	300	500	800	1200	2000	5000

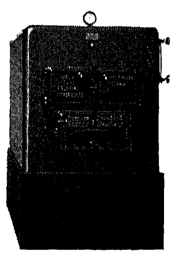
AMERICAN RADIATOR & Standard Sanitary

New York CORPORATION Pittsburgh



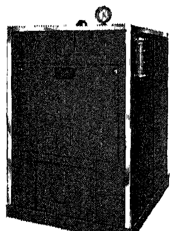
IDEAL OIL BOILER No. 8

A highly efficient, moderate priced Boiler for small homes. Also supplied as complete boiler-burner unit with Arcoflame Burner. Ratings: Steam—390 to 810 sq ft, Water—625 to 1,295 sq ft, installed radiation.



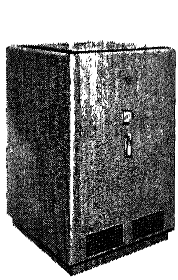
IDEAL ARCOFIRE STOKER-BOILER

Extra efficient, extra economical—especially designed for automatic stoker operation only. Ratings: Steam—900 to 1,775 sq ft, Water—1,440 to 2,840 sq ft, installed radiation.



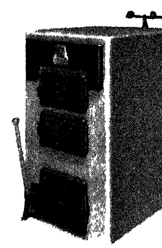
IDEAL BOILER No. 92

For dependable, economical automatic oil or stoker-fired coal operation in large homes, apartments or stores, etc. Ratings: Steam—1,450 to 2,600 sq ft, Water—2,320 to 4,160 sq ft, installed radiation.



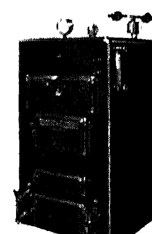
"EMPIRE" IDEAL GAS BOILER

Designed by experts to burn gas efficiently, economically. All controls concealed. Ratings: Steam—161 to 1,176 sq ft, Water—126 to 1,778 sq ft, installed radiation.



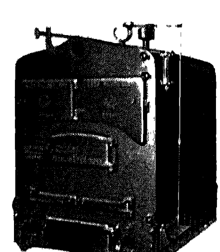
No. 7 IDEAL BOILER FOR COAL (stoker or hand-fired), OIL OR GAS

An exceptionally efficient, low-cost Boiler, for small and average size homes. Ratings: Steam—220 to 820 sq ft, Water—350 to 1310 sq ft, installed radiation.



IDEAL REDFLASH BOILERS (All Fuels)

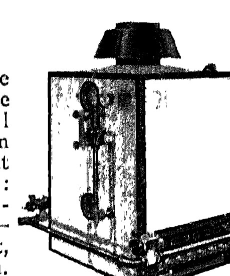
Economical heat for any size building. Multiple Asbestocel insulation. Five sizes. Ratings: Steam—290 to 9,900 sq ft, Water—465 to 15,840 sq ft, installed radiation.



IDEAL WATER TUBE BOILERS (All Fuels)

For large buildings. Five sizes, 23 in. to 79 in. Ratings: Steam—400 to 16,000 sq ft, Water—640 to 25,600 sq ft, installed radiation.

IDEAL GAS BOILER



Basically the same as the "Empire" Ideal Gas Boiler shown at left, but without jacket. Ratings: Steam—408 to 11,765 sq ft, Water—126 to 17,778 sq ft, installed radiation.

KEWANEE BOILER CORPORATION

Kewanee, Illinois

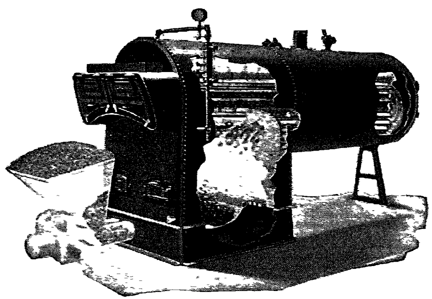
BRANCHES IN 64 PRINCIPAL CITIES

Steel Heating and Power Boilers, Water Heating Garbage Burners, Tabasco Heaters and Tanks.

KEWANEE STEEL HEATING BOILERS

Kewanee offers a dependable line of Steel Boilers built for heating every size building, with high efficiency, burning any kind of fuel. There are 365 standard sizes and 30 types of Kewanee Boilers most of which are kept in stock, ready for immediate delivery.

Seventy-two years of intensive study and effort are back of Kewanee Boiler designs. They are all constructed in our extensively equipped factory at Kewanee, Illinois, in conformity with these Codes: *American Society of Mechanical Engineers* for construction, and for rating with the *Steel Heating Boiler Institute* Simplified Practice.



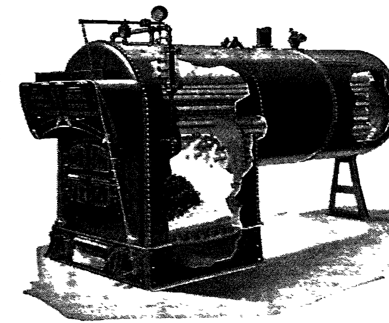
Firebox Boiler for Stoker "400" and "500" Series

The Kewanee series include:

HEAVY DUTY RIVETED FIREBOX TYPES: 1,240 ft to 42,500 ft. Brickset and portable settings, Updraft and Downdraft Smokeless Furnace, Single-pass tubes for rear smoke outlet; Two-pass tubes for front smoke outlet.

WELDED BOILERS: 2,200 ft to 42,500 ft. Direct Draft or Smokeless Arch with Corrugated Crown Sheet. Rear Smoke outlet and Weld + Rivet for front Smoke outlet.

RESIDENCE STEEL BOILERS: 790 ft to 2,924 ft. Square Type "R" with and without Jackets and Hot Water Heating Coils for Storage Tank or Instantaneous flow.



Firebox Boiler Portable Up-draft Type "400" and "500" Series

SPECIFICATIONS—PORTABLE UP-DRAFT BOILER

Boiler No.	576	577	578	579	580 480	581 481	582 482	583 483	584 484	585 485	586 486	587 487	588 488	589 489	590 490
Rated Steam Capacity:															
Coal .... Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker ..... Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length . In. x Ft In.	42x8-7	42x9-6½	48x8-10	48x9-6½	54x11-5	54x13-1½	60x13-4½	60x15-5½	66x15-6½	66x18-0½	72x17-0	78x17-7¼	78x21-3½	84x20-7	84x23-4
Overall Height Shell ...... In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line ...... In.	70	70	73	73	79½	79½	84½	84½	89½	89½	94	96½	96½	105	105
Approximate Weight: Coal . Lb	6100	6700	7300	7900	10200	11400	13300	14800	17300	19700	22000	24200	28400	32300	36100
Oil . . Lb	5500	6100	6600	7100	9300	10400	12200	13600	15800	18100	20300	22300	26400	30200	33800

Rated Capacity for Water Boiler is 60 per cent greater than Capacity for Steam Boiler.

Table for two series of Boilers lists maximum dimensions only.

Smith Twin Tubular Boiler Co., Inc.

State Road and Cottman Street

Philadelphia, Pa.

MANUFACTURERS STEEL HEATING BOILERS

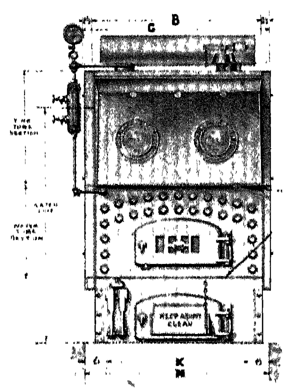
Smith Sectional Steel Boilers have gained favor with Architects and Engineers due to their compact construction and adaptability for installing in boiler rooms of building through existing doors or openings. Built on foundations or f.o.b. shop for coal, stoker or oil firing.

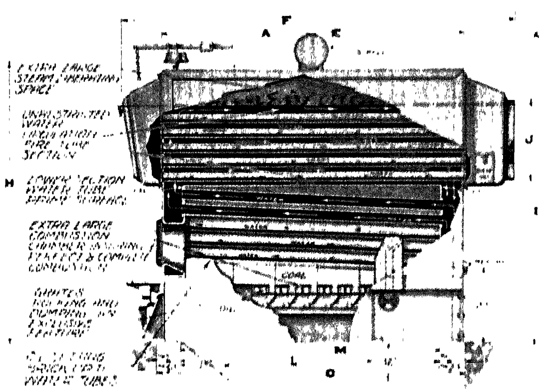
All the outstanding advantages of both the water tube and fire tube boilers without complicated baffle construction.

Faster steaming and higher efficiency obtained by rapid water circulation due to water tube construction that comprises 50 per cent of the boiler heating surface.

Write for circular for Domestic Boilers especially designed for Oil or Stoker firing.

SPECIFICATIONS—SMITH SECTIONAL-STEEL HEATING BOILER





Number of Boiler	2-60	2-72	2-84	2-96	2-108	3 84	3-96	3-108	3-120	4-108	4-120	4-132	4-144	5 132	5-144	5-156
Steam R. (Oil-Stoker)...	4800	5800	6800	7800	8800	10000	11400	12800	14200	16000	17800	19600	21400	24800	27100	29300
Heating Surface........	285	343	401	458	525	589	672	753	835	944	1050	1154	1262	1462	1593	1724
Water Line In..........	65	65	65	65	65	71	71	71	71	78	78	78	78	85	85	85
Floor to Water Tubes In.	42	42	42	42	42	42	42	42	42	46	46	46	46	50	50	50
Furnace Vol Cu Ft	51	63	75	88	100	93	108	124	139	161	181	201	221	253	279	304
Length of Shell In. A	60	72	84	96	108	84	96	108	120	108	120	132	144	132	144	156
Width of Shell In. B	51	51	51	51	51	61	61	61	61	71	71	71	71	81	81	81
Height Bottom Sec. C	31	31	31	31	31	42½	42½	42½	42½	48½	48½	48½	48½	53½	53½	53½
Height Top Sec. D	28	28	28	28	28	26½	26½	26½	26½	30½	30½	30½	30½	34½	34½	34½
Length Top Sec. E	60	72	84	96	108	84	96	108	120	108	120	132	144	132	144	156
Length of Boiler In. F	84	96	108	120	132	110	122	134	146	136	148	160	172	160	172	184
Width of Boiler In. G	56	56	56	56	56	66	66	66	66	76	76	76	76	86	86	86
Height of Boiler In. H	72½	72½	72½	72½	72½	78½	78½	78½	78½	84½	84½	84½	84½	92	92	92
Height Top Header I	84	84	84	84	84	90	90	90	90	98	98	98	98	107	107	107
No. & Size Outlets In...	1-8	1-8	1-8	1-8	1-8	1-8	1-8	1-8	1 8	1 10	1-10	1-10	1-10	1-10	1-10	1-10
No. & Size of Returns In.	1-4	1-4	1-4	1-4	1-4	1-4	1-4	1-4	1-4	1-6	1-6	1-6	1-6	1-6	1-6	1-6
Dia. Smoke Collar J	20	20	20	20	20	24	24	24	24	30	30	30	30	36	36	36
Height Stack Ft	50	55	60	65	70	65	70	75	80	75	80	85	90	85	90	95
Length of Pit In. M	50	62	74	86	98	68	80	92	104	92	104	116	128	104	128	140
Width Foundation N	56	56	56	56	56	66	66	66	66	76	76	76	76	86	86	86
Length Foundation O	62	74	86	98	110	86	98	110	122	110	122	134	146	134	146	158
Width Ash Pit K	44	44	44	44	44	54	54	54	54	64	64	64	64	74	74	74
Length Ash Pit L	50	50	56	56	62	52	58	64	64	58	64	70	70	64	64	70
Steam R. (Hand-Fired)..	4000	4800	5600	6400	7200	8250	9400	10500	11700	13200	14700	16200	17700	20500	22400	24000
Grate Size In..........	46x48	46x48	46x54	46x54	46x60	56x54	56x60	56x66	56x66	66x60	66x66	66x72	66x72	76x66	76x66	76x72
Grate Size Sq Ft.......	15.3	15.3	17.3	17.3	19.1	21.0	23.3	25.6	25.6	27.5	30.0	33.0	33.0	34.8	34.8	37.3

Standard Ratings conforming with the Industries "Simplified Practice Recommendations."

Special High-furnace Boilers furnished for Oil Burners or Stokers.

Iron Fireman Manufacturing Company

Automatic Coal Burners

Portland, Oregon

Factories: Portland, Ore.; Cleveland, Ohio; Toronto, Canada

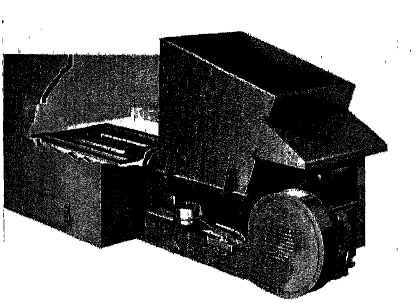
Retail Branches or Subsidiaries: Chicago, Ill.; Milwaukee, Wis.; St. Louis, Mo.; New York, N. Y.; Brooklyn, N. Y.; Montreal, Canada

Dealers in Principal Cities and Towns in the United States and Canada

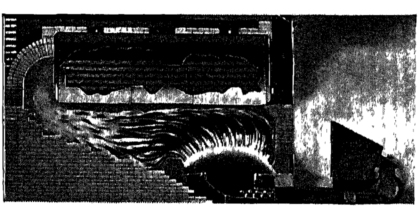
Representation in numerous foreign countries

IRON FIREMAN AUTOMATIC COAL STOKERS

COMMERCIAL HEATING MODELS



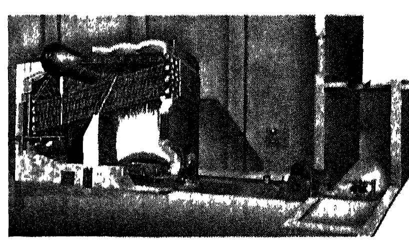
Commercial Installation—Hopper Model



Iron Fireman in Operation in Horizontal Return Tubular Boiler

Hopper Model

The Iron Fireman Commercial Heating stokers are general-purpose units, with wide application in schools, hotels, apartments, churches, office buildings, theaters and similar structures requiring a central heating plant . . . and in manufacturing plants, dairies and other establishments requiring heating and processing steam. The Commercial Heating stoker is the original type of Iron Fireman · the machine which made coal an automatic fuel.



Commercial Installation—Coal Flow model that carries coal direct from bunker to fire

MODEL	OUTPUT RANGE		
	Boiler Horsepower	Equivalent Direct Radiation	
		Steam (240 Btu)	Hot Water (150 Btu)
Standard Hopper	2 to 6	250 to 800	400 to 1,300
Deluxe Hopper	3 to 50	400 to 7,000	650 to 11,000
Coal Flow (available in all models)	3 to 500	400 to 70,000	650 to 110,000
Commercial and Industrial Standard Underfeed	20 to 350	2,500 to 50,000	4,000 to 75,000
Commercial and Industrial Poweram Underfeed	30 to 400	4,000 to 56,000	6,000 to 90,000
Commercial Anthracite	30 to 130	4,000 to 18,000	6,000 to 29,000
Pneumatic Spreader	50 to 1,000	7,000 to 140,000	11,000 to 225,000

Space Heaters—Capacities from 30,000 to 100,000 Btu per hour.

Combination Stoker—Winter Air Conditioning Units—Capacities from 60,000 to 210,000 Btu per hour.

Armstrong Machine Works

851 Maple Street **Three Rivers, Mich.**

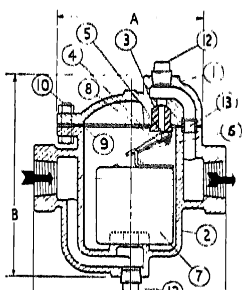
Representatives in All Principal Cities

Armstrong offers two types of traps for heating, air conditioning, and steam distribution service.

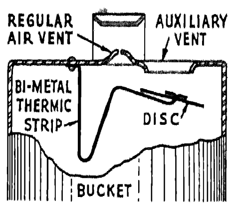
Standard Inverted Bucket Traps, the type originated by Armstrong, are *non-airbinding* and *self-scrubbing.* They are used for low, medium, and high pressure service where relatively little air must be handled along with the condensate. Their free-floating lever design makes it possible to open very large discharge orifices compared with the size of the trap itself.

Armstrong Blast Traps are used where large amounts of air must be vented quickly when steam is first turned on. They have several advantages over the conventional float and thermostatic trap.

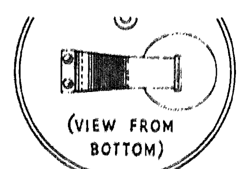
1. The Armstrong Blast Trap has but a *single orifice* to be maintained tight against the full pressure differential.
2. *Positive action.* The discharge valve in an Armstrong Blast Trap is either wide open or tight shut. Fast opening and fast closing prevent wire-drawing.
3. *Handles dirt.* There are no dead spots in an Armstrong Trap in which dirt can settle and interfere with the operation of the trap.



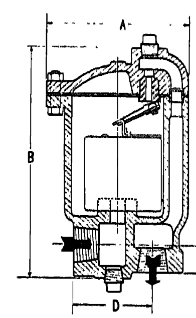
Cross-section of No. 800 trap for straight-through pipe connections. Part No. 13 is a new steel thimble in the discharge passage which insures positive positioning of gasket between cap and body.



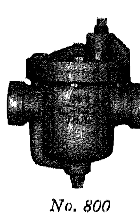
FOR BLAST TRAP JOBS



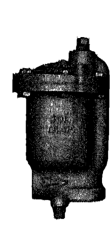
ALL Armstrong traps are readily convertible into "Blast" type traps merely by using buckets equipped with the patented auxiliary thermic air vent. As shown in the above sketches, the mechanism for this vent consists of a stainless steel disc slotted to receive the end of a bi-metal strip. Different coefficients of expansion in the bi-metal cause it to bend down when cold and up when hot. Normally, it is set to close at 212 deg, but it can be set to close at higher temperatures. Capacity, 50 to 100 times the air-venting capacity of a standard trap.



Cross-section of No. 801 trap for standard angle pipe connections.



No. 800



No. 801

Traps No. 800-801

Trap Size		No. 800	No. 801
Pipe Connections		½" or ¾"	½" or ¾'
List Price (Regular)		$7.00	$7.00
List Price (Blast Trap)		$8.50	$8.50
Telegraph Code (Regular)		Aloe	Arrow
Telegraph Code (Blast Trap)		Aloette	Arrowette
Dimension A		3¾"	3¾"
" B		5⅛"	6"
" C		5"	....
" D		....	2 1/16"
" E		....	11/16"
Number of Bolts		6	6
Diameter of Bolts		¼"	¼"
Weight		4½ lbs.	4½ lbs.
Maximum Pressure, lb		125	125
Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information see the Capacity Chart in Armstrong Steam Trap Book.	Lb Pressure 5	450	450
	10	560	560
	15	640	640
	20	690	690
	30	500	500
	50	580	580
	70	660	660
	100	640	640
	125	680	680

The American Brass Company

General Offices: Waterbury, Conn.

Manufacturing Plants:

ANSONIA, CONN. TORRINGTON, CONN. WATERBURY, CONN. BUFFALO N. Y. DETROIT, MICH. KENOSHA, WIS.

Offices and Agencies in Principal Cities

ANACONDA
from mine to consumer

CANADIAN PLANT: ANACONDA AMERICAN BRASS LIMITED, New Toronto, Ontario

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red-Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment

ANACONDA COPPER TUBES AND FITTINGS

For Heating, Plumbing and Air Conditioning

Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Fittings offer an unusual combination of advantages in hot water heating systems at a cost only slightly higher than black iron and approximately the same as wrought iron pipe. These advantages may briefly be summarized as follows:

Low Friction Loss—Because the inside surfaces of copper tubes are inherently smoother than those of pipe and tubes made of ferrous materials and also because they do not become roughened by the formation of rust, these tubes offer a minimum resistance to flow. In addition, the long radius turns of Anaconda Elbows and the smooth inside surface of Anaconda Wrought Copper Fittings further reduce friction losses.

These factors naturally increase the efficiency of the system, particularly when it includes a forced pressure circulator.

Ease of Installation—In many places the flexibility of copper tubes simplifies connections that ordinarily would be awkward and expensive to make with rigid pipe and threaded fittings. Anaconda Solder Fittings are compact. They can be installed in constricted space where the use of a wrench would be impossible.

Architects and builders naturally object to large holes and notches cut in the framing members of a building for the passage of piping. Anaconda Copper Tubes can be installed with a minimum of cutting in the structure, although holes should be large enough to permit movement of tubes due to expansion and contraction.

Appearance—Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Solder Fittings present an attractive appearance. It is a frequent practice to clean the tubes after they are installed and apply a coat of clear lacquer or similar substance. This keeps the tubes bright and makes an installation of which both plumber and owner can be proud.

Temper and Gauges—Anaconda Copper Tubes are made in both hard and soft temper and in standard wall thicknesses.

They meet the requirements for these types of tubes in U. S. Government Specification WW-T-799 and A.S.T.M. Specification B-88-39. Type K, the heaviest, is recommended for heating lines and general piping.

Accuracy of Dimensions—Anaconda Deoxidized Copper Water Tubes are all finished to the close tolerances required by the A.S.T.M. and Federal Specifications, which have been found essential for efficient assembly with solder fittings.

Permanent Identification—For permanent identification, the name "Anaconda" and the letter designating the type of tube is stamped in the metal at intervals of approximately 18 in., throughout every coil or straight length of tube.

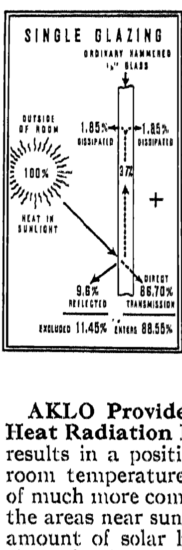
Libbey-Owens-Ford Glass Company

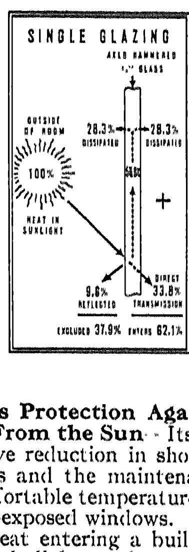
Toledo, Ohio

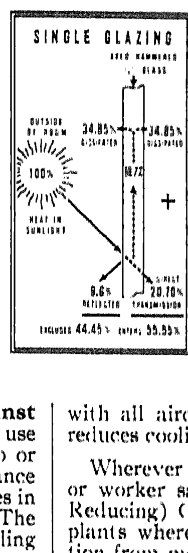
HEAT ABSORPTION AND GLARE REDUCTION WITH BLUE RIDGE AKLO

What AKLO Glass is

Blue Ridge AKLO is a blue-green low-expansion figured or wire glass that absorbs most of the sun's heat, admits an adequate amount of daylight, yet substantially reduces glare and eyestrain. For example, AKLO figured or wire glass ¼ inch thick absorbs about 97½ per cent of the infra-red (heat) rays of sunlight.







AKLO Provides Protection Against Heat Radiation From the Sun—Its use results in a positive reduction in shop or room temperatures and the maintenance of much more comfortable temperatures in the areas near sun-exposed windows. The amount of solar heat entering a building through windows, skylights and transoms is reduced as much as 48 per cent by the use of AKLO. And most important of all AKLO accomplishes this heat-absorption without excessive reduction in translucency—thus providing better illumination the year around.

AKLO Glass Results in Many Indirect Savings—Many indirect savings, too, result from the use of AKLO. For besides reducing glare and providing cooler summer temperatures, AKLO figured or wire glass reduces product spoilage, increases worker safety, decreases errors, speeds production and substantially improves worker efficiency. Used in windows and sky-lights instead of ordinary glass, it creates better working conditions and produces numerous direct savings in plant maintenance.

AKLO Reduces Airconditioning Costs—Because AKLO reduces air conditioning costs it is particularly vital in air conditioned buildings. It takes only about 100 sq ft of AKLO in a horizontal sky-light and from 200 to 250 sq ft in a westerly window to reduce the "design" cooling load by the equivalent of one ton of refrigeration as compared to ordinary glass. This amount of AKLO completely installed costs considerably less than a ton of extra capacity and cooling equipment. And thereafter the cost of operating the equipment is also reduced. AKLO glass should be specified in combination with all airconditioning installations. It reduces cooling load and operating costs.

Wherever glare is a problem, in efficiency or worker safety, AKLO Frosted (Glare Reducing) Glass should be installed. In plants where there is considerable reflection from moving parts of machinery, the safety factor alone, makes the installation of Frosted AKLO well worth while.

Despite the tremendous amount of heat that is absorbed, and its relatively low coefficient of expansion, AKLO resists thermal shock with amazing success.

SEND FOR FULL INFORMATION

For more detailed technical information about AKLO Glass, its properties and installation, send for our 12 page illustrated AKLO booklet—or refer to the engineering edition of Sweet's Catalog.

Johns-Manville

Executive Offices: 22 East 40th Street, New York, N. Y.

Offices in All Large Cities

Johns-Manville Home Insulation

Johns-Manville Rock Wool Home Insulation is a light, fluffy mineral wool, highly efficient in heat-proofing practically any building, old or new. It is durable, rot-proof, fire-proof and odorless, and will not corrode or settle. Full stud thickness of this material will cut fuel costs up to 30 per cent in winter and help keep rooms up to 15 deg cooler in hottest weather. J-M Rock Wool Home Insulation is furnished in two forms: for new construction, in easily handled batts; for existing buildings, in nodulated form to be installed pneumatically.

Applying J-M Super-Felt Type B batts in new home

For New Construction
J-M Super-Felt Batts Types B and C

Super-Felt Type B Home Insulation is furnished in pre-fabricated batts of uniform thickness and density, in both full stud thickness and semi-thick, in sizes 15 x 23 in. and 15 x 48 in., designed to fill completely the space between studs, joists and rafters on the usual 16 in. centers. The sturdy felted "wool" is strong enough to be handled rapidly without damage. The batts are backed with waterproof, vapor-resistant paper, extending on both the long sides in 1½ in. wide flanges, by which the batt is fastened in place and which also aid in sealing the joints. This backing protects against penetration of moisture from wet plaster and also resists infiltration of moisture vapor from the house into the wall.

As a further protection against moisture, the felt itself is also waterproofed.

Type C Super-Felt Home Insulation is an improved form of loose wool, in pieces 8 x 15 in., without paper backing, which readily fluffs to full wall thickness when installed.

Type C Super-Felt can be readily installed in irregular spaces since it can be easily cut or torn with the hands.

For Existing Homes and Buildings
Type A "Blown" Rock Wool

Type A Rock Wool is blown pneumatically into the spaces between studs in outer walls and between rafters or joists in roofs or attic floors. Insulation thickness in walls corresponds to stud depth, approximately 3⅝ in.; the density, approximately 5 to 8 lb per cu ft, assures maximum thermal efficiency. This type of insulation is installed only by Approved J-M Home Insulation Contractors, who are equipped with the necessary apparatus and trained crews.

Write for Details

Complete information on all types of J-M Rock Wool Home Insulation will be furnished on request.

J-M Airacoustic Sheets for lining Air-Conditioning Ducts

J-M Airacoustic Sheets, for duct linings of air conditioning systems, are flameproof, highly sound-absorbent and moisture-resistant, with a surface which will not materially increase friction losses in the duct system. Write for Bulletin AC-23A.

United States Gypsum Company

General Offices: 300 W. Adams Street, **Chicago, Ill.**

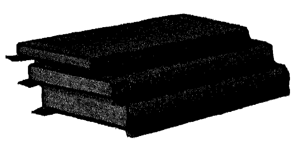
INSULATION PRODUCTS

Blanket **Decorative** **Structural**

Reflective **Vapor Barriers**

BLANKET

Red Top Insulating Blankets--Mineral wool fibers of great strength and uniformity, clean and free from non-insulating material are felted into light-weight blankets and securely fixed to an efficient asphalt-type vapor barrier. The vapor barrier forms the warm side of a complete paper enclosure, providing proper resistance to the passage of vapor, while a tough, perforated, vapor permeable paper on the cold side prevents the accumulation of any moisture within the blanket.



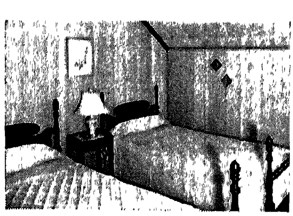
Red Top Insulating Blanket

Red Top Insulating blankets are so constructed that the insulating material is uniform in thickness throughout the length of each roll. They are as thick at the edges as in the center. Flanges at either side create an air space immediately back of the lath and plaster and effectively seal the joint against vapor leakage. Made in three thicknesses: One inch, medium and thick, in rolls of 50, 75 and 150 square feet (net area), respectively. Also available in bats 3 feet long in the same thicknesses.

Red Top Junior Bats 11 x 14 In.--Composed of the same fibers, 4 in. thick, to fit standard stud and joist spaces reduce costs for buildings which do not require vapor protection. Their "springy" fibers help hold the bats securely in position.

DECORATIVE INSULATION PRODUCTS

Weatherwood Building Board--An insulating wallboard 4 ft wide which fits across three standard stud or joist spaces, made in full foot lengths from 4 ft to 12 ft, inclusive, ½ in. and 1 in. thick in a "skin" finish in Ivory and a textured finish in either Ivory or Tan shades. Nailed directly to studs and joists this pleasing wall finish effectively insulates and decorates.



Decorative

Weatherwood Plank These half-inch "planks" of insulative wood fiber are manufactured in widths of 6, 8, 10, 12 and 16 inches, and in lengths 8, 9, 10 and 12 feet. They take advantage of the ingenious "Ogee" edge on their long edges (see cut) which conceals the nails and fully accommodates any necessity for movement in the board because of expansion or contraction without producing a crack through which air may leak to discolor the surface and reduce the insulation value of the material. When applied horizontally, they are blind nailed to the studding, and to cross furring when used vertically.

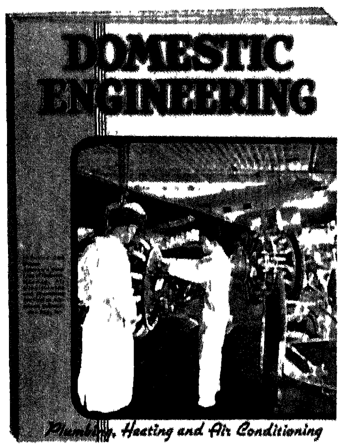
All sizes are shipped in a blend of gray and tan shades, in bundles of one size. When combined in the variations in shade and width, Weatherwood Plank produce maximum values in both insulation and decoration.

Domestic Engineering

Published by

DOMESTIC ENGINEERING PUBLICATIONS

1900 Prairie Avenue — **Chicago, Illinois**



IN the field of heating, plumbing and air conditioning, one publication has consistently demonstrated its leadership throughout its long years of service . . . one publication has consistently stood out.

Established in 1889, *Domestic Engineering* has devoted the intervening years to faithfully, untiringly serving its industry . . . to providing the type of technical and merchandising information that is accurate, authoritative and best suited to the needs of the men in this industry . . . to keeping them constantly abreast of changing times, changing conditions, changing trends.

For fifty-two years *Domestic Engineering* has been a leader in every major movement in its field. Editorially it has sponsored many programs which have been of material benefit to the industry as a whole.

As just one example of its leadership, *Domestic Engineering* forecast, long before the industry had become aware of it, the vital part heating, plumbing and air conditioning would play in National Defense. In a continuing program, *Domestic Engineering* has made the industry fully conscious of the tremendous responsibilities which National Defense, with its great era of industrial, commercial and housing expansion, has brought to it.

Each month *Domestic Engineering* is read by the men who buy, sell, specify and install the heating, plumbing and air conditioning equipment on these and similar projects.

Each month *Domestic Engineering* reaches the top-notch group of contractor-dealers who are responsible for the major portion of sales made in the heating, plumbing and air conditioning field.

Each month these men look to *Domestic Engineering* as their authority on matters pertaining to their business . . . as their source of dependable, helpful editorial co-operation and assistance.

These men have utmost confidence in *Domestic Engineering* and this confidence is carried over in fullest measure to the advertising pages. That's why, through *Domestic Engineering*, manufacturers of heating, plumbing and air conditioning equipment are afforded their most ideal, most effective means for reaching these men.

For complete data concerning *Domestic Engineering*, the field it serves, advertising rates, circulation, etc., write to Advertising Department, 1900 Prairie Avenue, Chicago, Illinois.

CONTROLLERS AND CONTROL EQUIPMENT (*See also Humidity and Temperature Control*)

American Moistening Co., 878
American Radiator & Standard Sanitary Corp., 992-993
Barber-Colman Co., 921, 940
Barber Gas Burner Co., 1014
Bristol Company, The, 941
Detroit Lubricator Co., 942-943
C. A. Dunham Co., 970-971
Julien P. Friez & Sons, Div. of Bendix Aviation Corp., 946
Fulton Sylphon Co., 944-945
General Electric Company, 916-917
Hoffman Specialty Co., Inc., 976-977
Illinois Engineering Co., 978-979
Illinois Testing Laboratories, Inc., 950
Johnson Service Co., 948-949
Kieley & Mueller, Inc., 1035
Leeds & Northrup Co., 951
Manning, Maxwell & Moore, Inc., 952
Mercoid Corporation, 953
Minneapolis-Honeywell Regulator Co., 954-955
Parks-Cramer Company, 822-823
Penn Electric Switch Co., 957
Powers Regulator Co., 958-959
Sarco Company, Inc., 980-981
Simplex Controls Div., The Herbusch Corp., 960
Spence Engineering Co., 961
Taylor Instrument Companies, 962-963
Warren Webster & Co., 984-987
Webster Engineering Co., 1015
Westinghouse Elec. & Mfg. Co., 826-827, 876
White-Rodgers Elec. Co., 965
Young Radiator Co., 860

CONVECTION HEATERS

American Radiator & Standard Sanitary Corp., 992-993
Crane Company, 998-999
C. A. Dunham Co., 970-971
Grinnell Co., Inc., 972-974, 1034
Kramer Trenton Co., 847
McQuay, Incorporated, 848-849
Modine Mfg. Co., 850-851
John J. Nesbitt, Inc., 854
Trane Company, The, 858-859
Tuttle & Bailey, Inc., 927-929
United States Air Conditioning Corp., 828
United States Radiator Corp., 1010-1011
Warren Webster & Co., 984-987
Weil-McLain Company, 1013
Young Radiator Co., 860

COOLING EQUIPMENT, Air

Aerofin Corporation, 883-885
Airtemp Div., Chrysler Corp., 830-831
American Blower Corp., 818-819
Autovent Fan & Blower Co., 899
Baker Ice Machine Co., Inc., 888
Binks Manufacturing Co., 880
Buffalo Forge Company, 903
Carbondale Division Worthington Pump & Machinery Corp., 896-897
Carrier Corporation, 820
Champion Blower & Forge Co., 904
Clarage Fan Company, 821
Curtis Refrigerating Machine Co., Div. of Curtis Manufacturing Co., 980
DeBothezat Division, American Machine & Metals, Inc., 905
C. A. Dunham Co., 970-971
Fedders Manufacturing Co., 842-843
Fitzgibbons Boiler Co., 1000-1001
Frick Company, Inc., 891
G & O Manufacturing Co., 886
General Electric Company, 916-917
General Refrigeration Corp., 892
Ilg Electric Ventilating Co., 906
F. Jaden Manufacturing Co., 846
Lau Blower Co., 907
Kramer Trenton Co., 847
Marlo Coil Company, 893
McQuay, Incorporated, 848-849
Meyer Furnace Co., 840
Modine Mfg. Co., 850-851
D. J. Murray Mfg. Co., Unit Heater and Cooler Div., 852
J. J. Nesbitt, Inc., 854
Herman Nelson Corp., 853
Refrigeration Economics Co., 823
Servel, Inc., 894
B. F. Sturtevant Co., 908-909
Trane Company, The, 858-859
United States Air Conditioning Corp., 828
Utility Fan Corporation, 914
Vilter Manufacturing Co., 898
Westinghouse Elec. & Mfg. Co., 826-827, 876
Williams Oil-O-Matic Heating Corporation, 836
L. J. Wing Mfg. Co., 855-857
York Ice Machinery Corp., 829
Young Radiator Company, 860

COOLING EQUIPMENT, Oil

Aerofin Corporation, 883-885
Carbondale Div., Worthington Pump & Machinery Co., 896-897
Frick Company, 891
G & O Manufacturing Co., 886
Modine Manufacturing Co., 850-851
Servel, Inc., 894
Universal Cooler Corp., 895
York Ice Machinery Corp., 829
Young Radiator Company, 860

COOLING EQUIPMENT, Oil, Cleaning Materials for

Oakite Products, Inc., 877

COOLING EQUIPMENT, Water (*See also Water Cooling*)

Acme Industries, Inc., 887
Aerofin Corporation, 883-885
Airtemp Div., Chrysler Corp., 830-831
American Blower Corp., 818-819
April Showers Company, 879
Baker Ice Machine Co., 888
Binks Manufacturing Co., 880
Buffalo Forge Company, 903
Carbondale Div., Worthington Pump & Machinery Corp., 896-897
Carrier Corporation, 820
Fedders Manufacturing Co., 842-843
Frick Company, Inc., 891
General Refrigeration Corp., 892
F. Jaden Manufacturing Co., 846
Marley Co., The, 881
McQuay, Incorporated, 848-849
Modine Mfg. Co., 850- 851
J. J. Nesbitt, Inc., 854
Refrigeration Economics Co., 823
B. F. Sturtevant Co., 908-909
Trane Company, The, 858-859
Universal Cooler Corp., 895
Vilter Manufacturing Co., 898
Westinghouse Elec. & Mfg. Co., 826-827, 876
Yarnall-Waring Co., 1038
York Ice Machinery Corp., 829
Young Radiator Company, 860

COOLING TOWERS, Atmospheric, Mechanical Draft, Forced Draft, Induced Draft (*See also Cooling Equipment, Water*)

American Blower Corp., 818-819
Baker Ice Machine Co., 888
Binks Manufacturing Co., 880
Buffalo Forge Company, 903
Marley Company, 881
Marlo Coil Company, 893
McQuay, Incorporated, 848-849
York Ice Machinery Corp., 829

COOLING TOWER FANS

American Blower Corp., 818-819
Baker Ice Machine Co., 888
Binks Manufacturing Co., 880
Buffalo Forge Company, 903
DeBothezat Div., American Machine & Metals, Inc., 905
Marley Company, 881
Marlo Coil Company, 893
McQuay, Incorporated, 848-849
York Ice Machinery Corp., 829

CORROSION, Treatment of

Oakite Products, Inc., 877
Vinco Company, Inc., 988-989

COVERING, Pipe

Armstrong Cork Company, 1060
Baker Ice Machine Co., 888
Carey, Philip, Co., 1061
Eagle-Picher Lead Co., 1063
Ehret Magnesia Mfg. Co., 1064-1065
Grinnell Co., Inc., 972-974, 1034
General Insulating & Mfg. Co., 1066
Insulation Industries, Inc., 1067
Johns-Manville, 1072-1073
Mundet Cork Corp., 1076
Owens-Corning Fiberglas Corp., 870-871
Pacific Lumber Co., 1077
H. W. Porter & Co., 1056
Ric-wiL Company, The, 1057
Ruberoid Company, The, 1078-1079
Standard Lime & Stone Co., 1081
United States Gypsum Co., 1082-1083

COVERING, Surfaces

Armstrong Cork Co., 1060
Baker Ice Machine Co., 888
Carey, Philip, Co., 1061
Eagle-Picher Lead Co., 1063
Ehret Magnesia Mfg. Co., 1064-1065
General Insulating & Mfg. Co., 1066
Insulation Industries, Inc., 1067
International Fibre Board, Ltd., 1071
Johns-Manville, 1072-1073
Mundet Cork Corp., 1076
Owens-Corning Fiberglas Corp., 870-871
Pacific Lumber Co., 1077
Reynolds Metals Co., Inc., 1080
Ruberoid Co., The, 1078-1079
Standard Lime & Stone Co., 1081
United States Gypsum Co., 1082-1083
York Ice Machinery Corp., 829

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Officers and Council

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

51 Madison Ave., New York, N. Y.

1940-41

President	F. E. GIESECKE
First Vice-President	W. L. FLEISHER
Second Vice-President	E. O. EASTWOOD
Treasurer	M. F. BLANKIN
Secretary	A. V. HUTCHINSON
Technical Secretary	JOHN JAMES

Council

F. E. GIESECKE, *Chairman*
W. L. FLEISHER, *Vice-Chairman*

One Year	*Two Years*	*Three Years*
N. D. ADAMS	E. K. CAMPBELL	J. F. S. COLLINS, JR.
A. P. KRATZ	S. H. DOWNS	E. N. MCDONNELL
J. F. MCINTIRE	A. J. OFFNER	T. H. URDAHL
J. H. WALKER	G. L. TUVE	C.-E. A. WINSLOW
G. L. WIGGS		

Committees of the Council

Executive: J. F. McIntire, *Chairman;* W. L. Fleisher, J. H. Walker.
Finance: A. J. Offner, *Chairman;* E. O. Eastwood, E. N. McDonnell.
Meetings: A. P. Kratz, *Chairman;* J. F. S. Collins, Jr., C.-E. A. Winslow.
Membership: N. D. Adams, *Chairman;* E. K. Campbell, T. H. Urdahl.
Standards: S. H. Downs, *Chairman;* G. L. Tuve, G. L. Wiggs.

Advisory Council

J. F. McIntire, *Chairman;* Homer Addams, R. P. Bolton, D. S. Boyden, W. H. Carrier, S. E. Dibble, W. H. Driscoll, H. P. Gant, E. Holt Gurney, John F. Hale, L. A. Harding, H. M. Hart, C. V. Haynes, E. Vernon Hill, John Howatt, W. T. Jones, D. D. Kimball, G. L. Larson, S. R. Lewis, Thornton Lewis, J. I. Lyle, F. B. Rowley and A. C. Willard.

Cooperating Committees

A.S.H.V.E. Representative on *National Research Council:* A. C. Willard (*one year*).

A.S.H.V.E. Representative on *A.S.A. Sectional Committee on Standardization of a Scheme for Identification of Piping Systems* (*A-13*): E. E. Ashley.

A.S.H.V.E. Representative on *N.F.P.A. Blower Systems Committee:* N. A. Hollister.

BEARMAN, Alexander A. (*M* 1937) Engr. (for mail) 20th Century-Fox Film Corp., 444 West 56th St., New York, and 47 Edward St., Baldwin, L. I., N. Y.

BEATTIE, James (*A* 1941) Prop., James Beattie, Htg. Contractor, 17215 Greenlawn Ave., Detroit, Mich.

BEAULIEU, Adrian A. (*M* 1937) Sales Engr., C. A. Dunham Co., 15 Hathaway St., Boston, and (for mail) 535 N. Elm St. W., Bridgewater, Mass.

BEAURRIENNE, Auguste* (*M* 1912) Consulting Engr., 18 Rue du Petit Val, Sucy en Brie, S et O, France.

BEAVERS, George R. (*M* 1929) Chief Engr. (for mail) Canadian Blower & Forge Co., Ltd., Woodside Ave., and 230 Cameron St. N., Kitchener, Ont., Canada.

BECHTOL, Jack J. (*J* 1937) Conditioner (for mail) Jos. E. Seagram & Sons, Inc., Lawrenceburg, Ind., and 4766 Rapid Run Rd., Cincinnati, O.

BECKER, C. S. (*M* 1939) Br. Mgr. (for mail) American Blower Corp., 1011 Majestic Bldg., Milwaukee, and 2532 North 62nd St., Wauwatosa, Wis.

BECKER, Roger K. (*M* 1938) Dept. Mgr., Ohio Valley Hardware & Roofing Co., and (for mail) 1017 E. Powell Ave., Evansville, Ind.

BECKER, Sidney J. (*J* 1940) Engr. (for mail) J. Becker, Inc., 282 Villeneuve St. W., and 272 Villeneuve St. W., Montreal, Que., Canada.

BECKER, Walter A. (*M* 1935) Sales Engr. (for mail) Grinnell Co., Inc., 4425 S. Western Ave., and 5728 N. Rockwell St., Chicago, Ill.

BECKWITH, F. J. (*M* 1940) Htg. Engr., Crane Co., 1007 W. Bay St., and (for mail) R. F. D. 1, Box 150, Jacksonville, Fla.

BEEBE, Frederick E. W. (*A* 1915) Sales Engr., Johnson Service Co., 28 East 29th St., New York, N. Y.

BEERY, Clinton E.* (*M* 1913) Pres., Heat & Fuel Engineering Co., Inc., 1454 Hood Ave., Chicago, Ill.

BEGGS, William E. (*M* 1927) Owner, W. E. Beggs Co., 416 Bell St., and (for mail) 2580 W. Viewmont Way, Seattle, Wash.

BEIGHEL, H. A. (*A* 1927) Sales Repr. (for mail) Herman Nelson Corp., 503 Columbia Bldg., Pittsburgh, and 207 Puritan Rd., Rosslyn Farms, Carnegie, Pa.

BEITZELL, Albert E. (*A* 1933; *J* 1930) Mgr., Air Cond. Div., Combustioneer Corp., 409 Tenth St. S. W., Washington, D. C., and (for mail) 6701 N. Central Ave., Chevy Chase, Md.

BELFORD, Louis duB. (*A* 1940) Dist. Supvr. (for mail) Minneapolis-Honeywell Regulator Co., Wayne & Roberts Ave., and 5722 Greene St., Philadelphia, Pa.

BELING, Earl H.* (*M* 1936; *A* 1930; *J* 1925) Owner, Beling Engineering Co., 405 State Trust Bldg., and (for mail) 2428-13th St., Moline, Ill.

BELL, E. Floyd (*M* 1933) Dist. Mgr., Buffalo Forge Co., 2102 Foshay Tower, Minneapolis, Minn.

BELL, Sydney R. (*M* 1939) Consulting Engr. (for mail) Sydney R. Bell & Associates, 374 Little Collins St., C-1, and 83 Queens Rd., S-C 3, Melbourne, Australia.

BELSKY, George A. (*A* 1937) Air Cond. Engr., Justin C. O'Brien Co., 734 Lexington Ave., New York, and (for mail) 78 Randolph Rd., Fulton Park, White Plains, N. Y.

BEMAN, Myron C. (*M* 1926) (Council 1934-39) Consulting Engr. (for mail) Beman & Candee, 374 Delaware Ave., and 699 Richmond Ave., Buffalo, N. Y.

BENHAM, Colin S. K. (*A* 1940; *J* 1937) Dir. (for mail) Benham & Sons, Ltd., 66 Wigmore St., London W. 1, and 5 Constable Close, London N. W. 11, England.

BENHAM, Ford C., Jr. (*J* 1938) Sales Engr. (for mail) c/o C. H. Ruebeck Co., P. O. Box 141, and 709-A James St., Waco, Tex.

BENNETT, Charles A. (*M* 1936) Asst. Mech. Inspector, Dept. of Justice, Bureau of Prisons, Rm. 1259 Dept. of Justic Bldg., and (for mail) 1752 Lamont St. N. W., Washington, D. C.

BENNETT, Edwin A. (*M* 1936; *J* 1929) Sales Engr. (for mail) American Blower Corp., 50 West 40th St., New York, and 45 Pondfield Rd. W., Bronxville, N. Y.

BENOIST, LeRoy L. (*M* 1934) Mgr. (for mail) Benoist Bros. Hardware & Supply Co., 117 S. Tenth St., and 1500 Main St., Mt. Vernon, Ill.

BENOIST, Raymond E. (*A* 1936) Mgr. & Engr., Benoist Bros. Hardware & Supply Co., 117 S. Tenth St., and (for mail) 811 North 12th St., Mt. Vernon, Ill.

BENSEN, Clarence L. (*M* 1939; *J* 1935) Chief Engr., McQuay, Inc., 1600 Broadway N. E., and (for mail) 3042 Benjamin St. N. E., Minneapolis, Minn.

BENSINGER, Mark (*J* 1936) Mech. Engr., Constr. Div. of Office of the Q. M. G., War Dept., and (for mail) 3718 Jocelyn St., N. W., Washington, D. C.

BENSON, Merrill L. (*M* 1938) Pres. (for mail) Northwest Engineering & Distributing Corp., 255 Foshay Tower, and 4521 Harriet Ave. S., Minneapolis, Minn.

BENTLEY, Clyde E. (*M* 1937) Consulting Engr., 216 Pine St., San Francisco, and (for mail) 1875 San Antonio Ave., Berkeley, Calif.

BENTZ, Harry (*M* 1915) Htg. & Vtg. Engr., 18 Holland Terrace, Montclair, N. J.

BERGAN, John R. (*A* 1941; *J* 1937) Dist. Repr. (for mail) Minneapolis-Honeywell Regulator Co., 1220 Madison Ave., Toledo, and 525 W. Broadway, Maumee, O.

BERGER, J. L. (*M* 1939) Vice-Pres. (for mail) The W. R. Rhoton Co., 5915 Bonna Ave., and 2652 Edgerton Rd., Cleveland, O.

BERMAN, Louis K. (*M* 1908) Pres. (for mail) Raisler Corp., 129 Amsterdam Ave., and 285 Central Park West, New York, N. Y.

BERMEL, Alfred H. (*A* 1933; *J* 1928) Estimator and Engr., August Arace & Sons, Inc., 642 Third Ave., Elizabeth, and (for mail) 177 Sunset Ave., N. Arlington, N. J.

BERNARD, Edgar L. (*J* 1941; *S* 1940) Heat Engr. (for mail) Reynolds Metals Co., 810 E. Franklin St., and 3132 Park Ave., Richmond, Va.

BERNERT, Lawrence A. (*A* 1937) Office Repr. (for mail) Warren Webster & Co., 316 Liberty Trust Bldg., and 515 Kensington Ave., Roanoke, Va.

BERNHARD, George (*M* 1935; *A* 1929) Prop. (for mail) National Improvement Co., Box 542, and 1316 High St., Marquette, Mich.

BERRIDGE, Winston W. (*M* 1938) Asst. Mgr., Industrial Dept. (for mail) McColl-Frontenac Oil Co., Ltd., Dominion Square Bldg., and 5169 Westbury Ave., Montreal, Que., Canada.

BERRINGER, Sidney H. (*M* 1926) Asst. Inspector of Engrg. Materials, U. S. Navy, and (for mail) 200 Main St., Buffalo, N. Y.

BERRY, Robert U. (*M* 1939) Engrg. Div., Air Cond. Dept. (for mail) General Electric Co., 5 Lawrence St., Bloomfield, and 423 N. Walnut St., East Orange, N. J.

BERRYMAN, Richard H. (*J* 1940) Htg. & Air Cond. Engr., Boyd-Cooper Heating Co., 14471 Livernois, and (for mail) 264 Worcester Pl., Detroit, Mich.

BERTOLETTE, Chester, Jr. (*M* 1940) Engr., Modern Automatic Heat Co., 14 Wood Lane, and (for mail) 206 S. Cherry St., Poughkeepsie, N. Y.

BERTRAND, George F. (*A* 1939) Sales Engr., Sheffler-Gross Co., 1000 Drexel Bldg., Philadelphia, and (for mail) 235 Richfield Rd., Upper Darby, Pa.

BERZELIUS, Carl E. (*M* 1936) Sales Engr., U. S. Gypsum Co., 300 W. Adams, Chicago, and (for mail) 2243-10th St., Rockford, Ill.

DICK, Harold (*S* 1940) Refrig. Serviceman, S. & S. Refrigeration, 183 Monticello Ave., Jersey City, and (for mail) 1235 Park Ave., Hoboken, N. J.

DICKASON, Gray D. (*M* 1938) Pres. (for mail) Genesee Heating Service, Inc., 950 Mercantile Bldg., and 140 Windemere Rd., Rochester, N. Y.

DICKENSON, Frederick R. (*M* 1936; *A* 1934) Mgr. Industrial Relations (for mail) American Blower Corp., 6000 Russell St., Detroit, and 715 Pilgrim Rd., Birmingham, Mich.

DICKENSON, Malcolm E. (*M* 1936) Pres. & Gen. Mgr. (for mail) Livingston Stoker Co., Ltd., 33 Sanford Ave. S., and 964 Cumberland Ave., Hamilton, Ont., Canada.

DICKEY, Arthur J. (*M* 1921) Vice-Pres. & Gen. Mgr., C. A. Dunham Co., Ltd., 1523 Davenport Rd., and (for mail) 9 Mossom Pl., Toronto, Ont., Canada.

DICKINSON, Robert P., Jr. (*J* 1938) Warehouse Mgr. (for mail) Burnham Boiler Corp. of Ohio, 301 Brushton Ave., Pittsburgh, and 219 Meade St., Wilkinsburg, Pa.

DICKSON, George P. (*M* 1919) Managing Dir., B. F. Sturtevant Co. of Canada, Ltd., 137 Wellington St. W., Toronto, Ont., Canada.

DICKSON, Robert B. (*M* 1919) Pres., Kewanee Boiler Corp., Kewanee, Ill.

DICKSON, Robert W., Jr. (*J* 1938) Sales Engr. (for mail) American Blower Corp., 1841 Oliver Bldg., Pittsburgh, and 520 Greendale Ave., Edgewood, Pa.

DIETZ, C. Fred (*M* 1938) Sales Engr. (for mail) Haynes Selling Co., Inc., 1124 Spring Garden St., and 1215 Allengrove St., Philadelphia, Pa.

DILL, Richard S.* (*M* 1939) Assoc. Mech. Engr., National Bureau of Standards, Washington, D. C., and (for mail) 1603 S. Springwood Dr., Silver Spring, Md.

DILLENDER, Eugene A. (*M* 1939) Htg. Engr. (for mail) United Gas Corp., Houston Div., Texas at San Jacinto, and 1841 W. Main St., Apt., 4, Houston, Tex.

DION, Alfred M. (*M* 1937) Sales Engr. (for mail) Trane Co. of Canada, King & Mowat Sts., and 356 Bloor St. E., Toronto, Ont., Canada.

DISNEY, Melvin A. (*A* 1934) Mgr. Air Cond. Dept., Fairbanks, Morse & Co., 1713 N. Market, and (for mail) 4223 Normandy, Dallas, Tex.

DISTEL, Robert E. (*A* 1941; *J* 1938) Gen. Mgr., Distel Heating Equipment Co., 404-406 Kalamazoo Plaza, Lansing, and 547 Bailey St., East Lansing, and (for mail) P. O. Box 133, Lansing Mich.

DIVER, Mortimer L. (*M* 1925) Consulting Engr., P. O. Box 1016, San Antonio, Tex.

DIXON, Arthur G. (*M* 1928) Sales Mgr. (for mail) Modine Mfg. Co., and 442 Wolff St., Racine, Wis.

DODDS, Forrest F. (*M* 1920) Mgr. (for mail) American Radiator & Standard Sanitary Corp., 1023 Grand Ave., and 4600 Mill Creek, Kansas City, Mo.

DODGE, Harry A. (*M* 1936) Elec. Engr., S. H. Kress & Co., 114 Fifth Ave., and (for mail) 514 West End Ave., New York, N. Y.

DOERING, Frank L. (*M* 1919) Sales, American Radiator & Standard Sanitary Corp., 238 Boston Ave., Lynchburg, Va.

DOLAN, Raymond G. (*M* 1926; *J* 1922) Secy.-Treas. (for mail) Tom Dolan Heating Co., Inc., 614-16 W. Grand, and 2112 Northwest 20th, Oklahoma City, Okla.

DOME, Alan G. (*A* 1938; *J* 1936) Air Cond. Engr., Bryant Air Conditioning Corp., 915 N. Front St., and (for mail) 28 Roumford Rd., Philadelphia, Pa.

DONELSON, William N. (*J* 1937) Sales Engr. (for mail) Modine Manufacturing Co., 633 S. Fifth St., and 2082 Douglas Blvd., Louisville, Ky.

DONNELLY, James A.* (*Life Member*; *M* 1904) (Treas., 1912-14) (Board of Governors, 1913) (Council, 1914) Largent, W. Va.

DONNELLY, Russell (*M* 1923) Sales Engr., Nash Engineering Co., Inc., 420 Lexington Ave., New York, N. Y.

DONOHOE, John B. (*A* 1937; *J* 1935) Engr. & Estimator (for mail) B. F. Donohoe Co., 51 Albany St., Boston, and 23 Primrose St., Roslindale, Mass.

DONOVAN, William J. (*A* 1930) 2239 North 27th St., Philadelphia, Pa.

DORFAN, Morton I. (*M* 1929) Dust Control Specialist, Pangborn Corp., 604 Chamber of Commerce Bldg., and (for mail) 1217 Malvern Ave., Pittsburgh, Pa.

DORNHEIM, G. A. (*M* 1912; *J* 1906) Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, and (for mail) 15 Hamilton Ave., Bronxville, N. Y.

DORSEY, Francis C. (*M* 1920) Engr.-Contractor (for mail) Francis C. Dorsey, Inc., 4520 Schenley Rd., Roland Park, and 212 Gittings Ave., Baltimore, Md.

DOSTER, Alexis (*A* 1934) Vice-Pres. & Secy., The Torrington Mfg. Co., 70 Franklin St., Torrington, Conn.

DOUGHTY, Charles J. (*M* 1925) Mgr. (for mail) C. J. Doughty & Co., 30 Brenan Rd., and 1202 Ave. Joffre, Shanghai, China.

DOUGLAS, Donald C. (*J* 1940) (for mail) Douglas & Seidler, Distributors, 1216 Connecticut Ave. N. W., and Shoreham Hotel, Washington, D. C.

DOUGLAS, Howard H. (*A* 1936) Air Cond. Engr. (for mail) Southern California Edison Co., 601 W. Fifth St., and 2317 Kelton Ave., Los Angeles, Calif.

DOWDY, Rufus B. (*M* 1939) Sales Engr. (for mail) Haydn Myer Co., Inc., P. O. Box 746, and 201 Capitol Pkwy., Montgomery, Ala.

DOWLER, Edward A. (*M* 1937) Sales Engr., B. F. Sturtevant Co. of Canada, Ltd., 137 Wellington St. W., and (for mail) 9 Prince Arthur Ave., Toronto, Ont., Canada.

DOWNE, Edward R. (*M* 1927) Chief Engr. (for mail) Bryant Heater Co., 17825 St. Clair Ave., Cleveland, and R. F. D. 2, Bell Rd., Chagrin Falls, O.

DOWNES, Alfred H. (*A* 1937) Draftsman, Holmes & Narver, 639 S. Spring St., and (for mail) 1342½ Bond St., Los Angeles, Calif.

DOWNES, Henry Hackett (*M* 1923) Mgr., American Blower Corp., 438 Woodward Bldg., Washington, D. C.

DOWNES, Nate W. (*M* 1917) Chief Engr. & Supt. of Buildings. (for mail) School District of Kansas City, Mo., 317 Finance Bldg., and 2119 East 68th St., Kansas City, Mo.

DOWNING, Clarence B. (*A* 1938) Secy.-Treas. (for mail) N. B. Downing Co., Jefferson Ave., and Clark Ave., Milford, Del.

DOWNS, Charles R. (*M* 1936) Vice-Pres. & Secy. (for mail) Weiss & Downs, Inc., 50 East 41st St, New York, N. Y., and 5 Sylvan Lane, Old Greenwich, Conn.

DOWNS, Sewell H. (*M* 1931) (Council, 1936-40) Chief Engr., Clarage Fan Co., and (for mail) 211 Creston Ave., Kalamazoo, Mich.

DRAIN, Harry E. (*M* 1940) Mfrs. Repr. (for mail) 404 Hubbell Bldg., and 6429 University, Des Moines, Ia.

DRAKE, George M. (*A* 1940; *J* 1936) Vice-Pres. (for mail) George H. Drake, Inc., 218 Lexington Ave., Buffalo, and 163 Renwood Ave., Kenmore, N. Y.

DREHER, Louis F. (*S* 1938) 4816 Margaretta Ave., St. Louis, Mo.

DRESCHER, Francis E. (*A* 1938) Sales Engr. (for mail) Straus-Frank Co., 1618 Fannin St., and 2725 Beatty St., Houston, Tex.

DRESSELL, Russell E. (*A* 1938) Mech. Engr., Riggs Distler Co., Inc., 216 N. Calvert St., and (for mail) 918 E. Preston St., Baltimore, Md.

HARTWEIN, Charles E. (*M* 1933) Htg. Engr., St. Louis County Gas Co., 231 W. Lockwood Ave., Webster Groves, and (for mail) 135 Pecke Ave., Kirkwood, Mo.

HARTWELL, Joseph C. (*M* 1922) Pres. & Treas. (for mail) Hartwell Co., Inc., 87 Weybosset St., and 16 Freeman Pkwy., Providence, R. I.

HARVEY, Alexander D. (*A* 1928; *J* 1925) Sales Mgr. (for mail) Kimberly-Clark Corp., and 819 E. Forest Ave., Neenah, Wis.

HARVEY, Lyle C. (*M* 1928) Pres., The Bryant Heater Co., 17825 St. Clair Ave., Cleveland, O.

HASHAGEN, John B. (*M* 1930) Plant Engr. (for mail) General Seafoods Corp., 1-15 Fish Pier, Boston, and 12 Park Ave., Cambridge, Mass.

HATTIS, Robert E. (*M* 1926) Consulting Air Cond. Mech. & Elec. Engr. (for mail) 332 S. LaSalle St., and 1454 Fargo Ave., Chicago, Ill.

HAUAN, Merlin J. (*M* 1923) Consulting Engr., 3412-16th S., Seattle, Wash.

HAUCK, Elden L. (*A* 1940; *J* 1936) Mgr. (for mail) Hauck Bros., 232 S. Center St., Springfield, and 1010 S. Main St., Dayton, O.

HAUER, Fred (*A* 1938) Owner (for mail) Fred Hauer & Co., 111 N. Water St., and 315 Hettinger Pl., Peoria, Ill.

HAUS, Irvin J. (*A* 1937; *J* 1935) Engr., Everett Smith Automatic Temperatures, Inc., 789 N. Water St., and (for mail) 5410 W. Center St., Milwaukee, Wis.

HAUSMAN, Louis M. (*M* 1935) Pres., L. M. Hausman & Co., 440 Dasmariñas, and (for mail) P. O. Box 1729, Manila, P. I.

HAUSS, Charles F.* (*Life Member*; *M* 1922) Via Gesu, No. 8, Milan, Italy.

HAWES, Harold D. (*S* 1940) Lab. Asst. in Engrg. Dept., Fulton Sylphon Co., and (for mail) Ft. Sanders Manor, Apt. 29, Knoxville, Tenn.

HAWISHER, Harold H. (*A* 1938) Mech. Engr. (for mail) Automatic Heating and Engineering Co., 416 N. Main St., and 411 S. Jamison Ave., Lima, O.

HAWK, Joseph K. (*M* 1939; *J* 1936) Sales Engr. (for mail) W. A. Case & Son Manufacturing Co., 31 Main St., and 203 Woodward Ave., Buffalo, N. Y.

HAYES, James J. (*M* 1920) Secy. (for mail) Stannard Power Equipment Co., 53 W. Jackson Blvd., and 7443 Jeffery Ave., Chicago, Ill.

HAYES, Joseph G. (*Life Member*; *M* 1908) Pres. & Engr. (for mail) Hayes Brothers, Inc., 236 W. Vermont St., and 2849 N. Capitol Ave., Indianapolis, Ind.

HAYMAN, A. Eugene, Jr. (*A* 1941; *J* 1935; *S* 1930) Draftsman, Moody & Hutchison, Cons. Engrs., 1701 Architects Bldg., Philadelphia, Pa., and (for mail) 2715 Washington St., Wilmington, Del.

HAYNES, Charles V. (*Life Member*; *M* 1917) (*Presidential Member*) (Pres., 1934; 1st Vice-Pres., 1933; 2nd Vice-Pres., 1932; Council, 1926-29; 1932-35) 115 Llanfair Rd., Ardmore, Mont. Co., Pa.

HAZLEHURST, Hamilton D. (*A* 1939) Southern California Gas Co., and (for mail) 1719 Morton Ave., Los Angeles, Calif.

HAZLETT, Dr. T. Lyle (*M* 1938) Medical Dir. (for mail) Westinghouse Electric & Manufacturing Co., East Pittsburgh, and 6634 Beacon St., Pittsburgh, Pa.

HEAGERTY, William H. (*A* 1923) Sales Engr. (for mail) 1427 Eye St. N. W., and 5100 N. Capitol St., Washington, D. C.

HEALY, Charles T. (*J* 1940) Minneapolis-Honeywell Regulator Co., 713 Maritime Bldg., New Orleans, La.

HEATH, William R. (*M* 1931) Asst. Chief Engr., Buffalo Forge Co., 490 Broadway, and (for mail) 119 Wingate Ave., Buffalo, N. Y.

HEBLEY, Henry F. J. (*M* 1934) Prod. Control Mgr., Pittsburgh Coal Co., P. O. Box 146, Pittsburgh, Pa.

HECHT, Frank H. (*M* 1930) Sales Engr. (for mail) B. F. Sturtevant Co., 2635 Koppers Bldg., and 1467 Barnesdale St., Pittsburgh, Pa.

HECKEL, Edmund P. (*M* 1918) E. P. Heckel & Associates, 407 S. Dearborn St., Chicago, and (for mail) 314 Cuttriss Pl., Park Ridge, Ill.

HECKLER, Samuel (*J* 1937) Engr. (for mail) Westchester Square Plumbing Supply Co., Inc., 4617 White Plains Ave., and 2800 Creston Ave., New York, N. Y.

HEDEEN, Laurel E. (*A* 1941; *J* 1938) Sales Engr., The Trane Co., 222 Walnut St., Waterloo, Ia.

HEDGES, H. Berkley (*M* 1919) Mgr., Industrial Sales (for mail) John J. Nesbitt, Inc., State Rd. & Rhawn Sts., Holmesburg, Philadelphia, and 114 Waverly Rd., Wyncote, Pa.

HEDLEY, Park S. (*M* 1923) Park S. Hedley Co., 361 Delaware Ave., Buffalo, N. Y.

HEEBNER, Walter M. (*M* 1922) Sales Engr., Warren Webster & Co., 20 Washington Pl., Newark, and (for mail) 282 Highwood Ave., Teaneck, N. J.

HEIBEL, Walter E. (*M* 1917) Dist. Mgr. (for mail) Aerofin Corp., 11 West 42nd St., New York, N. Y., and Old Greenwich, Conn.

HEIKKILA, Frank E. (*A* 1939) Resale Supvr., Westinghouse Electric & Manufacturing Co., 814 Ellicott Sq., Buffalo, and (for mail) 24 Somerton Ave., Kenmore, N. Y.

HEILMAN, Russell H.* (*M* 1923) Senior Industrial Fellow (for mail) Mellon Institute, 4400 Fifth Ave., and 2303 Beechwood Blvd., Pittsburgh, Pa.

HEINKEL, Charles E. (*J* 1938) Sales Engr., Control Equipment Co., 304 Selling Bldg., and (for mail) 3948 Northeast 20th, Portland, Ore.

HEISTERKAMP, Herbert W. (*A* 1940; *J* 1937) Sales Engr., Bryant Heater Co., 17825 St. Clair, and (for mail) 18103 E. Park Dr., Cleveland, O.

HELBURN, I. B. (*M* 1929; *J* 1927) Jr. Assoc. (for mail) Wyman Engineering, 1306 Chamber of Commerce Bldg., and 3815 Winding Way, Cincinnati, O.

HELLER, Joseph A. (*A* 1938) Sales (for mail) Air Conditioning Utilities Co., 8 West 40th St., and 150 West 82nd St., New York, N. Y.

HELLMERS, Charles C., Jr. (*J* 1937) Gas Htg. Engr. (for mail) Iowa-Nebraska Light & Power Co., 1401 O St., and 2554 Woodsdale Blvd., Lincoln, Nebr.

HELLSTROM, John (*A* 1929) Vice-Pres. (for mail) American Air Filter Co., Inc., 215 Central Ave., and 423 Lightfoot Rd., Louisville, Ky.

HELMRICH, G. Bernard (*M* 1936) Mech. Engr., The Detroit Edison Co., 2000 Second Ave., Detroit, and (for mail), 26590 Dundee Rd., Royal Oak, Mich.

HELSTROM, Clifford W. (*M* 1938) Mgr. Htg. & Plbg. Dept., Globe Machinery & Supply Co., and (for mail) 1614 Thompson, Des Moines, Ia.

HELSTROM, Herman G. (*M* 1928) Firebox Boiler & Stoker Div., William Bros. Boiler & Manufacturing Co., Nicollet Island, and (for mail) 4608 Arden Ave. S., Minneapolis, Minn.

HELWICK, Numa J. (*M* 1940) Secy. & Engr. (for mail) American Heating & Plumbing Co., Inc., 829 Baronne St., and 809 Greenwood St., New Orleans, La.

HENDERSON, Alexander S. (*J* 1940; *S* 1938) Engrg. Draftsman (for mail) S. F. (Australia) Pty. Ltd., Broughton House, 181 Clarence St., Sydney, and 65 Eastwood Ave., Eastwood, N. S. W., Australia.

HENDRICKSON, Harold M. (*M* 1934) Asst. Br. Engr. (for mail) York Ice Machinery Corp., 5051 Santa Fe Ave., Los Angeles, and 3901 Liberty Blvd., South Gate, Calif.

HENDRICKSON, Ralph L. (*M* 1938) Chief Engr. (for mail) Utilities Engineering Institute, 1314 Belden Ave., and 6125 Kenwood Ave., Chicago, Ill.

LEVITT, Leroy L. (*J* 1940) Air Cond. Engr., Calvert Distilling Co., Relay, and (for mail) 3326 Gwynns Falls Pkwy., Baltimore, Md.

LEVY, Marion I. (*M* 1938; *A* 1936; *J* 1931) Pres. (for mail) Viking Air Cond. Corp., 9500 Richmond Ave. S. E., and 3156 Ludlow Rd., Cleveland, O.

LEWIS, Carroll E. (*M* 1930) Sales Mgr., Delco Appliance Div., General Motors Sales Corp., 391 Lyell Ave., and (for mail) 9 Shelwood Rd., Rochester, N. Y.

LEWIS, Clyde A. (*A* 1939; *J* 1937) Engr.-Draft. Carrier Corp., Chrysler Bldg., New York, and (for mail) 23-51 123rd St., College Point, L. I., N. Y.

LEWIS, Harry E. (*J* 1939) Engr., Insulation Div., The Eagle Picher Sales Co., Temple Bar Bldg., and (for mail) 220 Atkinson St., Apt. No. 2, Cincinnati, O.

LEWIS, H. Frederick (*M* 1940; *A* 1937) Vice-Pres. & Gen. Mgr. (for mail) Harvey A. Dwight Oil Heat & Supply Co., Inc., 147 Dongan Ave., Albany, and Sweet's Crossing, Nassau, N. Y.

LEWIS, Kenneth C. (*A* 1938) 306 General Motors Research Bldg., Detroit, Mich.

LEWIS, L. Logan* (*M* 1918) Vice-Pres. (for mail) Carrier Corp., S. Geddes St., and 207 Sedgwick Dr., Syracuse, N. Y.

LEWIS, Samuel R.* (*M* 1905) (*Presidential Member*) (Pres., 1914; 2nd Vice-Pres., 1910; Board of Governors, 1909-12; Council, 1914-15) Consulting Engr. (for mail) 407 S. Dearborn St., and 4737 Kimbark Ave., Chicago, Ill.

LEWIS, Thornton* (*M* 1919) (*Presidential Member*) (Pres., 1929; 1st Vice-Pres., 1928; 2nd Vice-Pres., 1927; Council, 1923-30) Holiday Hill, R. D. No. 2, Newtown, Pa.

LEWIS, W. Warner (*A* 1930) Mgr. Appliance Dept. & Htg. & Air Cond. Dept. (for mail) Beeson Hardware Co., 214 N. Main St., and 604 Woodrow Ave., High Point, N. C.

LEY, Ralph B. (*A* 1940) Htg. & Vtg. Engr., 601 19th St. N. W., Apt. 803, Washington, D. C.

LIBBY, Ralph S. (*A* 1939; *J* 1933) Sales Engr., Sheldons, Ltd., 96 Grand Ave. S., and (for mail) 168 Grand Ave. S., Apt. 6, Galt, Ont., Canada.

LICANDRO, James P. (*J* 1938) Air Cond. Engr. (for mail) Carrier Corp., 704 Statler Bldg., Boston, and 3 Overlook Rd., Melrose, Mass.

LICHTY, Charles P. (*M* 1920) Mgr. (for mail) C. P. Lichty Engineering Co., 400½ S. 21st St., and 100 Devon Dr., Birmingham, Ala.

LIEBERMAN, Morris S. (*S* 1939) Draftsman, Engrg. Dept., Sears Roebuck & Co., Pittsburgh, and (for mail) 646 Sixth St., Oakmont, Pa.

LIEBLICH, Murray (*S* 1940) Student (for mail) Carnegie Institute of Technology, 4921 Forbes St., Pittsburgh, Pa., and 140 Riverside Dr., New York, N. Y.

LIFSHITZ, Hymen (*S* 1939) Student, Carnegie Institute of Technology, and (for mail) 2902 Webster Ave., Pittsburgh, Pa.

LIGE, Walter W. (*A* 1940) Resident Mgr., Bell & Gossett Co., Pittsburgh, and (for mail) 230 Birch Ave., Mount Lebanon, Pa.

LIGHT, John C. (*A* 1938) Mgr., Clow Gasteam Heating Co., 223 Wacaster St., Jackson, Miss.

LIGHTHART, Charles H. (*M* 1935) Mfrs. Sales Engr. (for mail) 511 Rhode Island St., Buffalo, and Eden, N. Y.

LILJA, Oscar L. (*A* 1937; *J* 1930) Mech. Engr., Magney, Tusler & Setter, 202 Foshay Tower, and (for mail) 5000 16th Ave. S., Minneapolis, Minn.

LINCOLN, Roland L. (*M* 1935) Mgr. Dust. Lab. (for mail) B. F. Sturtevant Co., Hyde Park, Boston, and 12 Lincoln St., Dedham, Mass.

LINDSAY, Griffith W., Jr. (*M* 1937) 301 N. Garland Ave., Dayton, O.

LINEBAUGH, John E. (*M* 1937) Frigidaire Engr., General Motors Overseas Operations, General Motors Bldg., and (for mail) 3859 Harvard Rd., Detroit, Mich.

LINGEN, Ralph A. (*A* 1939; *J* 1938) Dist. Mgr. (for mail) American Foundry & Furnace Co., 709 N. 11th St., Milwaukee, and 619 N. 61st St., Wauwatosa, Wis.

LINSENMEYER, Francis J. (*M* 1935) Director of Mech. Engrg. (for mail) University of Detroit, McNichols Rd. & Livernois, and 19020 Warrington Dr., Detroit, Mich.

LINSKIE, George A. (*J* 1939) Mgr., Furnace Dept. (for mail) Catlett Engineers, Inc., 2nd Unit, Santa Fe Bldg., 10th Floor, and 122 N. Jester St., Dallas, Tex.

LINTON, John P. (*M* 1927) Pres., Engineering Installations, Ltd., 1154 Beaver Hall Sq., Montreal, and (for mail) 247 Brock Ave. N., Montreal W., Que., Canada.

LIPSCOMBE, Harold W. J. (*M* 1938) Dir., The Lipscombe Air Conditioning Co., Ltd., Dacre House, Victoria St., London, S. W. 1, and (for mail) Glenmore, Woodland Way, West Wickham, Kent, England.

LITTLEFORD, Wallace H. (*M* 1936) Estimating Engr. (for mail) E. J. Febrey & Co., Inc., 616 New York Ave. N. W., Washington, D. C., and 106 Edmonston Rd., Hyattsville, Md.

LIVAR, Allen P. (*M* 1935) Chief Engr., Htg. Div. (for mail) Chrysler Corp., Airtemp Div., and 44 Ivanhoe Ave., Dayton, O.

LIVERMORE, James N. (*M* 1939) Engr., The Detroit Edison Co., 2000 Second Ave., Detroit, Mich.

LLOYD, Edmund H. (*J* 1936) Sales Engr., Bryant Air Conditioning Corp., 1626 K St. N. W., and (for mail) 2614 39th St. N. W., Washington, D. C.

LLOYD, Edward C. (*M* 1927) Dir., Tech. Service (for mail) Armstrong Cork Co., and Old Philadelphia Pike, Lancaster, Pa.

LOCK, Rowland H. (*M* 1939) Vice-Pres., J. H. Lock & Sons, Ltd., 221 Sterling Rd., and (for mail) 24 Kennedy Park Rd., Toronto, Ont., Canada.

LOCKE, James S. (*M* 1939) Sales Engr., Minneapolis-Honeywell Regulator Co., 433 E. Erie St., Chicago, and (for mail) 1828 Fairview Ave. S., Park Ridge, Ill.

LOCKE, Robert A. (*M* 1935) Mgr., Steel Heating Boiler Institute, and (for mail) Middletown, Pa.

LOCKHART, Harold A. (*A* 1936; *J* 1935) Chief Engr., Bell & Gossett Co., 3000 Wallace St., Chicago, and (for mail) 1453 Algonquin Rd., Des Plaines, Ill.

LOCKHART, William R. (*A* 1939; *J* 1936) Dist. Sales Mgr. (for mail) York Ice Machinery Corp., 215 Investment Bldg., and 3430 30th St. N. W., Washington, D. C.

LOEFFLER, Frank X. (*M* 1914) Pres. (for mail) Loeffler-Greene Supply Co., 1604 N. W. Fifth St., and 1811 N. W. Nineteenth St., Oklahoma City, Okla.

LOH, Nan-Shee (*M* 1933; *A* 1931; *J* 1927) Mgr., New Shanghai Heating & Plumbing Co., Room 330, National Commercial Bank Bldg., 400 Kiangse Rd., Shanghai, China.

LONG, Edward J. (*S* 1939) Student (for mail) Carnegie Institute of Technology, 6203 Monitor St., Pittsburgh, Pa., and 327 S. Douglas Ave., Springfield, Ill.

LONG, Herbert P. (*M* 1938) Sales Engr. (for mail) Buffalo Forge Co., 490 Broadway, Buffalo, and 336 Stillwell Ave., Kenmore, N. Y.

LONG, Wayne E. (*M* 1935) Prof., Mech. Engrg., A. & M. College of Texas., College Station, Tex.

LONGCOY, Grant B. (*M* 1933) Engr., Joseph Breslove, Cons. Engr., 845 Leader Bldg., Cleveland, and (for mail) 1215 Ramona Ave., Lakewood, O.

LOO, Ping Yok (*M* 1933) Gen. Mgr. (for mail) China Engineering Co., 30 Brenan Rd., Shanghai, and 271-73 Dunbarton Rd., Tientsin, China.

LOUCKS, David W. (*A* 1930) Supvr., Commercial Elec. & Steam Sales (for mail) Duquesne Light Co., 435 Sixth Ave., and 1049 Osage Dr., Wilkinsburg, Pittsburgh, Pa.

OLVANY, William J. (*M* 1912) Pres., Wm. J. Olvany, Inc., 100 Charles St., New York, N. Y.

O'NEILL, James W. (*M* 1929; *A* 1927; *J* 1925) Chief Engr., Trane Co. of Canada, Ltd., 4 Mowat Ave., and (for mail) 55 Highview Crescent, Toronto, Ont., Canada.

OONK, William J. (*M* 1937) Dist. Mgr., B. F. Sturtevant Co., 915 Olive St., and (for mail) 4548 Red Bud Ave., St. Louis, Mo.

OOSTEN, Louis S. (*J* 1938) Chief Draftsman, Bell & Gossett Co., 3000 Wallace St. S., and (for mail) 114 East Kensington Ave., Chicago, Ill.

O'REAR, Lawrence R. (*M* 1934) Pres. (for mail) Midwest Plumbing & Heating Co., 2450 Blake St., and 3033 West 37th Ave., Denver, Colo.

ORGAN, Frederick (*M* 1939) Archt. (for mail) 3724 Mason St., Omaha, Nebr.

ORGELMAN, George Henry (*S* 1940) 10 Pearl St., Danbury, Conn.

ORMISTON, Jack B. (*A* 1940) Owner (for mail) Ormiston Plumbing & Heating Co., 105 Manning Ave., and 1 Gramatan Dr., Yonkers, N. Y.

O'ROURKE, Hugh D., Jr. (*J* 1937; *S* 1936) Asst. Mgr., Unit Heater Dept., McCord Radiator & Mfg. Co., 2587 E. Grand Blvd., and (for mail) 14602 Wark Ave., Detroit, Mich.

ORR, George M. (*M* 1936) Pres. (for mail) G. M. Orr & Co., 542 Baker Arcade Bldg., and 1100 West 53rd St., Minneapolis, Minn.

ORR, Leighton* (*M* 1937) Research Engr., Pittsburgh Plate Glass Co., Research Laboratory, Creighton, and (for mail) 1116 Cambridge St., Tarentum, Pa.

OSBORN, Wallace J. (*A* 1927) Vice-Pres., Keeney Publishing Co., 1734 Grand Central Terminal Bldg., New York, N. Y., and (for mail) 1029 Old Post Rd., Fairfield, Conn.

OSBORNE, Gurdon H. (*M* 1922) Managing Dir., The Ventilating & Blow Pipe Co., Ltd., 714 St. Maurice St., and (for mail) 836 Pratt Ave., Outremont, Montreal, Que., Canada.

OSBORNE, Stanley R. (*M* 1939) Dist. Sales Mgr. (for mail) B. F. Sturtevant Co., 220 Delaware Ave., Buffalo, and 192 Euclid Ave., Kenmore, N. Y.

OSTDAHL, Harold E. (*J* 1941; *S* 1939) Research Engr. (for mail) Remington Arms Co., Research Lab., Bridgeport, and 236 California St., Stratford, Conn.

OSTER, William P. (*M* 1940) Vice-Pres. (for mail) Equitable Equipment Co., Inc., 410 Camp St., and 4651 Baccich St., New Orleans, La.

OSTROM, Eric W. (*M* 1937) Chief Engr. Air Cond. Dept., A/B Svenska Flaktfabriken, Kungsgatan 16, and (for mail) John Ericssonsgatan 18, Stockholm, Sweden.

OTT, Oran W. (*M* 1925) (Council, 1934-36) Consulting Mech. Engr. (for mail) 606 Washington Bldg., Los Angeles, and 1462 Waverly Rd., San Marino, Calif.

OURUSOFF, Leon* (*M* 1931) Mgr. of Utilization, Washington Gas Light Co., 411 Tenth St. N. W., Washington, D. C.

OUWENEEL, William A. (*M* 1937) Chief Engr., Standard Distributing Corp., 406 E. Wells St., Milwaukee, and (for mail) 801 Marshall Ave., South Milwaukee, Wis.

OVERTON, Sidney H. (*M* 1929) Repr. N. V. Radiatoren, Amsterdam, P. O. Box 5985, Johannesburg, South Africa.

OWEN, Jeff Davis (*M* 1937) Utility Engr., U. S. Army (Q. M. C.), March Field, and (for mail) 4003 Sixth St., Riverside, Calif.

OWINGS, Horace L. (*A* 1940) Air Cond. Engr. (for mail) Houston Natural Gas Corp., P. O. Box 1188, and 1305 McGowan, Houston, Tex.

P

PABST, Charles S. (*M* 1934) Pres. & Mgr., Pabst Air Conditioning Corp., 55 West 42nd St., New York, and (for mail) 8727-98th St., Woodhaven, L. I. ,N. Y.

PAETZ, George A. (*S* 1940) Engrg. Dept., Link Belt Co., 220 S. Belmont, and (for mail) 3308 College Ave., Indianapolis, Ind.

PAETZ, Herbert E. (*M* 1922) Div. Sales Mgr., American Blower Corp., 632 Fisher Bldg., and (for mail) 1415 Parker, Detroit, Mich.

PAGE, Arvin (*M* 1935) Chief Engr. (for mail) The Bahnson Co., 1001 S. Marshall St., and 628 Roslyn Rd., Winston-Salem, N. C.

PAGE, Harry W. (*M* 1923) Pres. (for mail) Wisconsin Equipment Co., 918 N. Fourth St., Milwaukee, and 7927 Warren Ave., Wauwatosa, Wis.

PAGE, Vernon C. (*A* 1936) Mgr., Air Cond. Div. (for mail) Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, and The Scarswold, Scarsdale, N. Y.

PAINE, H. Allan (*J* 1940) Mech. Engr., Healy Plumbing & Heating Co., Inc., 278 W. Kellogg Blvd., and (for mail) 452 Hamline Ave. S., St. Paul, Minn.

PAQUET, Jean-Marie (*A* 1940; *J* 1930) Civil Engr., J. A. Y. Bouchard, Inc., 97 Cote d'Abraham, and (for mail) 7 Ave. Belvedere, Apt. 2, Quebec, Que., Canada.

PARENT, Harold M. (*M* 1938) Partner (for mail) Parent & Kirkbride, N. W. Cor. Fourth & Locust Sts., Philadelphia, Pa., and 324 Pitman Ave., Pitman, N. J.

PARK, Harold E. (*A* 1938; *J* 1936) Sales Engr. (for mail) Shaw-Perkins Manufacturing Co., 1645 Oliver Bldg., Pittsburgh, and 43 W. Sixth St., Aspinwall, Pa.

PARK, J. Frank (*M* 1937; *A* 1936; *J* 1930) Sales Engr. (for mail) Western Air & Refrigeration, Inc., 1234 S. Grand Ave., and 2160 Kenilworth Ave., Los Angeles, Calif.

PARK, Nicholas W. (*M* 1936) Htg. Engr., Philadelphia Saving Fund Society, Real Estate Dept., 12 South 12th St., Philadelphia, and (for mail) 509 Jericho Rd., Abington, Pa.

PARKER, Dudley F. (*M* 1930) Sales Engr. (for mail) Nash Engineering Co., 420 Lexington Ave., New York, N. Y., and Morristown, N. J.

PARKER, Richard A. (*A* 1938) Pres. (for mail) Parker-Carpenter, Inc., 991 Bryant St., and 1464 Francisco St., San Francisco, Calif.

PARKERSON, William (*M* 1940) Partner (for mail) Air Control Engineering Co., 609 Baronne St., and 2105 General Pershing, New Orleans, La.

PARKINSON, John S.* (*A* 1940) Research Engr. (for mail) Johns-Manville Corp., Johns-Manville Research Labs., Manville, N. J.

PARKS, Charles E. (*M* 1937) Dist. Mgr. (for mail) Ilg Electric Ventilating Co., 805 Professional Bldg., and 284 W. Steuben St., Crafton, P. O., Pittsburgh, Pa.

PARRILLI, Roberto (*M* 1938) European Tech. Repr., Nash-Kelvinator Corp., Via Colonnetta 2, Milan, Italy.

PARROTT, Lyle G. (*M* 1922) Consulting Engr. (for mail) Snyder & McLean, 2308 Penobscot Bldg., and 3788 Gladstone, Detroit, Mich.

PARSONS, Leonard D., Jr. (*J* 1937; *S* 1936) Combustion Engr., Sears, Roebuck & Co., 925 S. Homan Ave., Chicago, and (for mail) 795 Park Blvd., Glen Ellyn, Ill.

PARSONS, Roger A. (*J* 1933) Htg. Engr. (for mail) Board of Water & Electric Light Commissioners, 114-116 W. Ottawa St., and 2609 Clifton St., Lansing, Mich.

PARTLAN, Robert L. (*J* 1940) Partner & Secy.-Treas. (for mail) Partlan Sheet Metal Works, 14268 Goddard Ave., and 20432 Yacama Ave. Detroit, Mich.

PARVIS, Ralph S. (*M* 1938) Mgr., Diamond Ice & Coal Co., 827 Market St., and (for mail) 2128 Biddle St., Wilmington, Del.

PASTOR, John C. (*M* 1938) 1091 Talbot Ave., Jacksonville, Fla.

SPENCER, Warner E. (*A* 1938) Repr., National Radiator Co., 220 Delaware Ave., Buffalo, N. Y.

SPIELMANN, Gordon P. (*A* 1931; *J* 1923) Vice-Pres. (for mail) Harrison-Spielmann Co., 480 Milwaukee Ave., Chicago, and 730 N. Prospect Ave., Park Ridge, Ill.

SPITZLEY, Joseph H. (*J* 1939) Test Dept., General Electric Co., 1 River Rd., and (for mail) 1032 Keyes Ave., Schenectady, N. Y.

SPITZLEY, Ray L. (*M* 1920) Pres. (for mail) R. L. Spitzley Heating Co., 1200 W. Fort St., Detroit, and 26 Renaud Rd., Grosse Pointe Shores, Mich.

SPOERR, Frank F. (*J* 1937) Chief Engr., Hearnen Air & Water Conditioning, S. Warren & W. Front St., Trenton, N. J., and (for mail) 140-19 Queens Blvd., Jamaica, L. I., N. Y.

SPOFFORTH, Walter (*M* 1930) Chief of Mech. Services, Federal Penitentiary, McNeil Island, and (for mail) 615 North Ainsworth Ave., Tacoma, Wash.

SPREKELMEYER, J. M. (*M* 1938) Pres. (for mail) General Engineering Corporation, 301 West 13th St., and 1912 Benhall Court, Fort Worth, Tex.

SPROULL, Howard E. (*M* 1920) Div. Sales Mgr. (for mail) American Blower Co., 1005-6 American Bldg., and 3588 Raymar Dr., Cincinnati, O.

SPURGEON, Joseph H. (*M* 1924) Sales Engr. (for mail) Spurgeon Co., 5-203 General Motors Bldg., and 17215 Pennington Dr., Detroit, Mich.

SPURNEY, Felix E. (*A* 1938) Bldg. Mgr., Board of Governors, Federal Reserve System, Washington, D. C., and (for mail) 28 W. Baltimore St., Kensington, Md.

STACEY, Alfred E., Jr.* (*M* 1914) Vice-Pres., Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, N. Y., and (for mail) 35 Wooton Rd., Essex Fells, N. J.

STACK, Arthur E. (*A* 1935) Asst. Mgr., Utilization Dept., Washington Gas Light Co., 411 Tenth St. N. W., Washington, D. C., and (for mail) 911 Gist Ave., Silver Spring, Md.

STACY, L. David (*A* 1936) Sales Engr., Ilg Electric Ventilating Co., 222 N. LaSalle St., Chicago, and (for mail) R. F. D. Prospect Heights, Mount Prospect, Ill.

STACY, Stanley C. (*M* 1931) Mech. Engr. (for mail) Board of Education, 13 S. Fitzhugh St., and 531 Wellington Ave., Rochester, N. Y.

STAFFORD, J. Fuller (*A* 1938) Owner, J. Fuller Stafford, Steam Specialties, 519 N. Snelling Ave., St. Paul, and (for mail) 2925-33rd Ave. S., Minneapolis, Minn.

STAFFORD, Thomas D. (*A* 1937) Secy.-Mgr., Alexander-Stafford Corp., 313-19 Allen St. N. W., and (for mail) 954 Ogden Ave. S. E., Grand Rapids, Mich.

STAHL, Walter A. (*M* 1938) Operating Mgr. (for mail) The Merchandsie Mart, Marshall Field & Co., 222 N. Bank Dr., Chicago, and 2504 Harrison St., Evanston, Ill.

STAMMER, Edward L. (*M* 1919) Supt. Htg. & Vtg. Repairs, Board of Education, 911 Locust St., and (for mail) 4430 Tennessee Ave., St. Louis, Mo.

STANDRING, Ronald A. (*J* 1938) Htg. Designer, (for mail) Gurney Foundry Co., Ltd., 100 Principal St., St. Laurent de Montreal, and 2358 Leclaire St., Maisonneuve, Montreal, Que., Canada.

STANGER, R. B. (*M* 1920) Prop. (for mail) Robinson & Stanger, Empire Bldg., Pittsburgh, and Middle Rd., Glenshaw, Pa.

STANGLE, William H. (*M* 1940) Mgr., New Products Div., Servel, Inc., 18th Floor, 51 East 42nd St., New York, and (for mail) 111-03 Seventy-Sixth Ave., Forest Hills, L. I., N. Y.

STANLEY, Robert L. (*M* 1938) Engr., Holly Heating & Manufacturing Co., 1000 S. Fairbanks Ave., S. Pasadena, and (for mail) 2518 Dearborn Dr., Los Angeles, Calif.

STANTON, Harold W. (*M* 1938) Commercial Engr. (for mail) Iowa-Nebraska Light & Power Co., and 2100 Ryons St., Lincoln, Nebr.

STARK, Clarence E. (*A* 1939) Vice-Pres., Montclair Plumbing Supply Co., 83-101 Walnut St., Montclair, and (for mail) 237 Sherman Ave., Glen Ridge, N. J.

STARK, W. E.* (*M* 1926) Regional Mgr., The Bryant Heater Co., 17825 St. Clair Ave., Cleveland, and (for mail) 1875 Rosemont Rd., East Cleveland, O.

STASZESKY, Francis M. (*S* 1938) Student (for mail) Massachusetts Institute of Technology, Cambridge, Mass., and 10 Roselawn Ave.' Roselle, Wilmington, Del.

STAUFFER, James E. (*S* 1940) Student (for mail) Carnegie Institute of Technology, 624 Clyde St., Pittsburgh, Pa., and 2000 N. Main St., Dayton, O.

STEEL, R. Justin (*A* 1938) Mgr. Htg. & Air Cond. Dept. (for mail) Wilmington Auto Sales Co., 221 W. Tenth St., Wilmington, and 175 W. Main St., Newark, Del.

STEELE, John B. (*M* 1932) Chief Operating Engr., Winnipeg School Board, Ellen and William Ave., and (for mail) 184 Waterloo St., River Heights, Winnipeg, Man., Canada.

STEFFNER, Edward F. (*A* 1937; *J* 1934) Htg. & Air Cond. Engr., The Henry Furnace & Foundry Co., 3471 East 49th St., Cleveland, and (for mail) 1429 East 133rd St., East Cleveland, O.

STEGGALL, Howard B. (*A* 1934) Br. Mgr. (for mail) United States Radiator Corp., 941 Behan St., and 1166 Murrayhill Ave., Pittsburgh, Pa.

STEIN, Jerome (*S* 1940) Estimator, Torrington Supply Co., Inc., 125 Maple St., and (for mail) 756 Waterville St., Waterbury, Conn.

STEINHORST, Theodore F. (*M* 1919) Pres. (for mail) Emil Steinhorst & Sons, Inc., 612 South St., and 1664 Brinckerhoff Ave., Utica, N. Y.

STEINKE, Bernard J. (*J* 1940; *S* 1937) Htg. & Vtg. Engr., 16 W. Palisade Ave., Englewood, and (for mail) Harrington Ave., Closter, N. J.

STEINMETZ, C. W. Arthur (*M* 1934) Office Mgr. (for mail) American Blower Corp., 249 High St., Newark, and 50 Oakwood Ave., Bogota, N. J.

STELLWAGEN, Frank G. (*A* 1937) Sales, Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, and (for mail) 8637-77th St., Woodhaven, L. I., N. Y.

STE. MARIE, Gaston P. (*M* 1930) Examiner Technician (for mail) Dept. of Labour, Quebec Provincial Government, 97 Notre Dame St. E., and 5329 Duquette Ave., N. D. G., Montreal, Que., Canada.

STENCEL, Reinhold Arthur (*M* 1938) Chief Engr. (for mail) Canadian Ice Machine Co., Ltd., 65 Villiers St., and 45 Willowbank Blvd., Toronto, Ont., Canada.

STENGEL, Frank J. (*A* 1935) Secy. (for mail) R. F. Stengel & Son, 76-80 Rosehill Pl., Irvington, and Mountain Ave., Murray Hill, N. J.

STEPHENSON, James R. (*A* 1940) (for mail) J. R. Stephenson & M. M. Perry, 311 Bank of Nova Scotia Bldg., and 647 Strathcona St., Winnipeg, Man., Canada.

STEPHENSON, Lewis A. (*M* 1917) Mgr. (for mail) The Powers Regulator Co., 409 East 13th St., and 801 West 57th Terrace, Kansas City, Mo.

STERMER, Clarence J. (*M* 1930) Engr., Crane Co., 836 S. Michigan Ave., and (for mail) 7839 Clyde Ave., Chicago, Ill.

STERNBERG, Edwin (*A* 1932; *J* 1931) Air Cond. Engr., Armo Cooling & Ventilating Co., 30 West 15th St., and (for mail) 16 East 98th St., New York, N. Y.

STERNE, Cecil M. (*A* 1934) Chief Engr., Metropolitan Refining Co., Inc., 50-23-23rd St., Long Island City, L. I., N. Y.

STERNER, Douglas S. (*A* 1940; *J* 1938; *A* 1936) Br. Mgr. Elec. Div. (for mail) Barber-Colman Co., 78 Baker St. N. W., and 1428 Peachtree St. N. E., Atlanta, Ga.

STETSON, Lawrence R. (*M* 1913) Engr. (for mail) McMurrer Co., 303 Congress St., Boston, and 35 Bradfield Ave., Roslindale, Mass.

Z

ZACK, Hans J. (*M* 1928) Pres., The Zack Co., 2311 Van Buren St., Chicago, Ill.

ZAKI, Hussein M. (*J* 1941; *S* 1940) Architectural Engr., Purdue Housing Research, Housing Research Foundation, Purdue University, Lafayette, and (for mail) 122 S. Grant St., West Lafayette, Ind.

ZEMELMAN, Irving M. (*J* 1941; *S* 1939) Asst. Engr., Sears-Roebuck & Co., Port Newark, and (for mail) 87 Schuyler Ave., Newark, N. J.

ZIBOLD, Carl E. (*M* 1929) Mech. Engr., Htg. & Vtg., 13 Chadwick Rd., Westminster Ridge, White Plains, N. Y.

ZIEBER, W. E. (*M* 1935) Dir. of Research (for mail) York Ice Machinery Corp., York, Pa.

ZIESSE, Karl L. (*A* 1931) Secy.-Treas. (for mail) Phoenix Sprinkler & Heating Co., 115 Campau Ave. N. W., and 315 Hampton Ave. S. E., Grand Rapids, Mich.

ZIMMERMAN, Alexander H. (*M* 1939; *A* 1930) Ventilation Engr., Chicago Dept. of Health, 54 W. Hubbard St., and (for mail) 6259 N. Francisco Ave., Chicago, Ill.

ZINK, David D. (*M* 1931) Engr.-Mgr., Air Cond. Dept. (for mail) B. D. R. Engineering Corp., 3316 Broadway, Kansas City, and Hickman Mills, Mo.

ZOKELT, Carl G. (*M* 1921) Consulting Engr., 3810–24th Ave. S., Seattle, Wash.

ZOLITSCH, Harry G. (*A* 1938) Partner (for mail) George J. Zolitsch & Son, 123 Grape St., and 392 Sanders Rd., Buffalo, N. Y.

ZUBER, Otto C. (*A* 1938) Chief Engr. (for mail) Amana Society, Amana, and South Amana, Ia.

ZUHLKE, William R. (*M* 1928) Hoffman Specialty Co., 77 Bedford St., Stamford, Conn.

ZUMWALT, Ross (*A* 1941; *J* 1938) Partner (for mail) Zumwalt & Vinther, 507 Thomas Bldg., Dallas, Tex.

ZUROW, William (*J* 1937) Sales Engr., St. Joseph Railway, Light, Heat & Power Co., 520 Francis St., and (for mail) 728 S. Tenth St., St. Joseph, Mo.

ZWALLY, August L. (*A* 1937) Chief Air Cond. Engr., Interstate Electric Co., and (for mail) 908 Elmwood St., Shreveport, La.

SPECIAL SERVICE ROLL

BISHOP, Joseph W. (*M* 1939) Mgr., Air Cond. Div., Toronto Dist. (for mail) Canadian General Electric Co., Ltd., 214 King St. W., Toronto, and 62 Highland Crescent, R. R. 2, York Mills, Ont., Canada.

BODMER, Emmanuel (*M* 1937) Head of Tech. Dept., Etablissements Dieny & Lucas, 223, Boulevard Pereire, and (for mail) 20, Rue Leon, Paris, France.

BUTT, Roderick E. W. (*A* 1936; *J* 1930) 605 Beatty House, Dolphin Square, London, S. W. 1, England.

CALVER, Robert W. (*A* 1937) 77 Queen St., Kirkland Lake, Ont., Canada.

FOSTER, Philip H. (*A* 1937) Business Mgr., Hudson Bay Plumbing Co., Flin Flon, Man., Canada.

FOX, John H. (*M* 1935) Sales Engr. (for mail) Minneapolis-Honeywell Regulator Co., Ltd., 117 Peter St., and 37 Macdonell Ave., Toronto, Ont., Canada.

JOHNS, Charles F. (*M* 1939; *A* 1931) Flight Lieutenant, Royal Canadian Air Force Headquarters, Jackson Bldg., and (for mail) 120 Wurtemburg St., Ottawa, Ont., Canada.

PENNOCK, William B. (*M* 1927) Pennock Engineering, 53 Queen St., and (for mail) 243 McLeod St., Ottawa, Ont., Canada.

PRICE, Ernest H. (*M* 1939; *A* 1937; *J* 1934; *S* 1932) Engr., W. G. Chester & Son, 179 Bannatyne Ave., and (for mail) 170 Harbison Ave., Winnipeg, Man., Canada.

PRYKE, John K. M. (*A* 1937) c/o Mrs. J. K. M. Pryke, 140 East 81st St., New York, N. Y.

SMITH, Gerald E. (*J* 1938) 52 Parkway Ave., Toronto, Ont., Canada.

Croton—
Elliott, I.

Dewitt—
Keane, G. F.

Dobbs Ferry—
Hewett, J. B.

Dunkirk—
Mayne, W. L.

Eden—
Ensign, W. A.

Elmira—
Davis, B. C.

Hastings-on-Hudson
Reynolds, T. W.

Irvington—
Bastedo, A. E.

Ithaca—
Ellwood, W. H.
Frederick, H. W.
Sawdon, W. M.
Williams, J. W.
Woods, E. H.

Kenmore—
Candee, B. C.
Criqui, A. A.
Heikkila, F. E.
Mollenberg, H. J.
Sommers, W. J.
Strouse, S. W.

Lockport—
Saunders, L. P.

Long Island—
Alt, H. L.
(St. Albans)
Apt, S. R.
(Flushing)
Bachmann, A. J.
(Jamaica)
Bastedo, G. R.
(Richmond Hill)
Blackburn, E. C., Jr.
(Garden City)
Bloom, L.
(Freeport)
Bonthron, R. C.
(Hempstead)
Carbone, J. H.
(St. Albans)
Clark, A. N.
(Manhasset)
Crone, T. E.
(Jackson Heights)
Eastwood, H. F.
(Merrick)
Fischer, L. W.
(Islip)
Franck, P.
(Jackson Heights)
Freeman, A. W
(Jackson Heights)
Fritz, C. V.
(Freeport)
Gebel, K. M.
(Rockaway Beach)
Graber, E.
(Douglaston)
Hiers, C. R.
(Great Neck)
Hirsch, M. H.
(Forest Hills)
Holland, G. R.
(Jackson Heights)
Iverstrom, C.
(Jackson Heights)
Jalonack, I. G.
(Jamaica)
Kadel, G. B.
(Valley Stream)
Kaiser, C. W.
(Woodside)
Keller, G. A.
(Wantagh)
Kern, J. F., Jr.
(Elmhurst)
Kunen, H. K.
(Kew Gardens)
Lane, D. D.
(Elmhurst)
Lang, J.
(Richmond Hill)
LaRaus, J.
(Jackson Heights)
Lewis, C. A.
(College Point)
MacWatt, D. A.
(Great Neck)
Magnusson, N.
(Jamaica)
Matzen, H. B.
(Rockville Centre)
McLeish, W. S.
(Flushing)
Meyer, C. L.
(Hollis)
Olsen, G. E.
(Arverne)
Pabst, C. S.
(Woodhaven)
Rainson, S.
(Port Washington)
Ranzinger, G.
(Whitestone)
Raynis, T.
(Manhasset)
Richfield, N. H.
(Floral Park)
Rudd, D. J.
(Babylon)
Schwartz, M.
(Far Rockaway)
Sherbrooke, W. A.
(Kew Gardens)
Spoerr, F. F.
(Jamaica)
Stangle, W. H.
(Forest Hills)
Stellwagen, F. G.
(Woodhaven)
Sterne, C. M.
(Long Island City)
Thomson, T. N.
(Huntington)
Trambauer, C W.
(Whitestone)
Tucker, F N.
(Freeport)
Wallace, G. J.
(East Elmhurst)
Whittaker, W. K.
(St. Albans)
Wittig, F. E.
(Glenwood Landing)

Mt. Vernon—
Cullin, W. W.
Freitag, F. G.
Herske, A. R.
Morse, F. W.

New Rochelle—
Abrams, A.
Fitz, J. C.
Mank, M.
Virrill, G. A.
Rose, H. J.

New York—
Adams, E. E.
Addams, H.
Addington, H. B.
Adler, H.
Akers, A. W.
Ambrose, E. R.
Ashley, E. E.
Baker, W. H., Jr.
Barbieri, P. J.
Baum, A. L.
Bearman, A. A.
Beebe, F. E. W.
Bennett, E. A.
Berman, L. K.
Blackmore, J. J.
Bodinger, J. H.
Bolton, R. P.
Bond, H. H.
Brown, D.
Buensod, A. C.
Carpenter, R. H.
Chase, C. L.
Chester, T.
Citron, D. J.
Clay, W.
Cucci, V. J.
Daly, R. E.
Darts, J. A.
Davis, C.
Davison, R. L.
Dee, L. H.
Demarest, R. T.
Denny, H. R.
Deterling, W. C.
Dodge, H. A.
Donnelly, R.
Downs, C. R.
Eadie, J. G.
Eder, J.
Elliott, L.
Engle, A.
Faile, E. H.
Farley, W. F.
Feder, N.
Feldman, A. M.
Fiedler, H. W.
Firestone, M. T.
Fleisher, W. L.
Foster, J. G.
Friedman, M.
Fuller, C. A.
Giannini, M. C.
Gitterman, H.
Goldsmith, E.
Gordon, P. B.
Greenburg, L.
Guler, G. D.
Gumaer, P. W.
Hall, C. J.
Hanburger, F. W.
Heckler, S.
Heibel, W. E.
Heller, J. A.
Henry, A. S., Jr.
Hering, A.
Herman, N. B.
High, J. M.
Hinrichsen, A. F.
Hobbie, E. H.
Hoffman, C. S.
Honerkamp, F.
Hosking, H. L.
Hotchkiss, C. H. B.
Hyman, W. M.
d'Issertelle, H. G.
Jacobus, D. S.
James, J. W.
Janos, W. A.
Jarcho, M. D.
Johnson, H. S.
Kaczenski, C.
Kahn, C. R., Jr.
Kepler, D. A.
Kimball, D. D.
Kingsland, G. D.
Koehler, C. S.
Kreiner, J.
Kuhlmann, R.
Kurth, F. J.
Lawrence, F. D.
Lee, B. H.
Lucke, C. E.
MaGee, K. B.
Markush, E. U.
Martens, E. D.
Martin, G. W.
McCann, F. D.
McClintock, W.
Meinke, H. G.
Meyer, H. C., Jr.
Milener, E. D.
Miller, C. A.
Miller, J.
Morro, J. J.
Mumford, A. R.
Munier, L. L.
Munkelt, F. H.
Offner, A. J.
Oldes, W. E. A. E.
Olson, R. G.
Olvany, W. J.
Page, V. C.
Parker, D. F.
Patorno, S. A. S.
Place, C. R.
Pollak, R.
Purinton, D. J.
Pryke, J. K. M.
Quirk, C. H.
Raisler, R. K.
Rather, M. F.
Reid, H. P.
Reis, R.
Reynolds, W. V.
Ritter, A.
Rosenberg, P.
Ross, J. O.
Roth, C. F.
Ryan, H. J.
Salter, E. H.
Samuels, S.
Sawhill, R. V.
Schechter, J. E.
Schoepflin, P. H.
Schulze, B. H.
Scott, G. M.
Seelig, A. E.
Sellman, N. T.
Senior, R. L.
Sklenarik, L.
Smith, M. S.
Sobel, F.
Sternberg, E.
Stevens, E. K.
Strock, C.
Strunin, J.
Syska, A. G.
Taylor, F., Jr.
Tiltz, B. E.
Timmis, W. W.
Tuthill, A. F.
Tyler, R. D.
Vetlesen, G. U.
Vroome, A. E.
Wadsworth, R. H.

1902

President .... A. E. Kenrick
1st Vice-President .... Andrew Harvey
2nd Vice-President .... Robert C. Clarkson
Treasurer .... Judson A. Goodrich
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, A. E. Kenrick

John Gormly, *Vice-Chm.*	J. H. Kinealy
R. C. Carpenter	C. B. J. Snyder
Wm. Kent	Wm. M. Mackay, *Secy.*

1903

President .... H. D. Crane
1st Vice-President .... Wm. Kent
2nd Vice-President .... R. P. Bolton
Treasurer .... Judson A. Goodrich
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, H. D. Crane

C. B. J. Snyder, *Vice-Chm.*	A. E. Kenrick
R. C. Carpenter	Geo. Mehring
John Gormly	Wm. M. Mackay, *Secy.*

1904

President .... Andrew Harvey
1st Vice-President .... John Gormly
2nd Vice-President .... Robert C. Clarkson
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, Andrew Harvey

John Gormly	H. D. Crane
Robert C. Clarkson	A. E. Kenrick
J. J. Blackmore	C. B. J. Snyder
R. C. Carpenter	Wm. M. Mackay, *Secy.*

1905

President .... Wm. Kent
1st Vice-President .... R. P. Bolton
2nd Vice-President .... C. B. J. Snyder
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, Wm. Kent

R. P. Bolton	James Mackay
C. B. J. Snyder	B. F. Stangland
B. H. Carpenter	J. C. F. Trachsel
A. B. Franklin	Wm. M. Mackay, *Secy.*

1906

President .... John Gormly
1st Vice-President .... C. B. J. Snyder
2nd Vice-President .... T. J. Waters
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, John Gormly

C. B. J. Snyder, *Vice-Chm.*	James Mackay
R. C. Carpenter	B. F. Stangland
Frank K. Chew	T. J. Waters
A. B. Franklin	Wm. M. Mackay, *Secy.*

1907

President .... C. B. J. Snyder
1st Vice-President .... James Mackay
2nd Vice-President .... Wm. G. Snow
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, C. B. J. Snyder

James Mackay, *Vice-Chm.*	Frank K. Chew
R. E. Atkinson	A. B. Franklin
R. C. Carpenter	Wm. G. Snow
Edmund F. Capron	Wm. M. Mackay, *Secy.*

1908

President .... James Mackay
1st Vice-President .... Jas. D. Hoffman
2nd Vice-President .... B. F. Stangland
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, James Mackay

Jas. D. Hoffman, *Vice-Chm.*	John F. Hale
B. F. Stangland	August Kehm
R. C. Carpenter	C. B. J. Snyder
Frank K. Chew	Wm. M. Mackay, *Secy.*

1909

President .... Wm. G. Snow
1st Vice-President .... August Kehm
2nd Vice-President .... B. S. Harrison
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, Wm. G. Snow

August Kehm, *Vice-Chm.*	Samuel R. Lewis
John R. Allen	James Mackay
R. C. Carpenter	B. F. Stangland
B. S. Harrison	Wm. M. Mackay, *Secy.*

1910

President .... Jas. D. Hoffman
1st Vice-President .... R. P. Bolton
2nd Vice-President .... Samuel R. Lewis
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. M. Mackay

Board of Governors

Chairman, Jas. D. Hoffman

R. P. Bolton, *Vice-Chm.*	John F. Hale
Geo. W. Barr	Samuel R. Lewis
R. C. Carpenter	James Mackay
Judson A. Goodrich	Wm. M. Mackay, *Secy.*

1911

President .... R. P. Bolton
1st Vice-President .... John R. Allen
2nd Vice-President .... A. B. Franklin
Treasurer .... Ulysses G. Scollay
Secretary .... Wm. W. Macon

Board of Governors

Chairman, R. P. Bolton

John R. Allen, *Vice-Chm.*	A. B. Franklin
John T. Bradley	Jas. D. Hoffman
R. C. Carpenter	August Kehm
James H. Davis	Wm. W. Macon, *Secy.*

Steam Heating in the Sky (*Heating, Piping and Air Conditioning*, October, 1938, p. 631).

Steam Heating for Modern Air Transports, by Jack McQuire (*Heating and Ventilating*, February, 1939, p. 37).

Altitude Conditioning (*Heating and Ventilating*, August, 1940, p. 16).

Ships

Ventilation and Air Conditioning of the S. S. Panama (*Heating and Ventilating*, September, 1939, p. 47).

Air Conditioning the New Mauretania (*Heating, Piping and Air Conditioning*, July, 1939, p. 431).

Heating, Ventilating and Air Conditioning on Shipboard, by J. H. Clarke (*Heating, Piping and Air Conditioning*, August, p. 467; September, p. 529; October, p. 610, 1940).

Chapter 38

INDUSTRIAL AIR CONDITIONING

Atmospheric Conditions Required, General Requirements, Classification of Problems, Control of Regain, Moisture Content and Regain, Conditioning and Drying, Control of Rate of Chemical Reaction, Control of Rate of Biochemical Reactions, Control Rate of Crystallization, Elimination of Static Electricity

IN the application of air conditioning to industrial processes, too much stress cannot be laid upon a thorough understanding by the air conditioning engineer of the problems involved. A complete knowledge of these problems is necessary before a satisfactory design can be made.

Individual processes and machines are changing rapidly and air conditions must be constantly revised to meet the new conditions.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity during processing depends upon the product and the nature of the process. As far as the behavior of the material itself and its desired final condition are concerned, each material and process presents a different problem. The best relative humidity may range up to 100 per cent. Similarly the most desirable temperature may range between wide limits for different materials and treatments. Extremes in either relative humidity or temperature require relatively expensive equipment for maintaining these conditions automatically. In departments where people are working, their health, comfort, and productive efficiency must be considered and often a compromise between the optimum conditions for processing and those required for the comfort of the worker is desirable.

It is generally considered that relative humidities below 40 per cent are on the dry side, conducive to low regains, a brittle condition of fibrous materials, prevalence of static electricity, and a tendency toward dryness of the skin and membranes of human beings. At the other end of the scale, humidities above 80 per cent are relatively damp, conducive to high regains, extreme softness, and pliability.

Table 1 lists desirable temperatures and humidities for industrial processing. In using this table, care must be taken in qualifying the process. In preparing many materials, conditions are not maintained constantly, but different temperatures and humidities are held for varying lengths of time.

TABLE 1. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
AUTOMOBILE	Assembly line	65	40
BAKING	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	80 to 90	80 to 95
	Storage of flour	70 to 80	60
	Storage of yeast	28 to 40	60 to 75
BIOLOGICAL PRODUCTS	Vaccines	below 32	
	Antitoxins	38 to 42	
BREWING	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
CERAMIC	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60	35
CHEMICAL	General storage	60 to 80	35 to 50
CONFECTIONERY	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
DISTILLERY	General manufacture	60	45
	Storage of grains	60	30 to 45
DRUG	Storage of powders and tablets	70 to 80	30 to 35
ELECTRICAL	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
FOOD	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats	0 to 10	50
	Storage of sugar	80	35
FUR	Drying of furs	110	
	Storage of furs	28 to 40	25 to 40

TABLE 1. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING (Concluded)

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
INCUBATORS	Chicken	99 to 102	55 to 75
LABORATORY	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
LEATHER	Drying of hides	90	
	Mulling	95 to 100	95
LIBRARY	Book storage (see discussion in this chapter)	65 to 70	38 to 50
LINOLEUM	Printing	80	40
MATCHES	Manufacturing	72 to 74	50
	Storage of matches	60	
MUNITIONS	Fuse loading	70	55
PAINT	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	
	Air drying of oil paints	60 to 90	25 to 50
PAPER	Binding, cutting, drying, folding, gluing	60 to 80	40 to 60
	Storage of paper	60 to 80	40 to 60
	Testing Laboratory	60 to 80	55 to 65
PHOTOGRAPHIC	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
PRINTING	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	60 to 75	20 to 60
	Storage of rollers	70 to 90	50 to 55
RUBBER	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
SOAP	Drying	110	70
TEXTILE	Cotton— carding	75 to 80	50 to 55
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	50 to 70
	weaving	68 to 75	85
	Rayon— spinning	70	85
	throwing	70	60
	weaving	75 to 88	60 to 75
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
	Testing Laboratory	70	65
TOBACCO	Cigar and cigarette making	70 to 75	55 to 75
	Softening	90	85
	Stemming or stripping	75 to 85	70

GENERAL REQUIREMENTS

In general, air conditioning apparatus for industrial purposes must be capable of absorbing heat from various sources such as machinery power, electric lights, people, sunlight and chemical reaction; of warming or cooling to any desired temperature, and of providing ample air supply at all times. Refrigeration may or may not be required, depending upon natural conditions, the required relative humidity and the maximum permissible temperature. Washing, purifying and recirculating of the air may be desirable. Good distribution is essential for the control of air motion and for the prevention of uneven conditions. Accurate, sensitive and reliable automatic control of humidity or temperature, or both, is vital in most cases.

Ordinarily, outside weather conditions and the ventilation required for workers are of secondary importance in relation to the total work to be done by the air conditioning system. In extreme cases of high concentration of industrial heat from machinery and ovens the error of entirely omitting the heat gain through the building structure would not be serious. At the other extreme, where low temperatures must be produced with refrigeration and where comparatively little power is used for driving the machinery, the heat gain through the building structure will become the major factor in determining the size of equipment and in this case the ventilation requirement assumes a normal degree of importance.

Buildings which are to be air conditioned should therefore be designed with careful consideration of overall cost and efficiency. Condensation resulting from high humidities must be prevented by suitable materials and construction, or else collected and drained to prevent loss of product or quick deterioration of the structure. Air leakage or filtration may add greatly to operating costs or make the maintenance of low humidities (relative or absolute) wholly impossible. Low temperatures require good insulation.

It is apparent that the subject of air conditioning for industrial processes is extensive and greatly involved, and that a detailed treatment is therefore beyond the scope of this book. A few of the salient points of the general subject are covered in this chapter.

CLASSIFICATION OF PROBLEMS

In general, any industrial air conditioning problem may be listed under one or more of the following five classes:

1. Control of Regain.
2. Control of Rate of Chemical Reactions.
3. Control of Rate of Biochemical Reactions.
4. Control of Rate of Crystallization.
5. Elimination of Static Electricity.

CONTROL OF REGAIN

In the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs, the temperature and relative humidity of the air have a marked influence upon the rate of production and upon the weight, strength, appearance and general

quality of the product. This influence is due to the fact that the moisture content of materials having a vegetable or animal origin, and to a lesser extent minerals in certain forms, comes to equilibrium with the moisture of the surrounding air.

In industries where the physical properties of a product affect its value, the percentage of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Standards of regain are firmly fixed in trade with fair penalties for excesses. Deficiencies result in loss of revenue to seller and loss of desirable quality to buyer.

Manufacturing economy therefore requires that the moisture content be maintained at a percentage favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A uniform condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight, it is proper that they contain a normal or standard moisture content.

MOISTURE CONTENT AND REGAIN

The terms *moisture content* and *regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains is dried to a constant weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not imply that the material as a whole has been completely dried out and has re-absorbed moisture. During the processing of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb moisture. A basis for calculating the regain of textiles is obtained by drying under standard conditions a sample from the lot and the dry weight thus obtained is used as a basis in the calculations to determine the regain.

The moisture content of an hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire different percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying out varies with the nature of the material, its thickness and density.

Table 2. Regain of Hygroscopic Materials

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Per Cent 10	20	30	40	50	60	70	80	90	Authority
Natural Textile Fibers	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fibre	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tire	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Foodstuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulphuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

Table 2 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 deg has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

The regain or moisture content affects the physical properties of textiles to a marked degree, changing the strength, pliability and elasticity.

The fact that the regain of textiles will come into equilibrium with the conditions of the surrounding air and vary with its temperature and relative humidity is the fundamental basis for the control of physical qualities during manufacture. During the preparation processes in a cotton mill, the cotton fibers should be in a condition to be easily carded.

These preliminary processes are carried out best in a relative humidity of 50 to 55 per cent. As the cotton fiber comes to the spinning operation, more flexibility is needed and the relative humidity is increased in this department. For many years, 65 per cent relative humidity was considered the optimum. To offset the extra work performed on the fiber as the spindle speed is increased, many cotton mills now carry 70 per cent relative humidity in the spinning rooms.[1] Winding, warping and weaving are all processes calling for great flexibility and a consequent need for higher humidity.

Other textile fibers, due to their different natural characteristics, are processed under relative humidities and temperatures applicable to each.

Rayons, on account of great loss of strength with the higher regains, should be processed in a relative humidity of 55 to 70 per cent. Acetate silk, another chemical fiber, with approximately 50 per cent of the regain of rayon, may be processed between 60 and 65 per cent relative humidity.

All hygroscopic materials release sensible heat equivalent to the latent heat of the moisture absorbed by the material, all of which may account for a small percentage of the total heat load.

CONDITIONING AND DRYING

In general, the exposure of materials to desirable conditions for treatment may be coincidental with the manufacture or processing of the materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The purpose of conditioning or drying is usually to establish a desired condition of moisture content and to regulate the physical properties of the material.

When the final moisture content is lower than the initial one, the term *drying* is applied. If the final moisture content is to be higher, the process is termed *conditioning*. In the case of some textile products and tobacco,

[1]The Present Status of Textile Regain Data, by A. E. Stacey, Jr. (*National Association of Cotton Manufacturers*, 1927).

skin cured and colored, after which the fruit is cooled to maintain as slow a rate of metabolism as possible. Ideal conditions range between 55 to 57 F and in no case should the temperature go below 49 F, as the starches then become fixed and are indigestible.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 to 59 F have been found to be best suited for this process. A high relative humidity of 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so it will be sufficiently tough to permit handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, the first three classifications of air conditioning are involved, and only through close atmospheric control can the best quality of the leaf be developed.

CONTROL RATE OF CRYSTALLIZATION

The rate of cooling of a saturated solution determines the size of the crystals formed. Both temperature and relative humidity are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating pans for pills, gum and nuts, a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right temperature and relative humidity. If the cooling and drying is too slow, the coating will be rough and semi-translucent, and the appearance unsightly; if too fast, the coating will chip through to the interior. Only by balancing temperature, relative humidity, and volume of air to the sugar solution, can the proper rate be obtained and a perfect coating assured.

ELIMINATION OF STATIC ELECTRICITY

The presence of static electricity is very detrimental to the satisfactory and economical processing of many light materials, such as textile fibers, paper, etc. It is also extremely dangerous where explosive atmospheres or materials are present. Fortunately, this hazard is easily eliminated by increasing the relative humidity.

In attempting to eliminate static electricity, it must be borne in mind that for successful elimination the air that actually comes in contact with the material in the machine must be at a relative humidity of 45 per cent or more. As some machines consume a great deal of power which is converted directly into heat, the temperature in the machine may be considerably higher than the temperature adjacent to the machine where the relative humidity is normally measured. In such cases, the relative humidity in the machine will be appreciably lower than that elsewhere in the room, and it may be necessary to maintain a room

relative humidity of 65 per cent, or even more, before the desired results can be obtained.

CALCULATIONS

The methods for determining the proper heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 5 and 6. Because of the large number of motors and heat processing units usually prevalent in an industrial application, it is particularly important that operating allowances for the latent and sensible heat loads be definitely ascertained and used in the calculations to determine the total design load.

REFERENCES

Effect of Air Conditioning upon Munitions, by J. I. Lyle (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 383).

Air Conditioning for Sausage Manufacturing Plants, by M. G. Harbula (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 343).

Air Conditioning and Refrigerating Large Bakeries, by W. L. Fleisher (*Heating, Piping and Air Conditioning*, December, 1929, p. 621; January, 1930, p. 24).

Air Conditioning for Textile Plants Making and Using Synthetic Yarns, by L. L. Lewis (*Rayon Textile Monthly*, July, August, September, 1930).

Air Conditioning in the Bakery, by W. L. Fleisher (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1931, p. 158).

Air Conditioning in Industrial Processes (Match Factory), by W. L. Fleisher (*Heating, Piping and Air Conditioning*, March, 1931, p. 196).

Air Conditioning of Press Rooms, by I. C. Baker (*Heating, Piping and Air Conditioning*, July, 1931, p. 553).

Pre-Cooling Fruits and Vegetables with Circulating Air, by C. E. Baker (*Heating, Piping and Air Conditioning*, January, 1932, p. 42).

Air Conditioning Maintains Quality of Fruits and Vegetables, by C. E. Baker (*Heating, Piping and Air Conditioning*, August, 1935, p. 369).

Air Condition the Bakery Throughout, by W. W. Reece (*Heating, Piping and Air Conditioning*, August, 1936, p. 149).

Air Conditioning as Applied in Theatres and Film Laboratories, by D. C. Lindsay (*Transactions Society of Motion Picture Engineers*, April, 1927, Vol. XI, No. 30, p. 335-365).

Air Conditioning Requirements of Multicolor Offset Printing, by C. G. Weber (*Refrigerating Engineering*, December, 1936, p. 6).

Banana Ripening Manual, Circular No. 14, Equipment Department, Fruit Dispatch Co., New York, N. Y.

The Commercial Storage of Fruits, Vegetables and Florists' Stocks, by D. H. Rose, R. C. Wright and T. M. Whiteman (*U. S. Department of Agriculture*, Circular No. 278).

Reactions of Lithographic Papers to Variations in Humidity and Temperature, by C. G. Weber and L. W. Snyder (*U. S. Bureau Standards Journal Research*, January, 1934).

Relation of Air Conditions to Tobacco Curing, by J. Johnson and W. B. Ogden (*Wisconsin Agricultural Research Bureau* 110: 1-48, 1931).

Temperature Studies of Some Tomato Pathogens, by Alice A. Nightingale and G. W. Ramsey (*U. S. Department of Agriculture Technical Bulletin* No. 520, August, 1936).

The Treatment of Offset Papers for Optimum Register, by C. G. Weber and M. N. V. Geib (*U. S. Bureau Standards Journal Research*, February, 1936).

Summary Report of National Bureau of Standards Research on Preservation of Records, by A. E. Kimberly and B. W. Scribner (*U. S. Bureau of Standards Miscellaneous Publication*, 154, March 16, 1937).

Chapter 39

INDUSTRIAL EXHAUST SYSTEMS

Classification of Systems, Design Procedure, Requirements for Suction and Velocity, Hoods, Design of Duct Systems, Collectors, Resistance of Systems, Efficiency of Exhaust Systems, Selection of Fans and Motors, Corrosion

IN almost every industry some type of exhaust or collecting system is essential to achieve efficient and economical control of dusts and fumes. General design information is included in this chapter which is intended to relate primarily to factory exhaust systems.

CLASSIFICATION OF SYSTEMS

There are two general arrangements, the central and the group systems. In the central system a single or double fan is located near the center of the shop with a piping system radiating to the various machines to be served. In the group system, which is sometimes employed where the machines to be served are widely scattered, small individual exhaust fans are located at the center of the machine groups. The group arrangement has the advantage of flexibility.

Exhaust systems are also classified by the means employed to collect dust or other material handled. The dust or refuse may be collected and controlled by enclosing hoods, open hoods, inward air leakage, or by exhausting the general air of the room.

With some classes of machinery it is not feasible to closely hood the machines and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

The open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. When the hood must be placed at some distance above the machine it should be large enough to encompass an area of considerable extent as diffusion is usually quite rapid.

Consideration must also be given to the natural movement of the fumes. For those that are lighter than air the hood should be over or above the machine and where a heavy vapor or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are required. If it is attempted to remove heavy dust such as lead oxides by an overhead hood the conditions may be worse than if no exhaust were used at all, owing to the rising air current carrying the dust up through the

breathing zones. The objective to keep in mind in all cases is to take advantage of the natural tendency of the material to move upward or downward.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, the removal of some dust from the machine or enclosure being merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is required to handle only the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating, and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a strong current of clean air is drawn across the operator, and away from him toward the work, where the dust is picked up and carried from the room.

DESIGN PROCEDURE

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. Finally, in choosing the fan, the area of the intake should be equal to or greater than the sum of the areas of the branch ducts. The speed, of course, must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements of an efficient exhaust and collecting system are as follows[1]:

1. Hoods, ducts, fans and collectors should be of adequate size.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should not interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.
6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.
7. The design of an exhaust system should afford easy access to parts for inspection and care.

[1]For more detailed requirements see Safe Practice Pamphlets Nos. 32 and 37, published by the *National Safety Council*, Chicago.

REQUIREMENTS FOR SUCTION AND VELOCITY

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it to a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood should be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible.

Table 1. Size of Connections for Wood-Working Machinery

Type of Machine	Diameter of Connections in Inches
Circular saws, 12-in. diam.	4
Circular saws, 12-24-in. diam.	5
Circular saws, 24-40-in. diam.	6
Band saws, blade under 2 in. wide	4
Band saws, blade 2-3 in. wide	5
Band saws, blade 3-4 in. wide	6
Band saws, blade 4-5 in. wide	7
Band saws, blade 5-6 in. wide	8
Small mortisers	6
Single end tenoners	6
Double end tenoners	7
Double end, double head tenoners	10
Planers, matchers, moulders, stickers, jointers, etc.—	
With knives, 6-10 in.	5–6
With knives, 10-20 in.	6–8
With knives, 20-30 in.	6–10
Shapers, light work	4–5
Shapers, heavy work	8
Belt sander, belt less than 6 in. wide	5
Belt sander, belt 6-10 in. wide	6
Belt sander, belt 10-14 in. wide	7
Drum sander, 24 in.	5
Drum sander, 30 in.	6
Drum sander, 36 in.	7
Drum sander, 48 in.	8
Drum sander, over 48 in.	10
Disc sander, 24 in. diam.	5
Disc sander, 26-36 in. diam.	6
Disc sander, 36-48 in. diam.	7
Arm sander	4

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. This is of considerable value where exhaust systems adapted to particular operations have been standardized by practice. Tables 1 and 2 present the duct sizes usually employed for standard wood-working machinery and for grinding and buffing wheels. Static pressures, which in practice have been found necessary to control and convey various materials, are given in Table 3. It must be remembered, however, that the *suction* is merely a rough

TABLE 2. SIZE OF CONNECTIONS FOR GRINDING AND BUFFING WHEELS

DIAMETER OF WHEELS	MAX. GRINDING SURFACE SQ IN.	MIN. DIAM. OF BRANCH PIPES IN INCHES
Grinding—		
6 in. or less, not over 1 in. thick	19	3
7 in. to 9 in., inclusive, not over 1½ in. thick	43	3½
10 in. to 16 in., " " " 2 in. "	101	4
17 in. to 19 in., " " " 3 in. "	180	4½
20 in. to 24 in., " " " 4 in. "	302	5
25 in. to 30 in., " " " 5 in. "	472	6
Buffing—		
6 in. or less, not over 1 in. thick	19	3½
7 in. to 12 in., inclusive, not over 1½ in. thick	57	4
13 in. to 16 in., " " " 2 in. "	101	4½
17 in. to 20 in., " " " 3 in. "	189	5
21 in. to 27 in., " " " 4 in. "	338	6
27 in. to 33 in., " " " 5 in. "	518	7

measure of the air volume handled and consequently of the air velocity at the opening of the hood. The elimination of any dusty condition requires added information concerning the shape, size and location of the hood used with regard to the operation in question.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. The static suction requirements, which range from 1½ to 5 in. water displacement in a *U*-tube, should be followed although in several instances they may appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm about the lower rim of the wheel.

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations they should not be less

TABLE 3. SUCTION PRESSURES REQUIRED AT HOODS

TYPE OF INSTALLATION	STATIC SUCTION IN INCHES OF WATER
Exhausting from grinding and buffing wheels	1½–5
Exhausting from tumbling barrels	2
Exhausting from wood-working machinery—light duty	2
Exhausting from wood-working machinery—heavy duty	2–4
Shoe machinery exhaust	2–3
Exhausting from rubber manufacturing processes	2
Flint grinding exhaust	2
Exhausting from pottery processes	2
Lead dust and fume exhaust	2–4
Fur and felt machinery exhaust	2–3
Exhausting from textile machinery	2–3
Exhausting from elevating and crushing machinery	2
Conveying bulky and heavy materials	3–5

than 200 fpm at the point of origin. For granite dust generated by pneumatic devices, Hatch et al[2] give velocities from 150 to 200 fpm, depending on the type of hood used, as sufficient for safe control. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given below.

HOODS

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dusts and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, the following formula may be used to determine the air velocity at any point along the axis[3]:

$$V = \frac{0.1\,Q}{x^2 + 0.1\,A} \tag{1}$$

where

V = velocity at point, feet per minute.
A = area of opening, square feet.
x = distance along axis, feet.
Q = volume of air handled, cubic feet per minute.

Velocity Contours

It is possible by use of a specially constructed Pitot-tube[4] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be expressed as percentages of the velocity at the hood opening and are purely functions of the shape of the hood[5].

[2]Control of the Silicosis Hazard in the Hard Rock Industries. I. A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools, by Theodore Hatch, Philip Drinker and Sarah P. Choate. (*Journal of Industrial Hygiene*, Vol. XII, No. 3, March, 1930).

[3]The Control of Industrial Dust, by J. M. DallaValle (*Mechanical Engineering*, Vol. 55, No. 10, October 1933).

[4]Studies in the Design of Local Exhaust Hoods, by J. M. DallaValle and Theodore Hatch (*A.S.M.E. Transactions*, Vol. 54, 1932).

[5]Velocity Characteristics of Hoods under Suction, by J. M. DallaValle (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 387).

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that when the velocity contour distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood with a side ratio of one-half. The distribution shown is identical for all openings with a similar side ratio provided the mapping is as shown in the figure. The contours, of course, are expressed as percentages of the velocity at the opening.

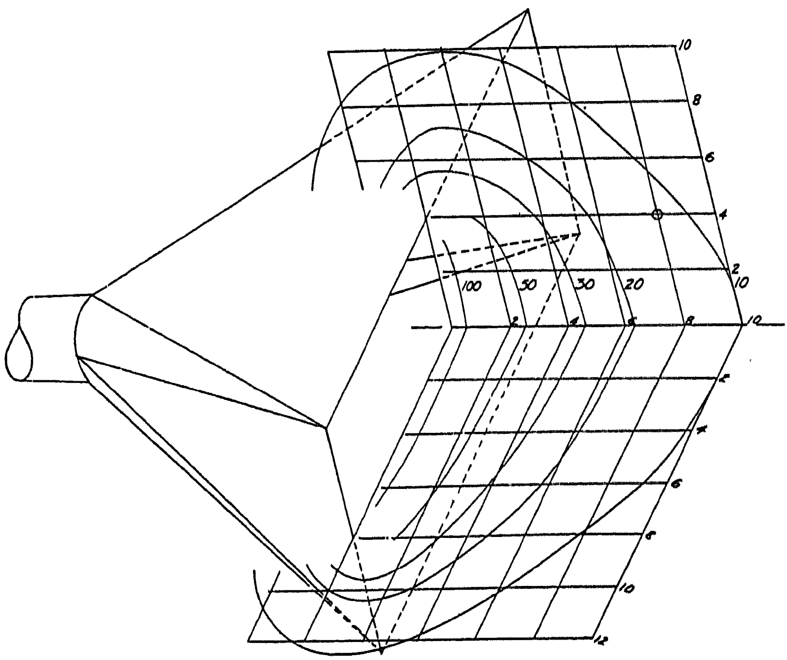


FIG. 1. VELOCITY CONTOURS FOR A RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

Air Flow from Static Readings

The volume of air flow through any hood may be determined from the following equation:

$$Q = 4005 f A \sqrt{h_t} \qquad (2)$$

where

Q = volume of air flow, cubic feet per minute.

A = area of connecting duct, square feet.

h_t = static suction at throat of hood, inches of water.

= orifice or restriction coefficient which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. The factor f is determined from the equation:

$$= \sqrt{\frac{h_v}{h_t}} \tag{3}$$

where h_v is the velocity head in the connecting duct.

The term *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 1 which shows that the velocity at any point along the axis varies inversely as the area of the opening and the square of the distance. However, this formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Large Open Hoods

Large hoods, such as are used for electroplating and pickling tanks, should be sub-divided so the area of the connecting duct is not less than one-fifteenth of the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distribution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12-in. elevation, and wherever possible they should have side and rear aprons so as to prevent short circuiting of air from spaces not directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the following simple formula:

$$Q = 1.4PDV \tag{4}$$

where

Q = total volume of air handled by hood, cubic feet per minute.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = air velocity desired along edges and surface of tank, feet per minute.

Lateral Exhaust Systems

The lateral exhaust method, as developed for chromium plating[6], is applicable in many instances in preference to the canopy type hoods. The method makes use of drawing air and fumes laterally across the top of vats or tanks into slotted ducts at the top and extending fully along one or more sides of the tanks. The slots are 2 in. wide and for effective

[6]Health Hazards in Chromium Plating, by J. J. Bloomfield and Wm. Blum (*U. S. Public Health Report*, Vol. 43, No. 26, September 7, 1928).

ticularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an

TABLE 4. GAGE OF SHEET METAL TO BE USED FOR VARIOUS DUCT DIAMETERS

DIAMETER OF DUCT	GAGE OF METAL
8 in. or less	24
9 to 18 in.	22
19 to 25 in.	20
26 in. or more	18

acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points where the material from one branch would tend to enter the branch on the opposite side of the main.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely.

The passing of pipes through fire-walls should be avoided wherever possible, and sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

TABLE 5. AIR SPEEDS IN DUCTS NECESSARY TO CONVEY VARIOUS MATERIALS

MATERIAL	AIR VELOCITIES (FPM)
Grain dust	2000
Wood chips and shavings	3000
Sawdust	2000
Jute dust	2000
Rubber dust	2000
Lint	1500
Metal dust (grindings)	2200
Lead dusts	5000
Brass turnings (fine)	4000
Fine coal	4000

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always necessary and is frequently done at the expense of a reduced air velocity, it is none the less advisable where future expansion of the exhaust system is contemplated.

Air Velocities in Ducts

When the static suction has been fixed for a given hood, the air velocity in the duct may be determined from Equation 2. Air velocities for conveying a material should be moderate. Table 5 gives the velocities generally employed for conveying various substances. Equations 5 and 5a may be used as tests to determine the conveying efficiency of a system[8]. Velocities determined from these formulae should be increased by at least 25 per cent since they represent the minimum at which a stated size and density of material can be transported.

For vertical ducts:

$$V = 13{,}300 \frac{s}{s+1} d^{0.57} \qquad (5)$$

For horizontal ducts:

$$V = 6000 \frac{s}{s+1} d^{0.40} \qquad (5a)$$

where

V = air velocity in duct, feet per minute.
s = specific gravity of particles.
d = average diameter of largest particles conveyed, inches.

Example 1. Granular material, the largest size of which is approximately 0.37 in. in diameter, with a specific gravity of 1.40 is to be conveyed in a vertical pipe, the velocity of the air in which is 4100 fpm; find whether the material can be transported at this velocity.

Substitute data in Equation 5a and multiply by 1.25:

$$V = 1.25 \times 13{,}300 \times \frac{1.4}{2.4} \times 0.37^{0.57}$$

Antilog ($0.57 \times \log 0.37$) = 0.568; the required velocity is, therefore, 5500 fpm.

[8]Determining Minimum Air Velocities for Exhaust Systems, by J. M. DallaValle (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1932, p. 639).

TABLE 6. LOSS THROUGH 90-DEG ELBOWS

ELBOW CENTER LINE RADIUS IN PER CENT OF PIPE DIAMETER	LOSS IN PER CENT OF VELOCITY HEAD
50	75
100	26
150	17
200 to 300	14

Hence, the duct velocity must be increased either by speeding up the fan or decreasing the diameter of the duct, or both.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 2, Chapter 31. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 50 to 300 per cent of pipe diameter, the loss may be estimated from Table 6. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a throat radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a throat diameter radius 1½ times the diameter, the resistance is about the same as that of seven diameters of straight pipe.

COLLECTORS

The most common method of separating the dust and other materials from the air is to pass the mixture through a centrifugal or *cyclone* collector. In this type of collector the mixture of the air and material is introduced on a tangent, near the cylindrical top of the collector, and the whirling motion sets up a centrifugal action causing the comparatively heavy materials suspended in the air to be thrown against the side of the separator, from which position they spiral down to the tail piece, while the air escapes through the stack at the center of the collector.

The diameter of the cyclone should be at least 3½ times the diameter of the fan discharge duct. When two or more separate ducts enter a cyclone, gates should be provided to prevent any back draft through a system which may not be operating. Cyclones working in conjunction with two or more fans should be designed to operate efficiently at two-thirds capacity rating. The following formula is useful in computing the loss through a cyclone when the velocity of the air in the fan discharge duct is known:

$$h_c = 0.13 \left(\frac{V}{1000}\right)^2 \tag{6}$$

where

h_c = the pressure drop through the cyclone, inches of water.
V = the air velocity in the fan discharge duct, feet per minute.

If a cyclone is used to collect light dusts such as buffing wheel dusts,

feathers and lint, the exhaust vent should be large enough to permit an air velocity of 200 to 500 fpm. This will, of course, require a cyclone of larger dimensions than given for the foregoing general case.

When a high collection efficiency is desired, or the material is very fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation. The capacities and characteristics of this type of separator should be obtained from the manufacturers.

Cloth Filters

Filters are used when the material collected by an exhaust system is valuable or cannot be separated efficiently from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. *Bag houses* used in the manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The time increase of resistance varies with the amount of material permitted to build up on the surface of the filter and can be determined only by experiment. The limits of the increase may be regulated by adjustment of the shaking or cleaning mechanism. These limits may be regulated further according to the capacity of the fan and the effective performance of the hoods and the duct system.

For additional information on Dust and Cinders, see Chapter 28, Air Cleaning Devices.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the pipes.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. The collector drop in inches of water is given approximately by Equation 6, but where possible the resistance of the particular collector to be used should be ascertained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. Find the velocities in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in reducing the concentration of dusts, fumes, vapors and gases below the safe or threshold limits[9].

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level. Commonly accepted values of threshold limits for the usual gases and vapors are given in Table 7.

SELECTION OF FANS AND MOTORS

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. For particular features concerning special fans, consult the *Catalog Data Section* of The Guide and manufacturers' data. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a collector on the negative side of the fan or by lining the housing and blades with rubber.

Table 7. Threshold Limits of Common Vapors and Gases[a]

Substance	Spec. Grav. of Gas or Vapor (Air 1)	Inflammable Limits (%)	Physiological Action	Maximum Allowable Concentration (ppm)
Chlorine	2.486	non-inflamm.	irritant	0.35
Ozone	5.5	do	do	0.80
Hydrogen chloride	1.2678	do	do	10.0
Sulphur dioxide	2.2638	do	do	10.0
Carbon monoxide	0.9671	12.5–74	asphyxiant	100.0
Hydrogen sulphide	1.190	4.3–46	do	85–130
Benzene	2.73	1.4–7.0	anesthetic	100.0
Methanol	1.1	7.5–26.5	do	100.0
Carbon tetrachloride	5.3	non-inflamm.	do	100.0

[a]The Prevention of Occupational Diseases, by R. R. Sayers and J. M. DallaValle (*Mechanical Engineering*, Vol. 57, No. 4, April, 1935).

If no future expansion of an exhaust system is contemplated, the fan motor should be chosen to provide the calculated air volume. Should, however, the exhaust system be required to handle more air in the future, the motor should be adequate for the maximum load anticipated. Further information regarding the choice of fans and motors is given in Chapters 29 and 35.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to

[9]Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 353).

TABLE 8. MATERIALS TO BE USED FOR THE PROTECTION OF EXHAUST SYSTEMS AGAINST CORROSION[a]

TYPE OF FUME CONVEYED	PROTECTIVE MATERIAL TO BE USED
Chlorine	Rubber lining or chrome-nickel alloys
Hydrogen sulphide	Aluminum coated iron, aluminum, high chrome-nickel alloys
Ammonia	Iron or steel
Sulphurous gases	High chrome-nickel alloys
Hydrochloric acid	Rubber lining, chrome-nickel alloys
Nitrous gases	Nickel-chrome alloys

[a]Condensed from data given by Chilton and Huey (*Industrial and Engineering Chemistry*, Vol. 24, 1932).

chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 8. Hoods and ducts when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

Chapter 40

DRYING SYSTEMS

Drying Methods, Driers, Mechanism of Drying, Moisture, General Rules for Drying, Equipment, Humidity Chart, Combustion, Design, Estimating Methods

DRYING, in its broader sense, refers to the removal of water, or other volatile liquid from either a gaseous, liquid, or solid material. In practice, the process of direct drying gaseous material is referred to generally as dehumidifying, or condensing, and in some cases chemicals are used in the adsorption or absorption of moisture. Drying a liquid is called evaporation or distillation. The common usage of the word *drying* refers to the removal of water or other liquid, such as a solvent, by evaporation from a solid material.

When the solid to be dried contains large amounts of free water, the actual drying process is frequently preceded by the removal of part of the water by some mechanical means, such as filtration, settling, pressing or centrifuging. Removal of as much water as possible by such methods is usually advisable, as the cost of these operations, per pound of water removed, is generally much less than by evaporation.

DRYING METHODS

Drying may be accomplished in any one or combination of the following methods:

1. Radiation.
2. Conduction, or direct contact.
3. Convection.

Radiation

The source of heat for radiation may be either the sun, or heated surfaces. Sun drying is practiced where danger from rain is slight, and where sufficient time can be allowed. Where a strict adherence to a schedule is necessary, or where dusty atmosphere is present, this method is not in favor. Fruits are often dried in the sun.

Radiation from hot surfaces (heated by steam, electricity, or other means) furnishes generally, from one-third to one-half the total heat required for evaporation. Convection currents set up by these hot surfaces and the cooler materials carry the balance of the heat.

TABLE 1. DRIERS FOR EVAPORATION OF WATER

Type	Kind	Materials Handled	Means of Handling	Temp. Range Deg F	Heat Supply	Uses and Remarks
Batch or Intermittent	Com-partment	Paper, Leather, Yarns, Lumber, Foodstuffs	Suspended, Truck, Tray	80 to 180	Steam Coils, Air, Electricity	When production does not warrant continuous drier
	Agitated	Chemicals too sticky for Rotary Drier	Shoveled into Drum or Pan	100 to 330	Water, Steam Jacketed, may have Vacuum on top	Where dust must be saved
	Vacuum	Chemicals, Explosives, Pharmaceuticals, Food Products	Tray, Basket, Tumbling Drum	80 to 300	Water, Steam	Cost of operation high, for expensive materials
Continuous	Tunnel	Ceramics, Chemicals, Lumber, Food Products	Truck, Tray, Belt	100 to 350	Steam Coils, Air, Electricity, Products of Combustion	For high production
	Rotary	Bulk	Cascades through	80 to 500	Air, Steam, Products of Combustion	Where material will stand rough handling and is not subject to balling up
	Drum	Liquids, Slurries	Flowed on Drum, Dry Material Scraped off	to 310	Steam, may have Vacuum on Top	Hygroscopic materials dried with vacuum, and packed immediately
	Cylinder	Paper, Textiles, Chemicals	Continuous Sheets, Endless Chain Belt	to 350	Steam inside of Drum	Where material comes in sheets or rolls, and will stand direct contact with heating surface
	Festoon	Paper, Chemicals	Continuous Sheets, Suspended on Metal Screens	to 200	Air, Steam Coils	Where one side cannot come in contact with supports until dry
	Tower or Column	Grains, Sand	Falls through by Gravity	125 to 250	Air, Steam Coils	Where headroom is available
	Spray	Solutions over 30% Solids	Sprayed into Chamber	120 to 350	Air, Products of Combustion	Drying is almost instantaneous
	Induction	Metals, for removal of traces of Water	Placed in High Frequency Field	to 400	Electricity	Where heating of metal from inside out is important

Conduction or Direct Contact Drying

This method of drying is advantageous where the material can be flowed on to the drying surface and the dried material scraped off, or where the material to be dried can be handled in a sheet, and where there is no danger of subjecting the product to the full temperature of the heating medium. The source of heat for this method may be steam, electricity, hot oil or hot water.

Convection

The circulation of heated air or other gases about the material to be dried is generally termed convection drying. The convection may be either natural or forced. With forced circulation, the temperature of the

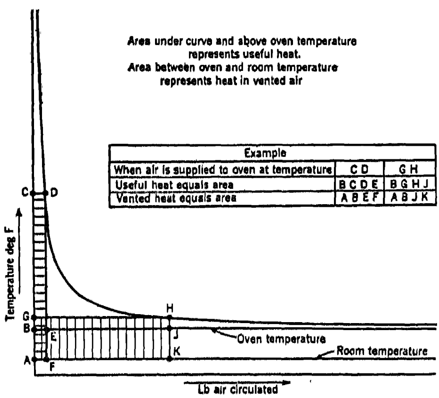


FIG. 1. RELATION BETWEEN USEFUL AND TOTAL HEAT SUPPLIED

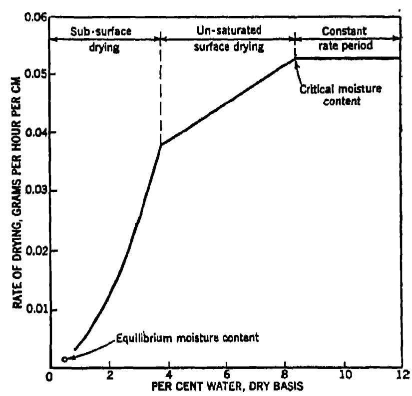


FIG. 2. RATE OF DRYING OF WHITING SLAB

drier is more uniform and the rate of drying is much higher than with natural circulation. Where humidity is used, the control is much easier, and more accurate.

The source of heat for a convection drier may be steam, electricity, hot water, oil-fired heater, gas-fired heater, or coal furnace. Where either oil, gas or coal is used, the type of heater may be direct or indirect; *i.e.*, the products of combustion may be used (direct), or the circulated air may be heated through an interchanger (indirect).

Where the direct type is used, there is naturally a higher thermal efficiency, but it can only be used where the odor, soot, or the chemical elements of the products of combustion do not affect the material being dried. When heat economy is an important consideration this method (Fig. 1) may be used, permitting a small amount of air to be circulated, if a sacrifice of accurate control of temperature and humidity can be justified.

DRIERS

The term *adiabatic drier* is applied to a drier in which all the heat is supplied by air externally heated. The temperature of the air in the

intermediate state of unsaturated surface drying does not occur and the drying is of the sub-surface type during practically the whole of the falling rate period. With other kinds of material, particularly thin sheets, such as newsprint paper, sub-surface drying may occur at such a low moisture content that it is not encountered in commercial work, the falling rate period being confined solely in practice, to unsaturated surface drying.

TABLE 2. FACTORS INFLUENCING DRYING

Factor	Drying Period	
	Constant Rate, Unsaturated Surface	Sub-Surface
Temperature	Increase in temperature increases drying rate	Increase in temperature increases drying rate, because with decreased viscosity, diffusion increases
Humidity	Drying rate increases as humidity is decreased	No effect until equilibrium content is reached; drying then ceases
Air Velocity	Drying rate varies approximately as the 0.6 power of the velocity	No effect
Air direction	Drying rate increases, the more nearly the air blows perpendicular to surface; for dead air film becomes thinner	No effect
Thickness of Material	Drying rate is not affected by the thickness	Drying rate varies inversely as the square of the thickness

MOISTURE

Moisture in the solid may be in either of two forms:

1. Capillary or free.
2. Hygroscopic or chemically combined.

Free moisture is contained in the capillary spaces between the particles or fibers of the materials. The loss of this moisture changes only the weight of the material. *Chemically combined* or hygroscopic moisture is intimately associated with the physical nature of the material and its removal changes both the physical characteristics as well as the chemical properties. The amount of hygroscopic moisture a material can contain is limited. This limit is called the *fiber saturation point.* When material is dried below this point, care must be exercised to avoid physical changes in the material, such as shrinkage, hardening, etc. All hygroscopic materials have definite equilibrium moisture contents dependent on temperature and humidity. Materials are frequently dried to a lower moisture content than those of equilibrium conditions in use, and allowed to regain the necessary moisture after leaving the drier to equalize the moisture in the material. Fig. 3[1] shows the equilibrium moisture content of wood.

[1]*U. S. Department of Agriculture Bulletin*, No. 1136.

GENERAL RULES FOR DRYING

Temperature

The highest temperature possible should be used because of faster drying and smaller requirements for ventilation. The amount of moisture that can be carried by a pound of air increases rapidly with rise in temperature as shown in the humidity chart of Fig. 4. Too high a temperature may cause spoilage of materials; many materials calcine or change their chemical properties if heated too hot; gypsum and glauber salts lose some of the chemically combined water, fall apart, and change their chemical properties. Too high or rapid rise in temperatures in drying lumber or ceramics may create a liquid vapor tension within the material so high that the cells explode, causing permanent injury to the fiber. If too high a temperature is used on some chemicals, they begin to react

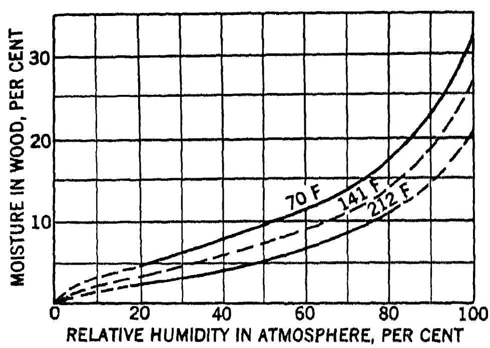


FIG. 3. RELATION OF EQUILIBRIUM MOISTURE CONTENT IN WOOD TO THE RELATIVE HUMIDITY OF SURROUNDING AIR

exothermally; a temperature rise and chemical action from within will burn the materials, *e.g.*, bakelite products, gunpowder, etc. During the constant rate period of drying, the material heats only to the wet-bulb temperature of the surrounding air, consequently high temperatures will not injure the material in this stage.

Humidity

Moisture in the drying air may be very important. Many materials tend to case-harden, dry on the outside, forming a skin which retards the moisture flow from the inside to the surface, or stops it completely, and so increases the drying time very much or causes a change of the physical properties of the material. It is often necessary to add humidity to the air in the initial stage of drying. Lumber case-hardens, cracks, and warps if the outside is dried too fast. Ceramics crack if not heated through before drying commences. Elastic materials warp while others crack if not evenly dried. Many paints case-harden if not dried under high humidity.

On the other hand, in the case of those materials whose physical or chemical properties require that they be dried at relatively low temperatures high humidity tends to retard drying in the first stage and may even stop it altogether in the final stage. Where drying temperatures

below 120 to 140 F are used the drying rate may be highly dependent on atmospheric humidity conditions. In such instances it is often desirable to dehumidify the air entering the drier during periods of high atmospheric humidity; where a high degree of uniformity is required it is often possible to secure complete independence of atmospheric conditions by recirculating the air in a closed system which includes a suitable dehumidifier. For this purpose absorptive dehumidifying systems have the advantage of accomplishing the desired reduction of humidity without appreciably elevating or lowering the dry-bulb temperature of the air; for this reason after-cooling is not required, and reheating is reduced to a minimum. Complete descriptions of such dehumidifying systems are given in Chapter 23 on Cooling and Dehumidification Methods.

Air Circulation

As noted under Mechanism of Drying, air velocity is more important in the first two stages of drying than in the last, and for this reason zone drying in continuous driers is frequently considered. It permits accurate regulation of temperature, humidity, and velocity in the different zones. High velocity results in more rapid drying, more even distribution of temperature and consequently more even drying in the first period. Too high a velocity may be detrimental because of excessive power needed for creating it, or because the material may blow away if it is light and fluffy. In the drying of paints, varnishes, and enamels, high velocity or improper distribution of the air even with the use of filters, may cause dust already in the drier, to be blown against the material, ruining the finish. Table 3 presents data on drying of various materials.

EQUIPMENT FOR DRYING

Equipment for drying may be divided into the following classes:

1. Heat and humidity supply.
2. Methods of handling.
3. Ovens.

The heat and humidity supply for low temperature work up to 250 F is often steam; steam coils either in the oven or outside, heat the air used for drying. Circulation of heated oil is used to a limited extent, but the danger of leaks is serious, for if the oil is hotter than the flash point, a fire may start if the oil is released to the atmosphere. In many cases where steam is not available, direct or indirect-fired heaters are used with gas or oil as fuel. Indirect heaters should be carefully selected from a standpoint of long life and efficiency. The heat exchange surface should be adequate in area and easily accessible for cleaning and removal. For extremely high temperatures, alloy surface may be used. With direct-fired equipment care must be used in the selection of burners and sufficient combustion space allowed to insure complete combustion of fuel. Humidity can be obtained in driers by the use of steam spray, humidifiers, or recirculation.

Methods of *handling of material* have been indicated in Table 1.

For low temperature work up to 200 F ovens and driers are commonly built of two thicknesses of insulating board (fireproof preferred), with air space between. As the temperature increases materials better able to

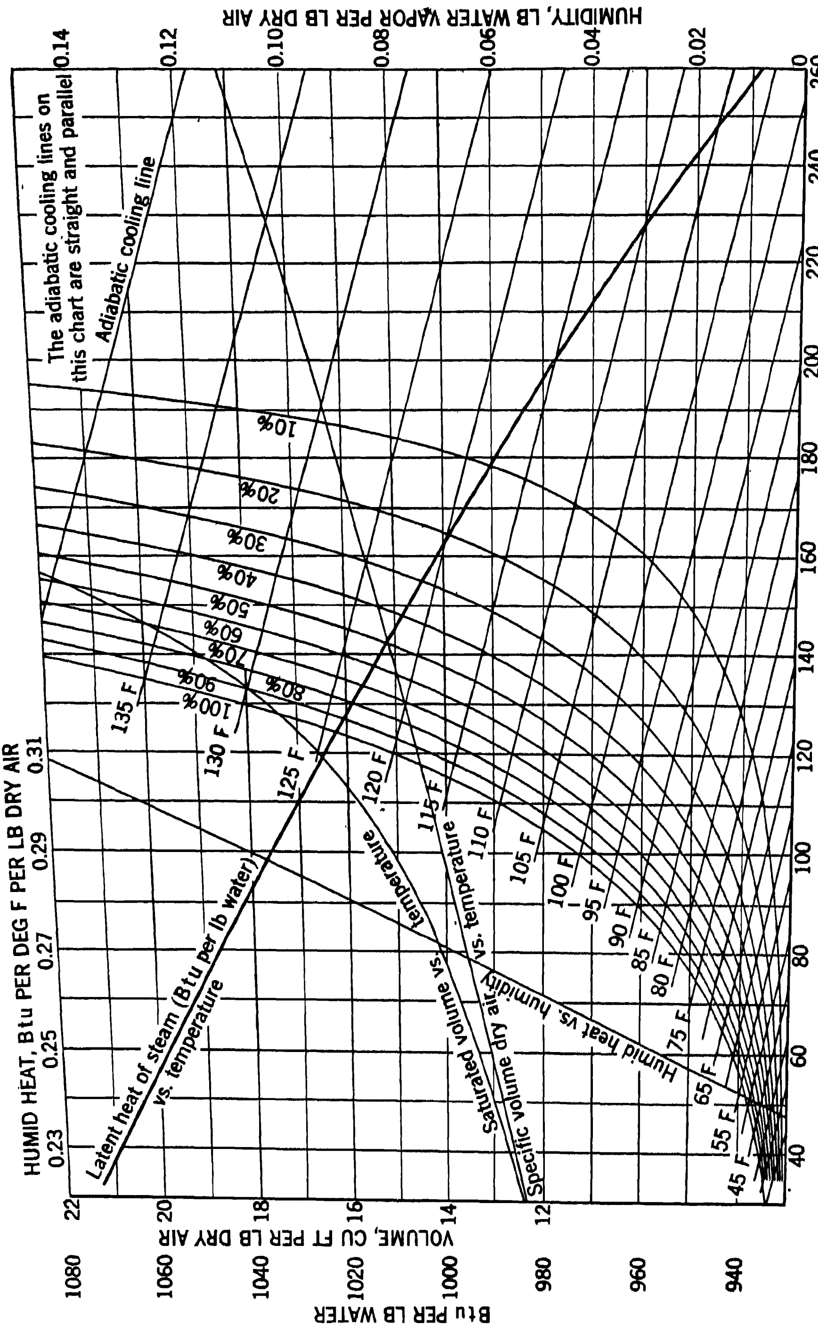


Fig. 4. Humidity Chart

withstand the heat must be used. Metal lined ovens are easy to keep clean, and many high temperature driers up to 1000 F are made of metal panels with insulation between. Care should be taken to avoid *through metal* (metal extending through the oven from inside to out). Batch type ovens are entirely closed while in use and control of air leakage is easily taken care of. In the continuous drier where the ends are open, heat and air leakage becomes important. Warm air leaking out of the ends of ovens means a heat loss, and often the temperature and humidity outside the oven becomes unbearable. For this reason, inclined or bottom entry ovens are used, as the warm air leakage can be more easily controlled. See Figs. 5 and 6.

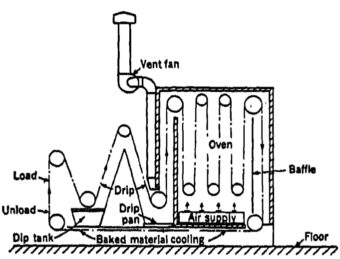


FIG. 5. SMALL PART MULTIPLE PASS OVEN

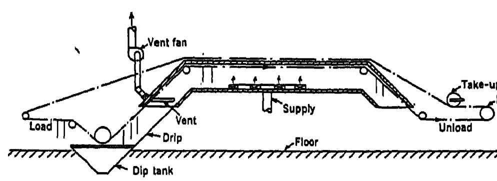


FIG. 6. INCLINED END ENAMELING OVEN

HUMIDITY CHART FOR DRYING WORK

In drying problems the chemical engineer uses different psychrometric values than those used by the heating, ventilating and air conditioning engineer. The humidity chart illustrated in Fig. 4 is based upon values determined from the following explanations:

Humidity (H) is the number of pounds of water vapor carried by one pound of dry air.

Percentage Humidity ($\% H$) is the number of pounds of water vapor carried by one pound of dry air at a definite temperature, divided by the number of pounds of vapor that one pound of dry air would carry if it were completely saturated at the same temperature.

Per Cent Relative Humidity (Φ) is the ratio of weight of water vapor contained in any given volume of air, to the weight of water vapor present in the same volume of saturated air, all values referring to the same temperature.

To convert from one relation to the other,

$$\% H = \frac{29.92 - p_s}{29.92 - p} \times \Phi \qquad (1)$$

where

p_s = vapor pressure of water, inches mercury; at dry-bulb temperature, degrees Fahrenheit.

$p = \Phi\, p_s$.

COMBUSTION

Where products of combustion are used directly in the oven, a knowledge of their formation and heat values is important. The properties of

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Apples	140–180		6 Hrs
Armatures Varnish	200		2.5 Hrs
Banana Food ¼ in. Thick	140		4–6 Hrs
Barrels	300		15 Min
Beans	140		18 Hrs
Bedding	150–190		
Blankets	120		40 Min
Brake Lining	325		12 Min
Brick continuous	350 to 90		24 Hrs
Briquets	1100		108 Min
Cabbage Raw	150		4.5 Hrs
Candied Peel	165		2 Hrs
Casein	180		5 Hrs
Cereals	110–150		
Ceramics before firing	150	70 to 20	24 Hrs
Chicle	95–100		
Coco-fiber mats	170–210		10 Hrs
Cocoanut	150–200		4–6 Hrs
Coffee	160–180		24 Hrs
Conduit (Enamel)	400 Max		2 Hrs
Cores, Oil sand for molding ½—1 in. thick	300		30 Min
Black sand with goulic binder about 0.6 of time 3 in. thick	480		2.5 Hrs
Black sand with goulic binder about 0.6 of time 8 in. thick	480		4.5 Hrs
Black sand with goulic binder about 0.6 of time 16 in. thick	700		10 Hrs
Cores, Crank case (in continuous ovens)	525–600		2–3 Hrs
Cores, Radiator (in continuous ovens)	275–450		1.5 Hrs
Cornstalk Board	150		2 Hrs
Cotton Linters	180		
Enamels synthetic			
Finish coat on autos	225		2 Hrs + Air Dry
Ice boxes all metal (white)	290–425		1 Hr
Ice boxes wood inside (white)	225		3 Hrs
Enamel not synthetic			
Fence posts green	200		1 Hr
Golf balls (white)	90–95	40–50	18–36 Hrs
Small parts (auto) black	450		1 Hr
Steel furniture	225–300		30–350 Min
License plates	250		1.5 Hrs
Feathers	150–180		
Films, Photographic	85–110		20–30 Min
Fruits and Vegetables	140		2–6 Hrs
Furs	110		
Gelatin	110		
Glue bone, thin sheets on wire trays	70–90		6–9 Days
Glue skin	70–90		2 Days
Glue size on furniture	130		4 Hrs
Gut	150		
Gypsum board ⅜ in. thick Start Wet	350		60 Min
Gypsum board ⅜ in. thick Finish	275		
Gypsum block	350–190		8–16 Hrs
Hair felt	180–200		
Hair goods	150–190		1 Hr
Hanks on poles	120		2 Hrs
Hats felt	140–180		
Hides thin leather	90		2–4 Hrs
Hides heavy	70–90		4–6 Days

[a]See references at end of chapter.

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]—Con.

MATERIAL	TEMPERATURE DEG F	PER CENT RELATIVE HUMIDITY	DRYING TIME
Hops	120–180		
Ink printing	70–300		
Japan beds	300		1.5–2 Hrs
Japan cash register	300–450		1.5 Hrs
Japan metal shelving	200		30 Min
Knitted fabrics	140–180		
Leather mulling	78–95	85	
Leather thick sole	90	70	4 Days
Leather uppers	80		2–3 Days
Linoleum varnish	110–145	10–30	6–10 Hrs
Lithographing on tin color work	250–270		18–25 Min
Lithographing on tin Japan	350		
Lumber green hardwood	100–180		3–180 Days
Lumber green soft wood	160–220		2–14 Days
Macaroni	90–110		24–48 Hrs
Matches	140–180		
Matrix	350		15 Min
Milk and other liquid foods spray dried	135–300		Instantaneous
Millboard sheets	95		10 Hrs
Moulds green sand C.I. flasks (one surface only exposed) 8 in. thick	600		6 Hrs
Moulds green sand C.I. flasks (one surface only exposed) 13 in. thick	700		13 Hrs
Motors, field coils	180		6 Hrs
Motors, stators	250		6.5 Hrs
Noodles	90–95		
Nuts	75–140		24 Hrs
Oil cloth	150		
Paint, wood wheels	150	35	8–24 Hrs
Paint, on sheet metal	135–140	22–30	2.5 Hrs
Paper, machine dried	180		
Paper, air dried	90–200		
Paper wall, ground coat	140		3 Min
Paper wall, varnished	140–160	45	15 Min
Paper cardboard, spirit varnish	150		1–2 Min
Peaches	135		26 Hrs
Pears	140		24 Hrs
Peas	150		6 Hrs
Potatoes sliced	85		4 Hrs
Potatoes steamed	170		6.5 Hrs
Prunes	140		
Rags	180		
Ramie fiber	140		10 Hrs
Rice	150		
Rock wool insulation	300		8 Hrs
Rubber	85–90		6–12 Hrs
Rubber reclaimed	140–200		1–2 Hrs
Rugs	190		4–8 Hrs
Salt	350		Rotary Drier
Sand loose 1 in. deep	300		10–15 Min
Sausage casings	110		5 Hrs
Shade cloth	240		1–2 Hrs
Shirts	120		20 Min
Soap	100–125		12–72 Hrs
Starch	180–200		1–4 Hrs
Stock feed mixed	180–220		20–30 Min
Storage battery plates	100–110	90 for	24 Hrs
	250	Low for	6 Hrs
Sugar	150–200		20–30 Min

[a]See references at end of chapter.

TABLE 3. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS[a]—Con.

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Tanin and other chemicals (spray dried)	250–300		Instantaneous
Terra Cotta (air drying in conditioned room)	150–200		12–96 Hrs
Tobacco leaves	85–130		12 Hrs
Tobacco stems	180–200		12 Hrs
Varnish refrigerator boxes	110	35	5–7 Hrs
Varnish steering wheels	110–140	25–35	Overnight
Veneer ¼ in. 3-ply	120–130	35–40	6–8 Hrs + 2 Hrs acclimation
13/16 in. 5-ply	120–130	35–40	16–18 Hrs + 4 Hrs acclimation
1¼ in. 5-ply	120–130	35–40	20–24 Hrs + 4 Hrs acclimation
Vitreous Enamel sheets before firing	170		
Wallboard pasted plywood	300		15–20 Min
Wallboard fiber insulating, roller type drier	300–385		2½–3 Hrs
Wallboard fiber insulating, truck type drier	300–385		24–48 Hrs
Walnuts	100		24 Hrs
Wheat, corn, oats, rice, barley	180		
Wire cloth Japan	200		20 Min
Wool	105		

[a]See references at end of chapter.

the common constituents of fuel are shown in Table 4. The heating values of oils are shown in Fig. 7. The sensible heat in Btu contained in the products of combustion of an average fuel oil and various gases is given in Fig. 8. The problem of securing complete combustion in a heater is important, in order to secure efficiency and the absence of soot formation, but unlike the ordinary power or heating boiler, excess air need not be maintained at a minimum in most cases. Excess air is generally admitted either in the heater or before the products go into the drier.

DESIGN

In all drying problems, data regarding temperatures, time, and humidity must be obtained by experiment or previous experience. Experiments are best performed at the temperatures, humidities, and velocities to be actually used in the full sized drier, and with full size samples.

The following nomenclature and explanation of terms will be used in the discussion of drying calculations:

H = humidity of air, pounds of water vapor per pound of dry air.
G = pounds of dry air supplied to the drier per unit of time.
S = pounds of stock dried per unit of time in a continuous drier.
S^1 = pounds of stock charged per batch to a discontinuous drier.
Θ = time.
Q = total heat supplied to the drier.

t = air temperature.
t' = stock temperature.
t'' = average stock temperature over short time interval, in a batch drier.
t_w = wet-bulb temperature.
s' = specific heat of the stock.
B = total radiation and conduction losses per unit time.
w = pounds of water per pound of dry stock.
r = heat of evaporation of water.
s = humid heat of air, *i.e.*, heat necessary to raise 1 lb of dry air + H lb of steam 1 F.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters and (2) where it leaves the drier.

Air driers may be divided into two classes, those in which *all moisture* evaporated from the stock *leaves the drier as vapor* in the effluent air, and those in which *part or all* of the moisture *is condensed* from the air *in the drying equipment itself*. In any continuously operating drier of the first type the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S(w_1 - w_2) \tag{2}$$

TABLE 4. GAS COMBUSTION CONSTANTS[a]

Gas	Chemical Formula	Molecular Weight	Cu Ft per Lb	Heat of Combustion, Btu per Lb, Gross	Heat of Combustion, Btu per Lb, Net	Lbs per Lb of Combustible, Required for Combustion, O_2	Required for Combustion, N_2	Required for Combustion, Air	Flue Products, CO_2	Flue Products, H_2O	Flue Products, N_2
Carbon	C	12.000	....	14,140	14.140	2.667	8.873	11.540	3.667	....	8.873
Hydrogen	H_2	2.015	187.723	61,100	51,643	7.939	26.414	34.353	....	8.939	26.414
Oxygen	O_2	32.000	11.819	....	....	....	....	....	....	....	....
Nitrogen	N_2	28.016	13.443	....	....	....	....	....	....	....	....
Carbon Monoxide	CO	28.000	13.506	4,369	4,369	0.571	1.900	2.471	1.571	....	1.900
Carbon Dioxide	CO_2	44.000	8.548	....	....	....	....	....	....	....	....
Methane	CH_4	16.031	23.565	23,912	21,533	3.992	13.282	17.274	2.745	2.248	13.282
Ethane	C_2H_6	30.046	12.455	22,215	20,312	3.728	12.404	16.132	2.929	1.799	12.404
Propane	C_3H_8	44.062	8.365	21,564	19,834	3.631	12.081	15.712	2.996	1.635	12.081
Sulphur Dioxide	SO_2	64.060	5.770	....	....	....	....	....	....	....	....
Water Vapor	H_2O	18.015	21.017	....	....	....	....	....	....	....	....
Air	....	28.900	13.063	....	....	....	....	....	....	....	....

[a]All gas volumes corrected to 60 F and 30 in. mercury barometric pressure dry.

into contact with the stock with sufficient intimacy so that the air leaving the drier is saturated, or nearly so. Counter-current as against parallel flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (G), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases, provided the increase in moisture carrying capacity of the air,

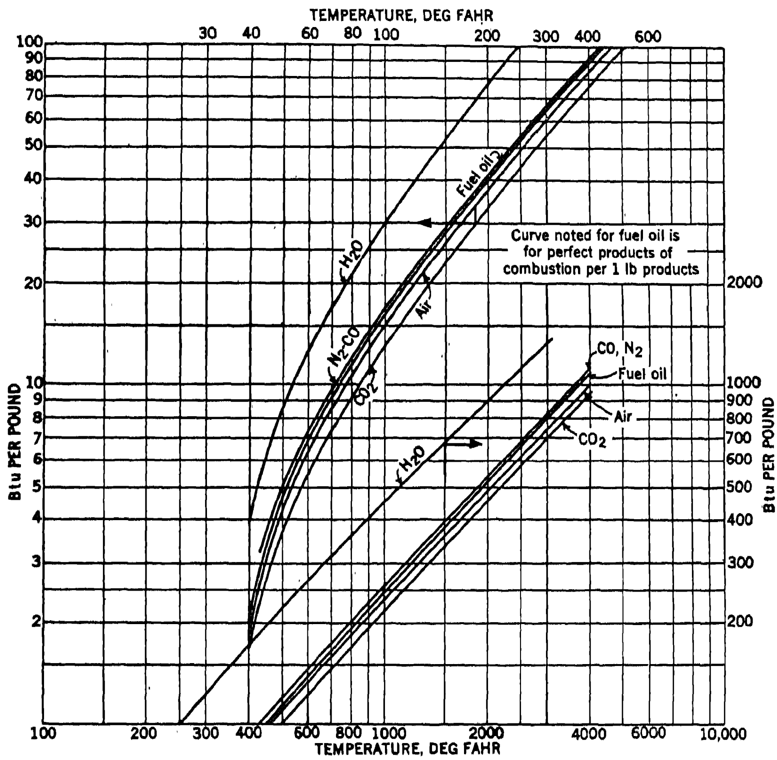


FIG. 8. HEAT CONTENT OF GASES ABOVE 32 F IN BTU PER POUND

due to high temperature, is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air is imperative.

Ventilation Phase

The technique of attack of the *ventilation phase* of a drying problem is best made clear by an illustration. Assume that a material containing 40 per cent moisture is to be dried until this quantity of moisture is reduced to 5 per cent by weight. The material will stand an air temperature of 150 F and it is possible to provide sufficiently good contact between the material and the drying air so that the effluent air can be

brought up to 50 per cent humidity at 150 F. The drier is to use room air, the temperature and humidity of which may be assumed to average 70 F and 50 per cent. A counter-current drier will be employed and the air in this drier will be kept at a substantially constant temperature of 150 F by heaters thermostatically controlled. The stock enters at 70 F, rises quickly to the wet-bulb temperature of the air, with which it is in

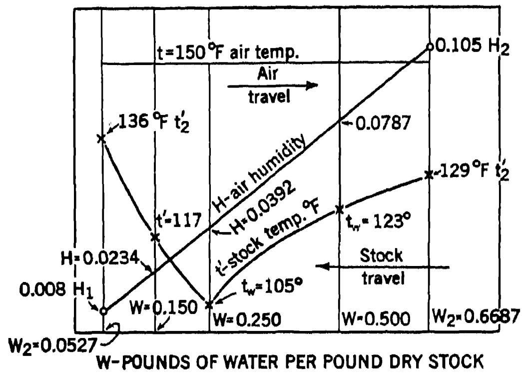


FIG. 9. TEMPERATURE HUMIDITY RELATIONS IN A DRIER

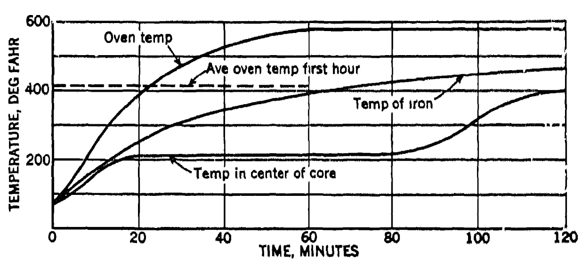


FIG. 10. CORE DRYING TIME TEMPERATURE RELATIONS

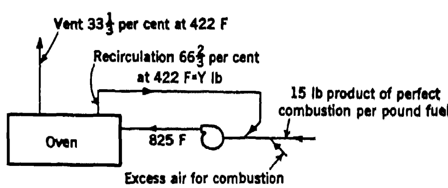


FIG. 11. CORE DRYING DIAGRAM OF COMBUSTION PRODUCTS AND AIR

contact, and is found experimentally to maintain wet-bulb temperature until the moisture content has fallen to 20 per cent. From this point its temperature rises progressively as it dries. In this range the difference in temperature between stock and air, divided by the wet-bulb depression, may be assumed proportional to the moisture content.

The moisture content of the entering stock, in the units here employed, is:

$$w_1 = \frac{40 \text{ per cent water}}{60 \text{ per cent dry stock}} = 0.6667:\ w_2 = \frac{5 \text{ per cent water}}{95 \text{ per cent dry stock}} = 0.0527$$

$w_1 - w_2 = \Delta w = 0.614$ lb water evaporated per pound of dry stock. Since the air leaving the drier is 50 per cent saturated at 150 F from Fig. 4, $H_2 = 0.105$. Similarly, $H_1 = 0.008$, corresponding to 50 per cent humidity at 70 F. Consequently $H_2 - H_1 = \Delta H = 0.097$ lb water evaporated per pound dry air.

Inspection of Equation 2 shows that (H) is linear in w. Hence, one can construct on Fig. 9, the line marked (H) being drawn connecting the initial and final points just computed.

Since the air leaving the drier has a temperature of 150 F and a humidity of 0.105, Fig. 4 shows that its wet-bulb temperature is 129 F. This is plotted at the right hand side of Fig. 9. Since the stock maintains a wet-bulb temperature down to 20 per cent moisture, where $w = 0.25$, the corresponding humidity can be computed by the use of Equation 2 or by reading directly from the diagram, the value being 0.0392. Fig. 4 shows that the corresponding wet-bulb temperature is 105 F. Any intermediate point on the wet-bulb temperature curve can be calculated similarly. The points for $w = 0.5$ are shown in Fig. 9.

Below the point, $w = 0.25$, the temperature of the stock begins to rise appreciably above the wet-bulb temperature. Its temperature at any given point in this range, for example at $w = 0.15$, may be computed as follows: At this point, $H = 0.0234$ (from Equation 2) and from Fig. 4, $t_w = 95$ F. Hence the wet-bulb depression, $t - t_w = 150 - 95 = 55$ F. The assumption made regarding the relation between stock temperature and moisture content in this range may be formulated:

$$\frac{\Delta t^1}{t - t_w} = \frac{w}{0.25}$$

At the point $w = 0.15$, $\Delta t^1 = 33$ F, $t^1 = 117$ F. The temperature of the stock leaving the drier, similarly computed, is 136 F.

Fig. 9 thus computed gives in graphical form the information as to the temperature humidity relationships in the drier. The air requirements can be computed by Equation 2. Thus, per 100 lb of dry stock, it is necessary to supply 633 lb of dry air. Furthermore, since from Fig. 4 it is seen that the volume of 50 per cent saturated air at 70 F, is 13.55 cu ft per pound; 8580 cu ft of room air must be supplied per 100 lb dry stock. Similarly, since the volume of 50 per cent saturated air at 150 F is 18.0 cu ft per pound, the volume of hot wet air discharged from the drier is 11,400 cu ft per 100 lb of dry stock. Finally, the heat necessary to supply to the drier, as a whole, or to any section of it, may be computed from Equation 3.

High Temperature Drier

In the design of a high temperature drier unit a method of approach to the necessary calculations involved is outlined as follows:

Example 1. Cores 4 and 5 in. thick are to be dried by heating to a temperature at 400 F. An intermittent type box oven is to be used, size 12 x 14 x 10 ft with 856 sq ft surface having an average heat transfer of 0.3 Btu per square foot per degree per hour. Drying time as determined by test is 2 hr (Fig. 10). Cores weighing 6 tons, and 15-ton steel plates, trucks etc. are delivered to the drier at 70 F. The oven is heated by an external heater; the products of combustion and 66⅔ per cent recirculated air will be delivered to the oven at 825 F. Fuel oil of 19,980 Btu gross and 18,830 Btu per pound net heating value, weighing 6.75 lb per gallon and having 15 lb product per pound fuel

for perfect combustion. Cores consist of 91 per cent sand, 3 per cent oil binder, and 6 per cent water.

Solution. Heat required per ton of cores:

	Lb Material × Temp. Rise × Sp. Ht. =	Btu
Sand	$0.91 \times 2{,}000 \times (400 - 70) \times 0.2 =$	120,120
Binder	$0.03 \times 2{,}000 \times (400 - 70) \times 0.4 =$	7,920
Water heating	$0.06 \times 2{,}000 \times (212 - 70) \times 1.0 =$	17,040
Water evaporation	$0.06 \times 2{,}000 \times$ 970 (Fig. 4) =	116,520
Water superheating (approx. 50 per cent reaches 575 F)	$= 0.5 \times 0.06 \times 2{,}000 \times (575 - 212) \times 0.45 =$	9,800
Total Heat		271,400 Btu

Heating Load First Hour

	Heated to		Btu
Sand	212 F	$\frac{142}{330} \times 120{,}120$ =	51,688
Binder[a]	212 F	$\frac{142}{330} \times 7{,}920$ =	3,408
Water	212 F	=	17,040
Evaporation	66.7%	$0.667 \times 116{,}520$ =	77,680
Superheat	66.7%	$0.667 \times 9{,}800$ =	6,530
Total Per Ton			156,346
For 6 ton		$6 \times 156{,}346$ =	938,076
Steel plates	390 F	$320 \times 30{,}000 \times 0.12$ =	1,152,000
Radiation[b]	422 F Avg.	$352 \times 856 \times 0.30$ =	90,394
Total			2,180,470

Heating Load Second Hour

Sand	400 F	$\frac{188}{330} \times 120{,}120$ =	68,432
Binder[a]	400 F	$\frac{188}{330} \times 7{,}920$ =	4,512
Water			
Evaporation	33.3%	$0.333 \times 116{,}520$ =	38,840
Superheat	33.3%	$0.333 \times 9{,}800$ =	3,270
Total Per Ton			115,054
For 6 ton		$6 \times 115{,}054$ =	690,324
Steel plates	460	$70 \times 30{,}000 \times 0.12$ =	252,000
Radiation[b]	575	$505 \times 856 \times 0.30$ =	129,684
Total			1,072,008

[a]Binder oxidizes and liberates heat, which is neglected in this calculation.

[b]Average value of coefficient is less than 0.3 because oven is not up to 575 F. This is neglected. 422 F is arrived at by taking area under curve as compared to area under 575 F ordinate.

Heat in 1 lb fuel oil	=		18,830 Btu
Heater Loss (10 per cent)	=	1883	
Duct Loss (5 per cent)	=	942	2,825 Btu
			16,005 Btu available to heat oven.

Heat content of gases in 1 lb fuel oil at 825 F is 205 Btu (Fig. 8)

15 lb × 205	=	3,075 Btu sensible heat in products of perfect combustion.
		12,930 Btu to heat air X and Y (Fig. 11).

$$Y\,(S_{825} - S_{422}) + X\,(S_{825} - S_{70}) = 12{,}930 \tag{4}$$

$Y = 2\,(X + 15)$ for 66.7 per cent recirculation

where

S = heat content of air at temperature noted taken from Fig. 8.

(Recirculation and exhaust contains water vapor, products of combustion, and a greater portion of air. Heat capacities of all vary so little that they have all been assumed to be air).

$S_{825} - S_{422} = 190 - 91 = 99$

$S_{825} - S_{70} = 190 - 8.6 = 181.4$

Substituting values of Y, H, etc. in Equation 4,

$(2\,X + 30)\,99 + 181.4\,X = 12{,}930$

$X = 26.3$ lb excess air.

$Y = 82.6$ lb recirculating air.

Total = 26.3 + 82.6 + 15 = 123.9 lb air and products of combustion circulated per pound fuel burned.

Heat in air exhausted from oven at 422 F per pound fuel burned = $0.333 \times 123.9 \times (S_{422} - S_{70}) = 41.3\,(91 - 8.6) = 3{,}400$ Btu.

Btu available for heating material = 16,005 − 3,400 = 12,605 Btu per pound fuel.

Fuel used in first hour = 2,180,470 ÷ 12,605 = 173 lb = 25.6 gal.

During the second hour the heater capacity will be much greater than required. If an automatic oven temperature control operates on the oil supply, the delivery temperature of the air entering the oven and the quantity of oil burned will decrease, the air supply being constant.

Heat in air exhausted = $41.3\,(S_{675} - S_{70}) = 41.3\,(127 - 8.6) = 4{,}880$ Btu per pound fuel.

Heat available for heating material = 16,005 − 4,880 = 11,125 Btu.

Fuel used in second hour = 1,072,008 ÷ 11,125 = 96.5 lb oil = 14.3 gal.

Total oil used per load = 25.6 + 14.3 = 39.9 gal.

ESTIMATING METHODS

Values based on practical experience are available for rough estimating of drying problems. The temperature will drop approximately 8.5 F per grain of water evaporated per cubic foot of air (measured at 70 F) or approximately 0.62 F per pound of air at any temperature. Air will drop 55 F per cubic foot for each Btu extracted. Generally air will absorb from 2 grains to 5 grains per cubic foot of air in one passage through an air drier, depending on the temperature and the degree of contact with the material. The amount of steam required to evaporate a pound of water will vary from 1.5 lb to a more usual figure of from 2.5 to 3 lb of steam per pound of water evaporated.

REFERENCES

Commercial Drying Apparatus, by L. P. Dwyer (A.S.H.V.E. TRANSACTIONS, Vol. 22, 1916, p. 479).

Artificial Drying with Special Reference to the Use of Gas, by G. C. Shadwell (A.S.H. V.E. TRANSACTIONS, Vol. 23, 1917, p. 231).

Drying by Evaporation, by F. R. Still (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 255).

High Temperature Drying, by Burt S. Harrison (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 7).

The Temperature of Evaporation, by W. H. Carrier (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 25).

Commercial Dehydration, by J. E. Whitley (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 551).

Drying as an Air Conditioning Problem, by A. W. Lissauer (A.S.H.V.E. TRANSACTIONS, Vol. 27, 1921, p. 251).

A Chronological Survey of Drying and Driers, by J. E. Bolling (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1921, p. 715).

Calculations for Drying Design, by Grosvenor (*Transactions, American Institute Chemical Engineering*, 1908, p. 184).

The Rate of Drying Solid Materials, by J. Lewis (*Industrial Engineering Chemistry*, 1921, p. 427).

Adiabatic Drying of Hygroscopic Solids, by A. M. McCready, and W. L. McCale (*Transactions, American Institute Chemical Engineers*, 1933).

Lignite Drier, by Lavine & Sutherland (*Chemical and Metallurgical Engineering*, July, 1929).

Enameling Oven Economy, by B. S. Harrison (*Fuels & Furnaces*, February, 1931).

Powdered Yeast Prepared by Spray Drying, by A. W. Farrell (*Food Industry*, December, 1931).

Factors that Influence Drier Performance, by A. Weisselberg (*Chemical and Metallurgical Engineering*, August, 1932).

Principles of Drying Lumber and Humidity Diagram, by H. D. Tiemann (*Forest Service Bulletin* 104, 1912).

Symposium on Drying. Articles by W. K. Lewis, W. H. Carrier, A. E. Stacey, Jr. and R. S. Fleming, R. G. Metz, G. B. Ridley, C. O. Lavett, D. J. Van Marle (*Journal Industrial Engineering Chemistry*, May, 1921).

The Drying of Solids, by T. K. Sherwood (*Bulletin Massachusetts Institute Technology* Nos. 237, 247 and 258).

Air Conditioning and Engineering, American Blower Co.

Combustion (*American Gas Association*, 3rd edition, 1932).

Die Trockentechnik, by M. Hirsch (Julius Springer, Berlin, 1932).

Drying (*Kent's, Mechanical Engineers Handbook*, 10th edition, 1923; 11th edition, 1936).

Drying, by W. H. Carrier (*Marks', Mechanical Engineers Handbook*, 3rd edition, 1930).

Drying, by Perry (*Chemical Engineers Handbook*, 1934).

Drying by Means of Air and Steam, by E. Hausbrand (D. Van Nostrand & Co., 1901).

Drying in Industrial Plants, by J. O. Ross.

Elements of Chemical Engineering, by Badger and McCabe (McGraw Hill Co., 1931).

Fan Engineering, Buffalo Forge Co.

Fuels and Their Combustion, by Haslam and Russell (McGraw Hill Co., 1926).

Heat Transmission, by W. H. McAdams (McGraw Hill Co., 1933).

Modern Drying Machinery, by H. B. Grenshaw, London, 1926.

Principles of Chemical Engineering, by Walker, Lewis, McAdams (Chapters on Evaporation, Humidity and Drying, 2nd edition, McGraw Hill Co.).

The Kiln Drying of Lumber, by A. Koehler and R. Thelen, New York, 1926.

The Kiln Drying of Lumber, by H. D. Tiemann (Lippincott, 1920).

Chapter 41

NATURAL VENTILATION

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces, supplemented in certain cases by wind-actuated devices, finds application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for the displacement of air in buildings are (*a*) wind forces, and (*b*) the difference in temperature between the air inside and outside the building, or a combination of the two. The results that are obtained by natural ventilating systems are variable, as they depend on wind action and temperature difference. The arrangement and control of ventilating openings should be such that the two forces act cooperatively and not in opposition.

WIND FORCES

In considering the use of natural wind forces for the operation of a ventilating system, account must be taken of (1) average and minimum wind velocities, (2) wind direction, (3) seasonal, daily and hourly variations in wind velocity and direction, and (4) local wind interference by buildings, trees, etc.

Table 1, Chapter 6, gives values for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 5, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While average wind velocities are seldom below 5 mph, there are many hours in each month during which the wind velocity is from 3 to 5 mph, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the hourly wind velocity falls much below 3 mph for more than 10 daylight hours per month. Usually a natural ventilating system should be designed to operate satisfactorily with a wind velocity of 3 to 6 mph, depending on locality.

The following formula may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings:

$$Q = EAV \tag{1}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlet (or outlet) openings, square feet.
V = wind velocity, feet per minute,
= miles per hour $\times$ 88.
E = effectiveness of openings.

(E should be taken at from 50 to 60 per cent if the inlet openings face the wind and from 25 to 35 per cent if the inlet openings receive the wind at an angle.)

If outlet openings, where air leaves a building, are smaller than inlet openings, where air enters a building, the air will be less effective than indicated by the constant E.

The accuracy of the results obtained by the use of Formula 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edge orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less, and if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the following four places:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 1).
3. In a monitor on the side opposite from the wind.
4. In roof ventilators or stacks exposed to the full force of the wind[1].

Forces Due to Stack Effect [2]

The stack effect produced within a building when the outdoor temperature is lower is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\,A\,\sqrt{H\,(t - t_o)} \tag{2}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlets or outlets (assumed equal), square feet.
H = height from inlets to outlets, feet.
t = *average* temperature of indoor air in height H, degrees Fahrenheit.
t_o = temperature of outdoor air, degrees Fahrenheit.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

The height between inlets and outlets should be the maximum which the building construction will allow.

---

[1]Airation of Industrial Buildings, by W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 159).
[2]Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 59).
Predetermining Airation of Industrial Buildings, by W. C. Randall and E. W. Conover (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 605).

TYPES OF OPENINGS

Types of openings may be classified as: (1) windows, doors, monitor openings and skylights, (2) roof ventilators, (3) stacks connecting to registers, and (4) specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in

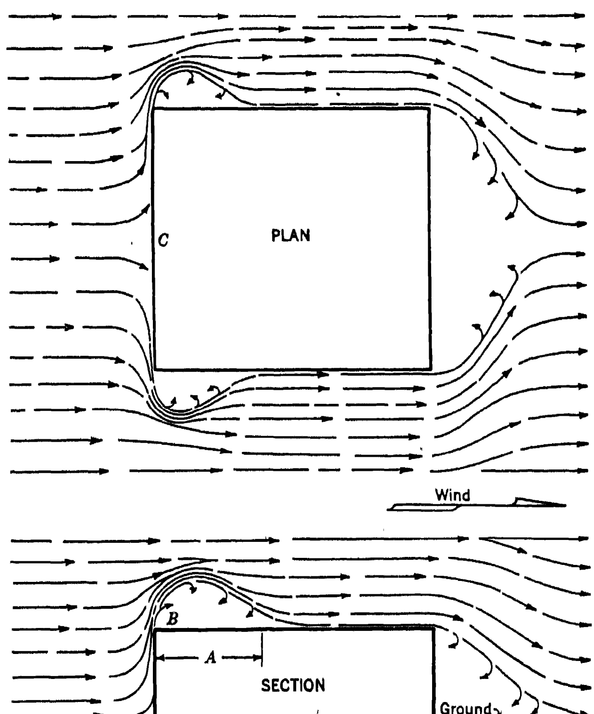


FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (*A*—LENGTH OF SUCTION AREA; *B*—POINT OF MAXIMUM INTENSITY OF SUCTION; *C*—POINT OF MAXIMUM PRESSURE)

various ways; they may open by sliding as in the ordinary double-hung windows, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top, bottom or side.

The proper distribution of the air in spaces to be ventilated is as important as that of sufficient air quantity. Advantageous pivoting of sash is very useful for securing good air distribution. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet. If it is of a type which is sensitive to wind action addi-

tional flow capacity will be produced. The capacity of a ventilator at a constant wind velocity and temperature difference, depends upon four things: (1) its location on the roof, (2) the resistance it and the duct work offers to air flow, (3) the height of draft, and (4) the efficiency of the ventilator in utilizing the kinetic energy of the wind for inducing flow by centrifugal or ejector action.

For maximum flow induction, a ventilator should be located on that part of the roof which receives the full wind without interference. (See Fig. 1.) This does not mean that any ventilators are to be installed within the suction region created by the wind jumping over the building, or in a light court, or on a low building between two high buildings. Ventilators are highly effective in such low-pressure areas, but their ejector action, caused by wind velocity, is of little value in these locations, and hence their size should be increased proportionally.

The base of the ventilator should always be provided with a taper-cone inlet in order to produce the effect of a bell-mouth nozzle (flow coefficient 0.97) rather than that of a square-entrance orifice (flow coefficient 0.60). If a grille is provided at the base of a ventilator it should be oversized as compared with the ventilator size.

Air inlet openings located at lower levels in the building should be at least equal to, and preferably larger than the combined throat areas of all roof ventilators. The air discharged by a roof ventilator depends on wind velocity and temperature difference, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Several types of roof ventilators are shown in Figs. 2 to 9. These may be classified as *stationary*, Figs. 2 to 4, *pivoted* or *oscillating*, Figs. 5 to 7, or *rotating*, Figs. 8 and 9. When selecting roof ventilators, some attention should be paid to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibilities of noise from dampers or other moving parts, and possible maintenance costs.

Natural ventilation units may be used to supplement power-driven supply fans, and under favorable weather conditions it may be possible to shut down the power-driven units. Where low operating costs are very important, such a combination has great advantages.

Controls

Gravity ventilators may have dampers controlled by (1) hand, (2) thermostatic, and (3) wind velocity, in combination with a fan. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks or vertical flues are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference (the so-called

gravity action). Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction.

With little or no wind, chimney effect depending on temperature difference and lower outdoor temperature will produce a removal of air from the rooms where the inlet openings are located.

HEAT REMOVAL

In problems of heat removal, knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain this temperature difference can be determined by means of the following equation:

$$Q = \frac{VH}{c\,60\,(t - t_o)} \tag{3}$$

where

c = 0.24 = specific heat of air.
V = specific volume of the air, cubic feet per pound, about 13.5. (See Chapter 1.)
H = heat to be carried off, in Btu per hour.
Q = air flow in cubic feet per minute.
t = inside temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.

For disposing of odors or other air impurities, the amount of outside air to be introduced must be of such quantity to dilute the impurities to a degree that they are no longer objectionable. See Chapter 2 for the minimum of outside air necessary for ventilation. For garage ventilation, sufficient air must be admitted to dilute the carbon monoxide content of the indoor air to 1 in 10,000 (see Garage Ventilation in this Chapter).

Suggested methods for estimating the air flow due to temperature difference alone and to wind alone have already been given. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The same openings have been assumed in both cases, and since the resistance to flow through the openings varies approximately with the square of the velocity[3], this resistance becomes a limiting factor as the flow through the openings is increased.

Recent investigations[4] show that the total flow is only 10 per cent above the flow caused by the greater force when the two forces are nearly equal, and this percentage decreases rapidly as one force increases above the other. Tests on roof ventilators indicate that this is too conservative in the direction of low total flow quantities, but there is in any case a large judgment factor involved. The wind velocity and direction, the outdoor temperature, or the indoor activities cannot be predicted with certainty, and great refinement in calculations is therefore not justified. When designing for winter conditions, an added variable is the heat lost by direct flow through walls and windows and by infiltration.

---

[3]Loc. Cit. Notes 1 and 2.

[4]This is true for *turbulent* flow only. It would be more correct to state that the resistance varies approximately with V^2 for high to moderate velocities, with $V^{1.8}$ for moderate to low velocities, and with the first power of the velocity for very low velocities through small openings.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per pound is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Temperature differences are 10 F in summer and 30 F in winter, and the wind velocity is 5 mph in summer and 8 mph in winter. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution. The system must be designed for the summer conditions as these are the more severe. The heat to be removed per hour is:

$$H = 15 \times 7.75 \times 18{,}000 = 2{,}092{,}500 \text{ Btu.}$$

By Equation 3, the air flow required to remove this heat with a temperature difference of 10 deg is:

$$Q = \frac{VH}{c\,60\,(t - t_o)} = \frac{13.5 \times 2{,}092{,}500}{0.24 \times 60 \times 10} = 196{,}172 \text{ cfm.}$$

This is equal to 19.6 air changes per hour. The assumption is made that the average temperature difference between indoors and outdoors is the same as the temperature rise of the air from the inlet opening to the outlet opening. Actually, the latter difference is larger and so the value of 19.6 air changes per hour is conservative as it allows for more cooling than is necessary for an *average* temperature difference of 10 deg.

If 196,172 cfm are to be circulated by the force of the temperature difference alone, the area of opening would be, by Equation 2:

$$A = \frac{Q}{9.4\sqrt{H\,(t - t_o)}} = \frac{196{,}172}{9.4\sqrt{30 \times 10}} = 1{,}205 \text{ sq ft.}$$

If this area of openings were provided, a wind velocity of 5 mph, acting alone, would produce a flow according to Equation 1, of:

$$Q = EAV = 0.50 \times 1{,}205 \times 5 \times 88 = 265{,}100 \text{ cfm.}$$

If the inlet openings do not face the wind, but are at an angle with it, about half this amount may be considered to flow.

A factor of judgment must now be exercised in making the selection of the area of openings to be specified. Apparently 1205 sq ft are a very generous allowance because either a direct wind of 5 mph or an average temperature difference of 10 deg acting alone will more than suffice to carry away the heat, and when the two forces are acting together, the system may have an excess capacity of 25 per cent to 50 per cent, especially if the outlets are made up partially of roof ventilators which employ the force of the wind for producing a suction effect. On the other hand, the wind may at times come from an unfavorable direction, or its velocity may fall below 5 mph or the building construction may not permit a full 2400 sq ft of inlet window area and an equal amount of monitor or roof ventilator outlet area. In case the two sets of openings are not equal, their effectiveness is reduced.

From this example, it must be apparent that while formulae may furnish a reliable guide, the final solution of a problem of natural ventilation requires a common sense analysis of local conditions to supplement and to modify the dictates of the formulae.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings in the building should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone to be ventilated.

2. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

3. Roof ventilators should be located 20 to 40 ft apart each way and preferably on the ridge of the roof. The closer spacings are used when ventilating rooms with low ceilings.

4. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

5. In an industrial building where furnaces, that give off heat and fumes, are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

6. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

7. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

8. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

9. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

10. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. On the other hand, where the direction of the wind is quite variable, it may be stated as a general principle that windows should be arranged in sidewalls and monitors so that there will be approximately equal area on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force of the wind, and others opposed to a suction force, and effective movement through the building will be assured.

11. The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. Special consideration should be given to the possibility of sidewall or monitor windows being closed on account of weather conditions. Such possibilities favor roof ventilators and specially designed stormproof inlets.

MEASUREMENT OF NATURAL AIR FLOW

The determination of the performance of any ventilating system involves measurements which are not easy to make. The difficulties are increased in the case of natural ventilation, since the motive forces and the air velocities are very small. The measurements necessary for giving the *capacity* of a system are (1) velocity of the wind, (2) velocity of the air through inlet and outlet openings, (3) outdoor air temperature, and (4) average indoor air temperature. (See Chapter 34.)

DAIRY BARN VENTILATION [5]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cu ft per hour of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per hour per cubic foot of barn space, and 0.197 to 0.305 Btu per hour per square foot of barn exposure.

GARAGE VENTILATION

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means particularly during the mild weather when doors and windows can be kept

[5]Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 181).
Cow Barn Ventilation, by Alfred J. Offner (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 149).
For additional information on this subject refer to *Technical Bulletin, U. S. Department of Agriculture* (1930), by M. A. R. Kelley.

open. However, the A.S.H.V.E. Code for Heating and Ventilating Garages, adopted in 1929 and revised in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.[6]

Research

Research on garage ventilation undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., in cooperation with the A.S.H.V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed in the following statements:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level.

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cu ft per hour, with an average rate of 35 cu ft per hour.

5. An air change of 350,000 cu ft per hour per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

---

[6]Code for Heating and Ventilating Garages (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 355), (A.S.H.V.E. Reprint, January, 1935).

Airation Study of Garages by W. C. Randall and L. W. Leonhard (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 233).

A.S.H.V.E. RESEARCH REPORT No. 874—Carbon Monoxide Concentration in Garages, by A.S. Langsdorf and R. R. Tucker (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 511).

A.S.H.V.E. RESEARCH REPORT No. 935—Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 439).

A.S.H.V.E. RESEARCH REPORT No. 934—Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 424).

A.S.H.V.E. RESEARCH REPORT No. 967—Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 395).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 263).

Chapter 42

PIPE AND DUCT INSULATION

Heat Transmission by Radiation and Convection, Heat Losses from Bare and Insulated Pipes, Heat Losses from Ducts, Low Temperature Insulation, Insulation of Pipes to Prevent Freezing, Economical Thickness of Pipe Insulation, Underground Pipe Insulation

HEAT is transmitted to or from pipes and ducts by radiation and convection. The radiant heat transfer per unit area is independent of the geometrical shape, whereas the convected heat depends to a considerable extent on the shape factor. In many cases, it is desirable to calculate the rate of heat transmission from a surface by radiation and convection, as the total rate of transfer is different, for instance, from a heating installation than from a cooling installation with an equal difference in temperature between the surface and the surrounding atmosphere.

HEAT TRANSMISSION BY RADIATION AND CONVECTION

The heat transmission by radiation from a surface to the surrounding surfaces can be calculated from the well-known Stefan-Boltzman formula:

$$q_r = 17.4 \times 10^{-10} \times p\,(T_1^4 - T_2^4) \tag{1}$$

where

q_r = heat transmission by radiation, Btu per square foot per hour.
p = *effective* emissivity of surface and surroundings.
T_1 = temperature of hotter surface, degrees Fahrenheit absolute.
T_2 = temperature of cooler surface, degrees Fahrenheit absolute.

The heat transmission by free or natural convection can be determined from the formula:

$$q_c = C\left(\frac{1}{D}\right)^{0.2}\left(\frac{1}{T\,\text{av.}}\right)^{0.181} dt^{1.266} \tag{2}$$

where

q_c = heat transmission by convection, Btu per square foot per hour.
C = a constant depending upon the surface shape.
D = diameter of pipe or circular duct or height of vertical wall, inches. (effect of diameter or height becomes constant at 24 in.)
T av. = average wall surface and surrounding air temperature, degrees Fahrenheit absolute.
dt = temperature excess between wall surface and surrounding air, degrees Fahrenheit.

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 1 for cold surfaces as low as − 39 F to warmer surfaces as high as 139 F. The emissivities of a number of surfaces ordinarily encountered in engineering practice are shown in Table 2[1].

For horizontal cylinders, the value of C = 1.016 has been well established by various investigations. For vertical plates, the value of C =

TABLE 1. HEAT TRANSMISSION BY RADIATION FOR BLACK-BODY CONDITIONS[a]

Expressed in Btu per square foot per hour

TEMP. DEG F	0	−1	−2	−3	−4	−5	−6	−7	−8	−9
−30	59.3	58.7	58.2	57.7	57.2	56.7	56.2	55.7	55.2	54.7
−20	65.2	64.7	64.1	63.5	62.9	62.3	61.7	61.1	60.5	59.9
−10	71.4	70.8	70.1	69.5	68.9	68.3	67.7	67.1	66.4	65.8
0	78.0	77.4	76.7	76.0	75.4	74.7	74.0	73.4	72.7	72.1
	0	+1	+2	+3	+4	+5	+6	+7	+8	+9
0	78.0	78.7	79.4	80.1	80.8	81.5	82.2	82.9	83.6	84.3
10	85.0	85.7	86.5	97.2	88.0	88.7	89.4	90.2	90.9	91.7
20	92.4	93.3	94.0	94.8	95.6	96.4	97.2	98.0	98.8	99.6
30	100	101	102	103	104	105	105	106	107	108
40	109	110	111	112	112	113	114	115	116	117
50	118	119	120	121	122	123	123	124	125	126
60	127	128	129	130	131	132	133	134	135	136
70	137	138	139	140	142	143	144	145	146	147
80	148	149	150	151	152	153	154	155	156	157
90	159	160	161	162	163	164	166	167	168	169
100	170	171	173	174	175	176	178	179	180	182
110	183	184	185	187	188	189	191	192	193	195
120	196	197	199	200	201	203	204	206	207	209
130	211	212	214	215	217	218	220	221	222	224

[a]*Example:* Radiation from walls of room at 32 F to surface at − 25 F for effective emissivity of 0.95 = (102 − 62.3) 0.95 = 37.7 Btu per square foot per hour.

TABLE 2. EMISSIVITY VALUES FOR VARIOUS SURFACES

SURFACE	TEMPERATURE, DEG F			
	100	200	300	400
Bright galvanized iron	0.14	0.15	0.16	0.17
Badly tarnished galvanized iron	0.53	0.53	0.53	0.53
Very bright copper	0.07	0.07	0.07	0.11
Tarnished copper	0.43	0.44	0.46	0.48
Aluminum foil	0.06	0.06	0.07	....
Aluminum paint	0.28	0.28	0.28	....
Asbestos lumber (smooth side)	0.79	0.79	0.79	0.79
Hardwood (sanded)	0.90	0.90	....	....
Black Feltex paper	0.87	0.89	....	....
Mica-surfaced asphalt roofing	0.76	0.76	....	....
Window glass	0.88	0.89	0.90	0.90
White canvas	0.88	0.88	0.89	....

[1]Heat Insulation in Air Conditioning, by R. H. Heilman (*Industrial and Engineering Chemistry*, Vol. 28, July 1936, p. 782).

1.394 has been fairly well established. A value of C = 1.79 for horizontal plates warmer than the surrounding air facing upward and 0.89 for horizontal plates warmer than air facing downward is indicated by recent investigations[2].

The heat transmission by free convection from vertical walls 24 in. or more in height is given in Table 3 as calculated from Equation 2 for ambient air temperature of 80 F. The values in Table 3 will not be changed appreciably by a considerable change in air temperature for a given temperature excess. For instance, a change in air temperature from 80 to 40 F will increase the heat transmission given in Table 3 by only 1.3 per cent.

TABLE 3. HEAT TRANSMISSION BY FREE CONVECTION FOR LARGE VERTICAL SURFACES

Expressed in Btu per square foot per hour

Temp. Deg F	Temperature Difference between Body and Surrounding Still Air at 80 F													
	0	10	20	30	40	50	60	70	80	90	100	110	120	130
0	0	4.4	10.4	17.4	25.0	33.2	41.8	50.6	59.9	69.4	79.4	89.2	99.4	109.8
1	0.3	4.9	11.1	18.1	25.8	34.1	42.6	51.5	60.8	70.3	80.4	90.2	100.4	110.9
2	0.6	5.5	11.8	18.9	26.7	34.9	43.5	52.4	61.8	71.3	81.4	91.2	101.5	112.0
3	1.0	6.0	12.5	19.7	27.5	35.7	44.3	53.4	62.7	72.3	82.4	92.2	102.6	113.0
4	1.4	6.6	13.2	20.5	28.3	36.6	45.2	54.3	63.7	73.3	83.3	93.3	103.6	114.1
5	1.8	7.3	13.9	21.2	29.2	37.4	46.1	55.2	64.6	74.3	84.2	94.3	104.7	115.2
6	2.3	7.9	14.6	22.0	30.0	38.3	47.0	56.1	65.6	75.3	85.2	95.3	105.7	116.3
7	2.8	8.5	15.3	22.7	30.8	39.1	47.8	57.1	66.5	76.3	86.2	96.3	106.7	117.3
8	3.3	9.1	16.0	23.5	31.6	40.0	48.7	58.0	67.5	77.4	87.2	97.4	107.8	118.4
9	3.8	9.7	16.7	24.3	32.4	40.9	49.7	59.0	68.4	78.4	88.2	98.4	108.8	119.5

TABLE 4. FREE CONVECTION FACTORS FOR VARIOUS SHAPES

Shapes	Factor
Horizontal cylinders 24 in. in diam. or over	0.73
Long vertical cylinders 24 in. in diam. or over	0.88
Vertical plates 24 in. in height or over	1.00
Horizontal plates warmer than air facing upward	1.28
Horizontal plates warmer than air facing downward	0.64
Horizontal plates cooler than air facing upward	0.64
Horizontal plates cooler than air facing downward	1.28

TABLE 5. FREE CONVECTION FACTORS FOR VARIOUS DIAMETER PIPES OR VARIOUS HEIGHT PLATES

Actual o. d., or height, in	1	2	3	4	5	6	7	8
Factor	1.88	1.64	1.52	1.43	1.37	1.32	1.28	1.25
Actual o. d. or height, in	9	10	12	14	16	18	20	22
Factor	1.22	1.19	1.15	1.11	1.09	1.06	1.04	1.02

[2]The Transmission of Heat by Radiation and Convection, by Griffith and Davis (*Special Report* No. 9, 1922, *Department of Scientific and Industrial Research*, His Majesty's Stationery Office, London, England).

multiplied to obtain the free convective transfer from various shapes whose characteristic dimensions are 24 in. or over, and Table 5 gives the factors to be used in conjunction with the factors in Table 4 for obtaining the free convection from Table 3 for pipes and ducts whose characteristic dimensions are less than 24 in.

TABLE 8. HEAT LOSS FROM BRIGHT COPPER PIPE GIVEN ONE THIN COAT OF CLEAR LACQUER

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.240	0.265	0.282	0.307	0.401	0.461	0.478
¾	0.320	0.356	0.373	0.414	0.477	0.571	0.578
1	0.390	0.437	0.463	0.507	0.598	0.681	0.710
1¼	0.470	0.537	0.554	0.614	0.700	0.812	0.840
1½	0.540	0.612	0.645	0.714	0.830	0.966	0.990
2	0.690	0.762	0.818	0.892	1.005	1.164	1.201
2½	0.840	0.937	0.991	1.085	1.178	1.361	1.420
3	0.960	1.025	1.135	1.270	1.400	1.625	1.700
3½	1.100	1.250	1.318	1.442	1.580	1.845	1.905
4	1.241	1.400	1.480	1.556	1.750	2.040	2.130
4½	.....	.....	.....	.....	1.910	2.240	2.350
5	1.480	1.685	1.790	1.965	2.130	2.415	2.610
6	1.700	1.936	2.052	2.272	2.450	2.810	2.990
8	2.200	2.500	2.630	2.854	3.120	3.425	3.730

TABLE 9. HEAT LOSS FROM HORIZONTAL TARNISHED COPPER PIPE

Expressed in Btu per hour per linear foot per degree Fahrenheit between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.250	0.287	0.300	0.321	0.433	0.500	0.530
¾	0.340	0.381	0.409	0.429	0.533	0.543	0.654
1	0.440	0.475	0.509	0.536	0.636	0.746	0.803
1¼	0.500	0.559	0.618	0.622	0.764	0.878	0.934
1½	0.580	0.656	0.710	0.750	0.904	1.053	1.120
2	0.730	0.825	0.890	0.957	1.101	1.273	1.364
2½	0.880	1.000	1.091	1.143	1.305	1.490	1.605
3	1.040	1.175	1.272	1.343	1.560	1.800	1.940
3½	1.180	1.350	1.454	1.535	1.750	2.020	2.170
4	1.460	1.500	1.635	1.715	1.941	2.240	2.430
4½	.....	.....	.....	.....	2.131	2.465	2.650
5	1.600	1.812	1.980	2.071	2.387	2.770	2.990
6	1.840	2.125	2.270	2.430	2.740	3.210	3.440
8	2.400	2.685	2.910	3.110	3.310	4.050	4.370

TABLE 10. RADIATING SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

TABLE 11. RADIATING SURFACE PER LINEAR FOOT OF COPPER TUBING
Outside diameter ⅛ in. greater than nominal size

Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)
½	0.164	2	0.556	5	1.342
¾	0.229	2½	0.687	6	1.604
1	0.295	3	0.818	8	2.128
1¼	0.360	3½	0.949	..	.....
1½	0.426	4	1.080	..	.....

For example, the free convection transfer from a 3 in. o.d. horizontal cylinder for a temperature difference of 40 F = 25.0 × 0.73 × 1.52 = 27.7 Btu per square foot per hour.

The increased rate of heat transfer due to forced convection can be calculated from the formula:

$$q_{fc} = 1 + 0.225V \tag{3}$$

where

q_{fc} = heat transfer by forced convection, Btu per square foot per hour per degree Fahrenheit temperature difference.

V = velocity of air, feet per second.

This formula is approximately correct for large surfaces exposed to air currents at temperatures of approximately 70 to 80 F.

HEAT LOSSES FROM BARE PIPES

Heat losses from horizontal bare steel pipes, based on tests conducted at *Mellon Institute* and calculated from Equations 1 and 2, are given in Table 6. The monetary values of the loss of heat given in Table 6 may be obtained by means of Fig. 1 for various heating system efficiencies, temperature differences, and calorific values, and costs of coal. To solve a problem, select the proper heat loss coefficient from Table 6 and locate this value on the upper left-hand margin of the chart. Then draw lines in the order indicated by the dotted lines, the dollar value of the heat loss per 100 linear feet of pipe per 1000 hours being given on the upper right-hand scale. In using the chart, the cost of coal should also include the labor for handling it, boiler room expense, etc.

Heat losses from horizontal copper tubes and pipes with bright, lacquered and tarnished surfaces are given in Tables 7, 8, and 9[3].

[3]Heat Loss from Copper Piping, by R. H. Heilman (*Heating, Piping and Air Conditioning*, September, 1933, p. 458).

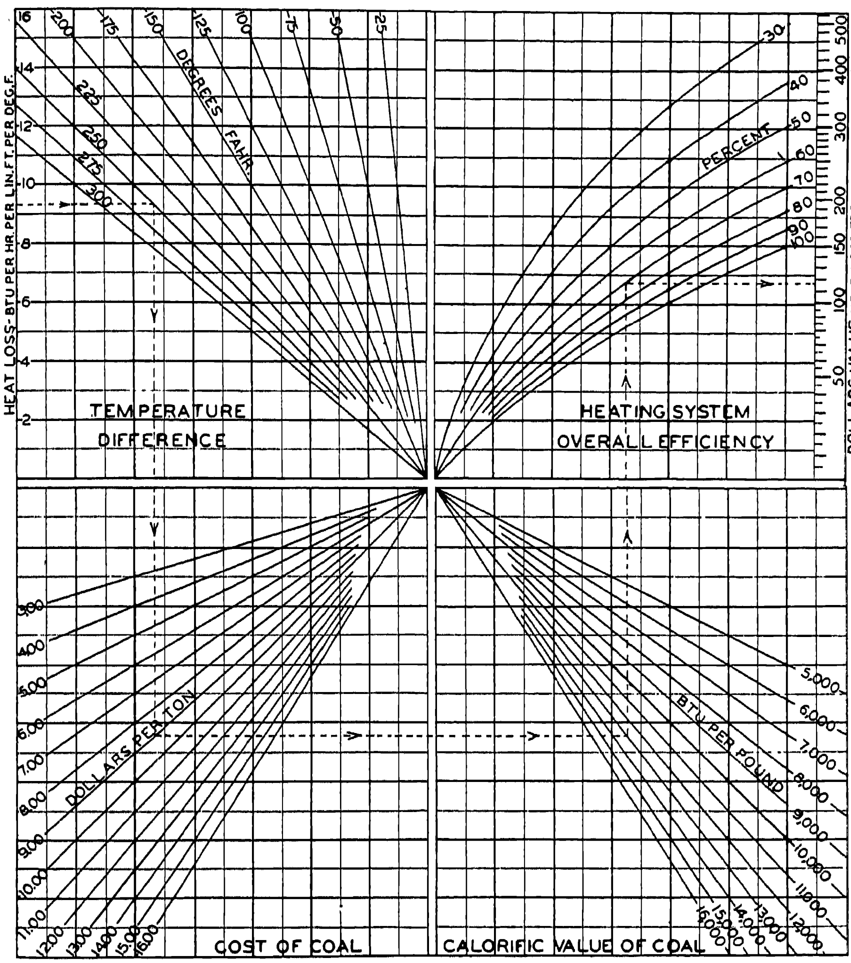


FIG. 1. CHART FOR ESTIMATING DOLLAR VALUE OF HEAT LOSS FROM BARE IRON PIPES. (SEE TABLE 6)[a]

[a]This chart is based on 100 linear feet per 1000 hours. For fractions or multiples of these factors, multiply by proper percentage.

The area in square feet per linear foot of pipe is given in Table 10 for various standard pipe sizes, and Table 11 for copper tubing, while Table 12 gives the area in square feet of flanges and fittings for various standard pipe sizes. These tables can be used to advantage in estimating the amount of insulating cement required for various equipment.

Very often, when pipes are insulated, flanges and fittings are left bare so as to allow for easy access to the fittings in case of repairs. The fact that a pair of 8-in. standard flanges having an area of 2.41 sq ft would lose, at 100 lb steam pressure, an amount of heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

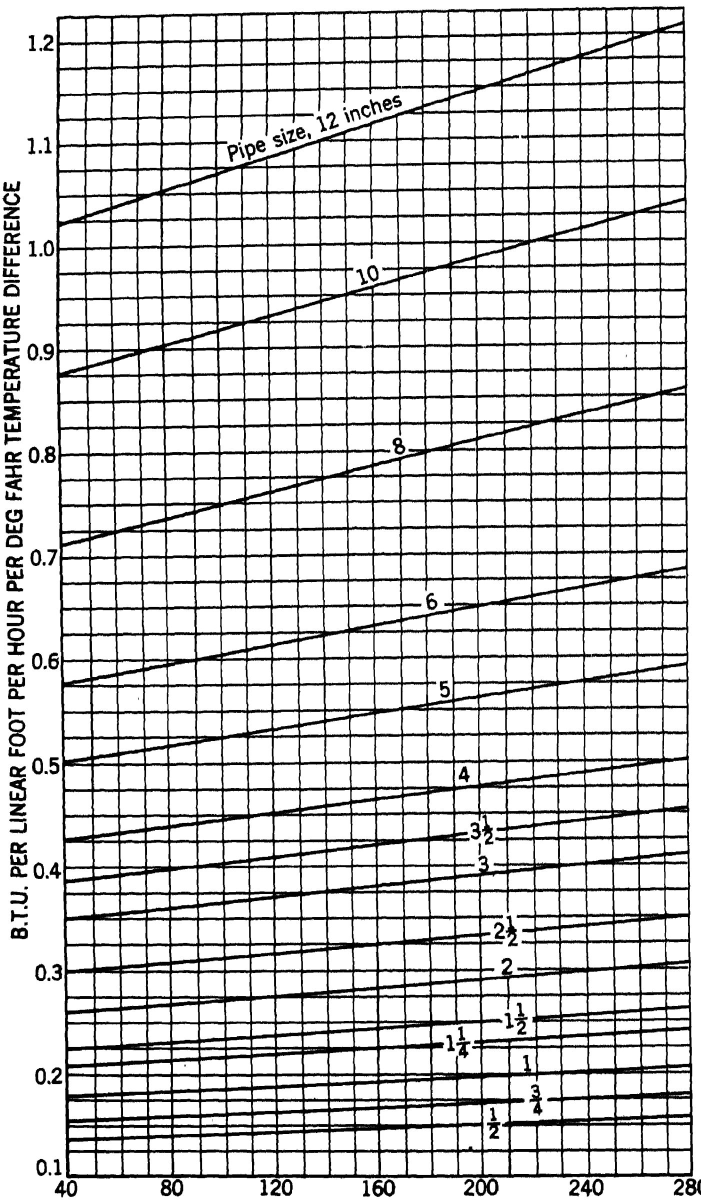


FIG. 2. HEAT LOSS THROUGH 1 IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

TABLE 12. AREAS OF FLANGED FITTINGS, SQUARE FEET[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a]Including areas of accompanying flanges bolted to the fitting.

Example 1. Compute the total annual heat loss from 165 ft of 2 in. bare pipe in service 4000 hours per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

Solution. The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained by interpolation from Table 8 Chapter 1. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 F. By interpolation of Table 6 between temperature differences of 157.1 and 227.7 F, the heat loss from a 2 in. pipe at a temperature difference of 169.4 F is found to be 1.624 Btu per hour per linear foot per degree temperature difference. The total annual heat loss from the entire line = 1.624 × 169.4 × 165 (linear feet) × 4000 (hours) = 181,600 *Mb*.

Example 2. Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in the previous example. If the system is operating at an overall efficiency of 55 per cent, determine the monetary value of the annual heat loss from the line.

Solution. The cost of heat per 1000 *Mb* supplied to the system = 1,000,000 × 11.5

TABLE 13. CONDUCTIVITY (k) OF VARIOUS TYPES OF INSULATING MATERIALS FOR MEDIUM AND HIGH TEMPERATURE PIPES[a]

Types of Insulating Materials	Mean Temperature, Deg F				
	100	200	300	400	500
85 per cent Magnesia Type	0.359	0.403	0.448	0.493	0.539
Corrugated Asbestos Type (4 Plies per 1 in. thick)	0.495	0.618	0.741	0.864	--------
Corrugated Asbestos Type (8 Plies per 1 in. thick)	0.505	0.598	0.692	0.786	--------
Laminated Asbestos Type (30-40 Laminations per 1 in. thick)	0.326	0.380	0.434	0.488	0.543
Laminated Asbestos Type (14-20 Laminations per 1 in. thick)	0.374	0.445	0.518	0.589	0.662
Mineral Wool Type	0.350	0.410	0.470	0.530	0.590
High Temperature Type (Diatomaceous Earth and Asbestos)	0.576	0.614	0.652	0.689	0.726
Brown Asbestos Type (Felted Fiber)	0.338	0.396	0.453	0.510	0.568

[a]From tests conducted at *Mellon Institute*.

(dollars) ÷ 13,000 (Btu) × 2000 (lb) × 0.55 (efficiency) = \$0.804. The total cost of heat lost per year = 0.804 × 181.6 (thousand Btu) = \$146.00. (A closely approximate solution of such a problem may be made quickly by the use of the estimating chart given in Fig. 1.)

HEAT LOSSES FROM INSULATED PIPES

The conductivities of various materials used for insulating steam and hot water systems are given in Table 13. They are given as functions of

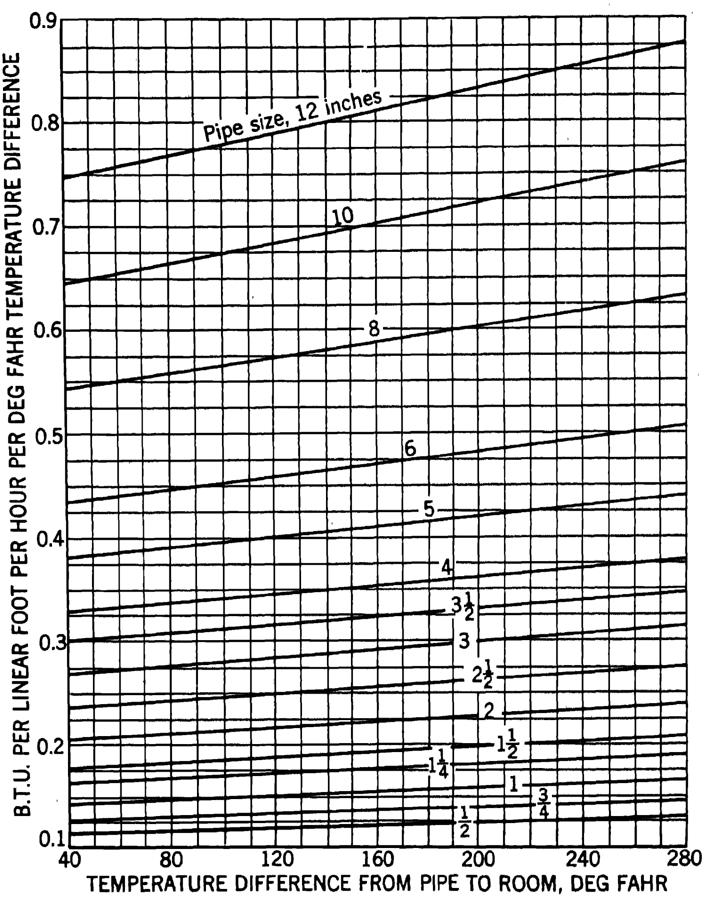


FIG. 3. HEAT LOSS THROUGH 1½ IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

the mean temperatures or the mean of the inner and outer surface temperatures of the insulations. It should be emphasized that they are the average values obtained from a number of tests made on each type of material, also that all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

The heat losses through 1, 1½, and 2-in. thick 85 per cent Magnesia

type of insulation for temperature differences between the pipe and the surrounding atmosphere up to 280 F are shown in Figs. 2, 3, and 4. Standard thicknesses of 85 per cent Magnesia pipe covering are not exactly 1 in. However, the loss through any given thickness of insulation can be obtained by interpolation. Also, the losses through any of the insulations given in Table 13 can be obtained by multiplying the losses obtained from Figs. 2, 3, or 4 by the factors given in Table 14.

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of

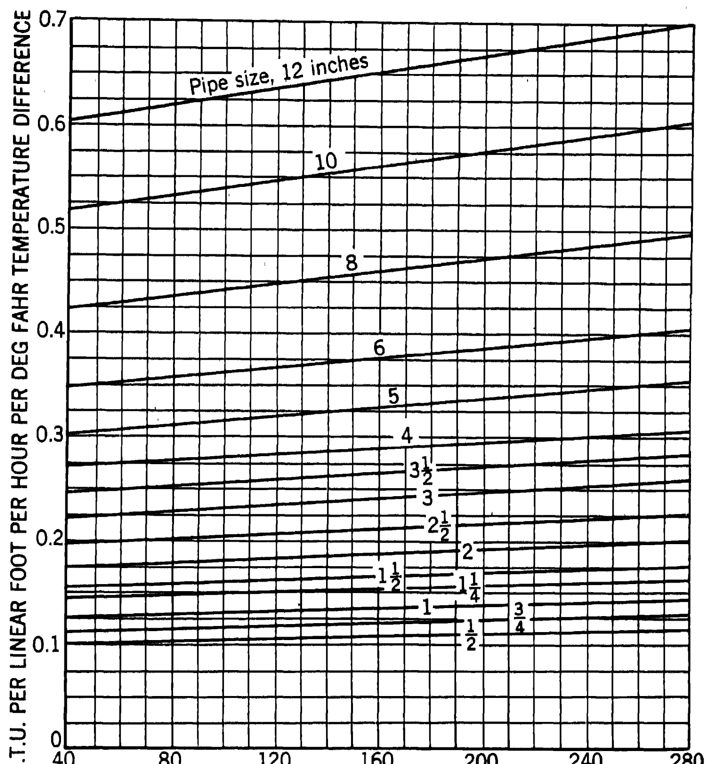


FIG. 4. HEAT LOSS THROUGH 2 IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

well-insulated surfaces, the increases in losses due to air velocity are very small as compared with increases from bare surfaces, which is indicated by Equation 3, because of the fact that air flowing over the surface of the insulation can increase only the rate of heat transfer from surface to air, and cannot change the internal resistance to heat flow inherent in the insulation itself. The maximum increase in heat loss due to air velocity ranges from about 30 per cent in the case of 1-in. thick insulation, to about 10 per cent in the case of 3-in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface.

If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be sealed as tightly as possible. Pipe insulation exposed to the elements should be thoroughly waterproofed.

Example 3. If the steam line given in Examples 1 and 2 is covered with 1 in. thick 85 per cent magnesia, determine the resulting total annual loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

Solution. By referring to Fig. 2, the coefficient for 1 in. magnesia on a 2 in. pipe is found to be 0.285 Btu per hour per linear foot of pipe per degree temperature difference at a temperature difference of 169.4 F. The total hourly loss per linear foot of pipe will then be $0.285 \times 169.4 = 48.3$ Btu. The total annual loss through the insulation $= 48.3 \times 165$ (linear feet) $\times 4000$ (hours) $= 31,900$ *Mb*. The annual bare pipe loss as determined in the solution of Example 1 was found to be 181,600 *Mb*. The saving due to insulation is then $181,600 - 31,900 = 149,700$ *Mb* per year.

From the solution of Example 2, it was found that the heat supplied to the system cost $0.804 per thousand *Mb*. Therefore, the monetary value of the saving = 0.804 (dollars) × 149.7 (thousand *Mb*) = $120.36, or 82.4 per cent of the cost when using uninsulated pipe.

TABLE 14. PIPE COVERING FACTORS

Types of Insulating Materials	Temperature Difference, Pipe to Air, Deg F					
	100	200	300	400	500	600
85 per cent Magnesia Type	1.050	1.024	0.997	0.971	0.944	0.918
Corrugated Asbestos Type (4 Plies per 1 in. thick)	1.425	1.465	1.505	1.545	....	....
Corrugated Asbestos Type (8 Plies per 1 in. thick)	1.435	1.437	1.438	1.440	....	....
Laminated Asbestos Type (30-40 Laminations per 1 in. thick)	0.969	0.960	0.951	0.942	0.933	0.924
Laminated Asbestos Type (14-20 Laminations per 1 in. thick)	1.103	1.104	1.105	1.106	1.107	1.108
Mineral Wool Type	1.023	1.028	1.033	1.038	1.043	1.048
High Temperature Type (Diatomaceous Earth and Asbestos)	1.560	1.489	1.418	1.347	1.276	1.205
Brown Asbestos Type (Felted Fiber)	1.003	0.997	0.990	0.984	0.977	0.971

HEAT LOSSES FROM DUCTS

The thermal transmission coefficient U for an uninsulated metal duct can be obtained from the equation:

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o}} \qquad (4)$$

where

U = thermal transmittance, Btu per square foot per hour per degree Fahrenheit difference in temperature between the average temperature inside the duct and the air outside the duct.

f_i = film conductance inside the duct, Btu per hour per square foot per degree Fahrenheit.

f_o = film conductance outside the duct, Btu per hour per square foot per degree Fahrenheit.

Film conductance f_i for air flowing in ducts apparently depends only on the velocity of the air and the diameter of the duct. A fairly reliable inside coefficient can be calculated from Schultz's modified equation:

$$f_i = \frac{0.32 V_o^{0.8}}{D^{0.25}} \tag{5}$$

where

V_o = velocity of air in duct, feet per second.
D = diameter of duct, feet.

Film conductance f_o depends on a number of variables including temperature, diameter, and emissivity of the outer surface. Conductance f_o can be readily calculated from Tables 1, 2, 3, 4, and 5. From this explanation, it is seen that it is unwise to recommend a given value of U for all uninsulated metal ducts.

The heat loss from a given length of duct can be expressed by:

$$Q = UPL\left[\left(\frac{t_1 + t_2}{2}\right) - t_3\right] \tag{6}$$

The heat given up by the air in the duct is:

$$Q = 0.24\ M\ (t_1 - t_2) = 14.4\ AVd\ (t_1 - t_2) \tag{7}$$

Equating 6 and 7 enables the determination of the temperature drop in the duct:

$$\frac{t_1 + t_2 - 2t_3}{t_1 - t_2} = \frac{28.8\ AVd}{UPL}$$

Let $x = \frac{28.8\ AVd}{UPL}$ for rectangular ducts, $= \frac{7.2\ DVd}{UL}$ for round ducts, solving for t_1 and t_2:

$$t_1 = \frac{t_2\ (x + 1) - 2t_3}{(x - 1)} \tag{8}$$

$$t_2 = \frac{t_1\ (x - 1) + 2t_3}{x + 1} \tag{9}$$

For low velocities and long ducts of small cross-section, a somewhat more accurate formula may be used as follows:

$$t_2 = \frac{t_1 - t_3}{e^{\left(\frac{UPL}{14.4\ AdV}\right)}} + t_3 \tag{10}$$

In these equations

Q = heat loss through duct walls, Btu per hour.
U = thermal transmission coefficient, Btu per square foot per hour per degree Fahrenheit.
P = perimeter of duct, feet.
L = length of duct, feet.
t_1 = temperature of air entering duct, degree Fahrenheit.
t_2 = temperature of air leaving duct, degree Fahrenheit.
t_3 = temperature of air surrounding duct, degree Fahrenheit.
M = weight of air per hour, through the duct, pounds.
A = cross-sectional area of duct, feet.

Heat losses for insulated ducts are given in the warm air column of Table 15. The losses are based on a uniform series of material conductivities at 86 F mean temperature and an air temperature of 50 F outside of the duct. The losses may be interpolated for odd material conductivities and temperatures. The conductivities of various materials will be found in Table 2 of Chapter 3. For cases where the surrounding air temperature is other than 50 F, the losses may be selected on the basis of temperature difference.

Recently, a new prefabricated insulated duct built entirely of asbestos has been placed on the market.

Example 4. Determine the entering air temperature and heat loss for a duct 24 × 36 in. cross-section and 70 ft in length, insulated with ½ in. of a material having a conductivity of 0.35 Btu at 86 F mean temperature, carrying air at a velocity of 1200 fpm, measured at 70 F, to deliver air at 120 F with air surrounding the duct at 40 F.

Solution. Assume the entering air temperature to be 130 F. Thus, the mean temperature difference will be 85 F. Referring to the warm air column of Table 15 and interpolating for 85 F temperature difference, the overall heat transmission coefficient is found to be 0.516 Btu. From Table 6 Chapter 1 the density of air at 70 F and 29.92 in. Hg. is found to be 0.0749 lb per cubic foot. Substituting these and the other given values in Formula 8:

$$x = \frac{28.8 \times 6 \times 0.0749 \times 1200}{0.516 \times 10 \times 70} = 42.61$$

$$t_1 = \frac{120\ (42.61 + 1) - 80}{42.61 - 1} = 123.8$$

Based on 123.8 F entering air temperature, the new mean temperature difference will be 81.9 F and the new transmission coefficient will be 0.515. Resubstituting in Formula 8, t_1 becomes 123.9 F.

Substituting in formula 6:

$$Q = 0.515 \times 10 \times 70 \left[\left(\frac{123.9 + 120}{2} \right) - 40 \right]$$

$$Q = 29.543 \text{ Btu}$$

LOW TEMPERATURE INSULATION

Surfaces maintained at temperatures lower than the surrounding air are insulated to reduce the flow of heat and to prevent condensation and frost. The insulating material should absorb a minimum amount of moisture, for one reason that the absorption of moisture substantially increases the conductivity of the material. This property is particularly important in the case of surfaces to be insulated that are below the dew-point of the surrounding air. In such cases, due to vapor pressure difference, it is necessary to seal the surface of the insulating material against the penetration of water vapor which would condense within the material, causing a serious increase in heat flow, possible breakdown of the material and corrosion of metal surfaces. An insulating material with a high degree of moisture absorption might pick up moisture before application and then, when the seal is in place and the temperature of the insulated surface reduced, release that moisture to the cold surface.

The thickness of insulation required to prevent sweating is that thickness which will raise the temperature of the outer surface of the

insulation to a point slightly higher than the dew-point for the corresponding air temperature and relative humidity. The difference in temperature between the air and the dew-point for various humidities can be readily ascertained from a psychrometric chart.

The thickness of insulation required to prevent sweating for rectangular ducts will be slightly greater than the value obtained by solving the equation:

$$L = k\left[\frac{T_1 - T_3}{q} - \left(\frac{1}{f_o} + \frac{1}{f_i}\right)\right] \quad (11)$$

where

L = thickness of insulation for flat surfaces, inches.
T_1 = temperature of air in room, degrees Fahrenheit.
T_2 = temperature of surface of insulation, degrees Fahrenheit.
T_3 = average temperature of cooler air in duct, degrees Fahrenheit.
k = conductivity of insulation, Btu per hour per square foot per degree Fahrenheit per inch.
q = heat loss per square foot of outer surface of insulation, Btu per hour.
f_o and f_i have the same values as given in Formula 4.

In any practical case T_1 and T_3 will be known, and, for any assumed value of relative humidity $(T_1 - T_2)$ can be obtained from a psychrometric chart. Values of q and f_o can be obtained from Tables 1, 2, 3, and 4; f_i from Formula 5; and k from conductivity Table 2 of Chapter 3.

The approximate thickness of insulation used to prevent condensation on pipes and flat metallic surfaces may be obtained from Fig. 5. The maximum permissible temperature drop is indicated at the point where the guide line passes through the horizontal scale at the left center of the

TABLE 16. HEAT GAINS FOR INSULATED COLD PIPES

Rates of heat transmission given in Btu per hour per degree Fahrenheit temperature difference between fluid in pipe and surrounding still air

Based on materials having conductivity, k = 0.30

Nominal Pipe Size (Inches)	Ice Water Thickness			Brine Thickness			Heavy Brine Thickness		
	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
½	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087	0.394
¾	1.6	0.119	0.431	2.0	0.111	0.405	2.9	0.094	0.340
1	1.6	0.139	0.403	2.0	0.124	0.352	3.0	0.104	0.294
1¼	1.6	0.155	0.357	2.4	0.131	0.300	3.1	0.113	0.260
1½	1.5	0.174	0.351	2.5	0.134	0.270	3.2	0.118	0.238
2	1.5	0.200	0.322	2.5	0.151	0.244	3.3	0.134	0.214
2½	1.5	0.228	0.303	2.6	0.170	0.226	3.3	0.147	0.197
3	1.5	0.269	0.293	2.7	0.186	0.202	3.4	0.162	0.176
3½	1.5	0.295	0.282	2.9	0.191	0.183	3.5	0.176	0.167
4	1.7	0.294	0.248	2.9	0.209	0.176	3.7	0.182	0.154
5	1.7	0.349	0.239	3.0	0.241	0.165	3.9	0.202	0.138
6	1.7	0.404	0.233	3.0	0.259	0.150	4.0	0.228	0.130
8	1.9	0.455	0.201	3.0	0.318	0.140	4.0	0.263	0.116
10	1.9	0.559	0.198	3.0	0.383	0.135	4.0	0.309	0.110
12	1.9	0.648	0.194	3.0	0.438	0.131	4.0	0.364	0.108

chart. This temperature drop represents the difference between the dry-bulb temperature and the dew-point temperature for the conditions involved. (See discussion of Condensation in Chapter 3.) The surface resistances used for calculating the family of curves in Fig. 5 are based on tests made on canvas covered pipe insulation surfaces at *Mellon Institute*. However, it has been found that the resistance for asphaltic and roofing

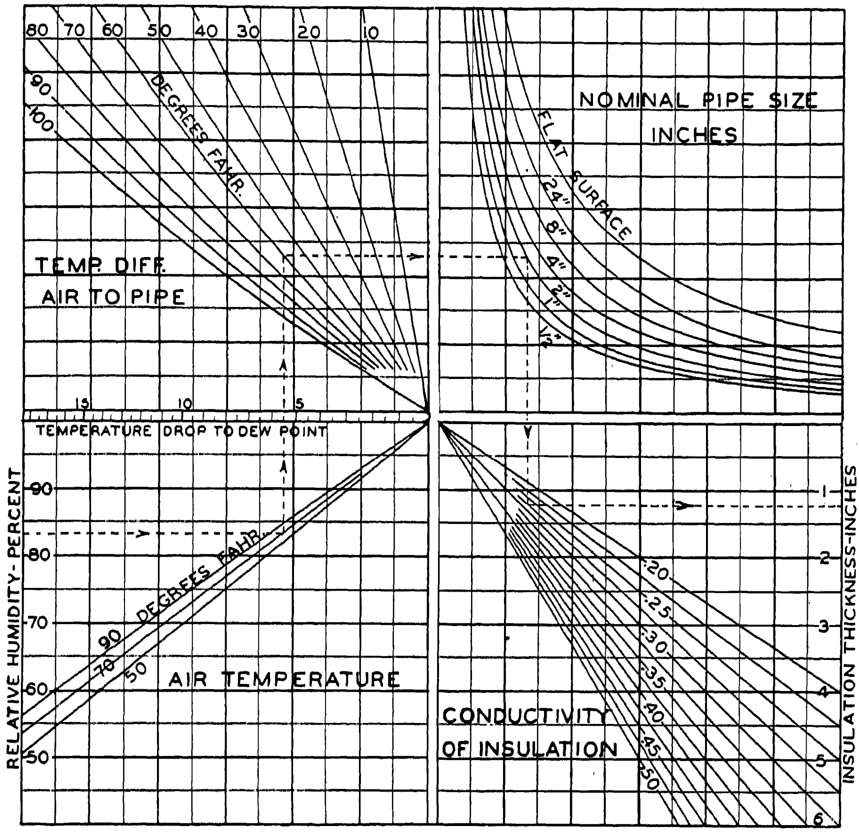


FIG. 5. THICKNESS OF PIPE INSULATION TO PREVENT SWEATING[a]

[a]Solve problems by drawing lines as indicated by dotted line, entering chart at lower left hand scale.

surfaces is practically the same as for canvas surfaces, so that the curves may be followed with no alteration for surfaces commonly used.

Heat gains for pipes insulated with a material having a conductivity of 0.30 Btu per square foot per hour per degree Fahrenheit difference per inch thickness are given in Table 16.

Heat gains for insulated ducts are given in the cold air column of Table 15. The heat gains are based on a uniform series of conductivities at 86 F mean temperature and an air temperature of 90 F outside of the duct. The gains may be interpolated for odd material conductivities and

temperatures. For cases where the surrounding air temperature is other than 90 F the gains may be selected on the basis of temperature difference.

INSULATION OF PIPES TO PREVENT FREEZING

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 17 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that an efficient insulation be utilized. This table is based on the use of a material having a conductivity of 0.30. The initial water temperature is assumed to be 10 F above, and the surrounding air temperature 50 F below the freezing point of water (temperature difference, 60 F).

The last column of Table 17 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 17. However, if the water enters the pipe at 34 F it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is − 38 F (temperature difference 80 F) instead of − 18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 17, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown

TABLE 17. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES WITH SURROUNDING AIR AT − 18 F

Nominal Pipe Size (Inches)	Number of Hours to Cool 42 F Water to Freezing Point			Water Flow Required at 42 F to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches (Conductivity, $k = 0.30$)					
	2	3	4	2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.45	1.93

in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

When water must remain stationary longer than the times designated in Table 17, the only safe way to insure against freezing is to install a steam or hot water line or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

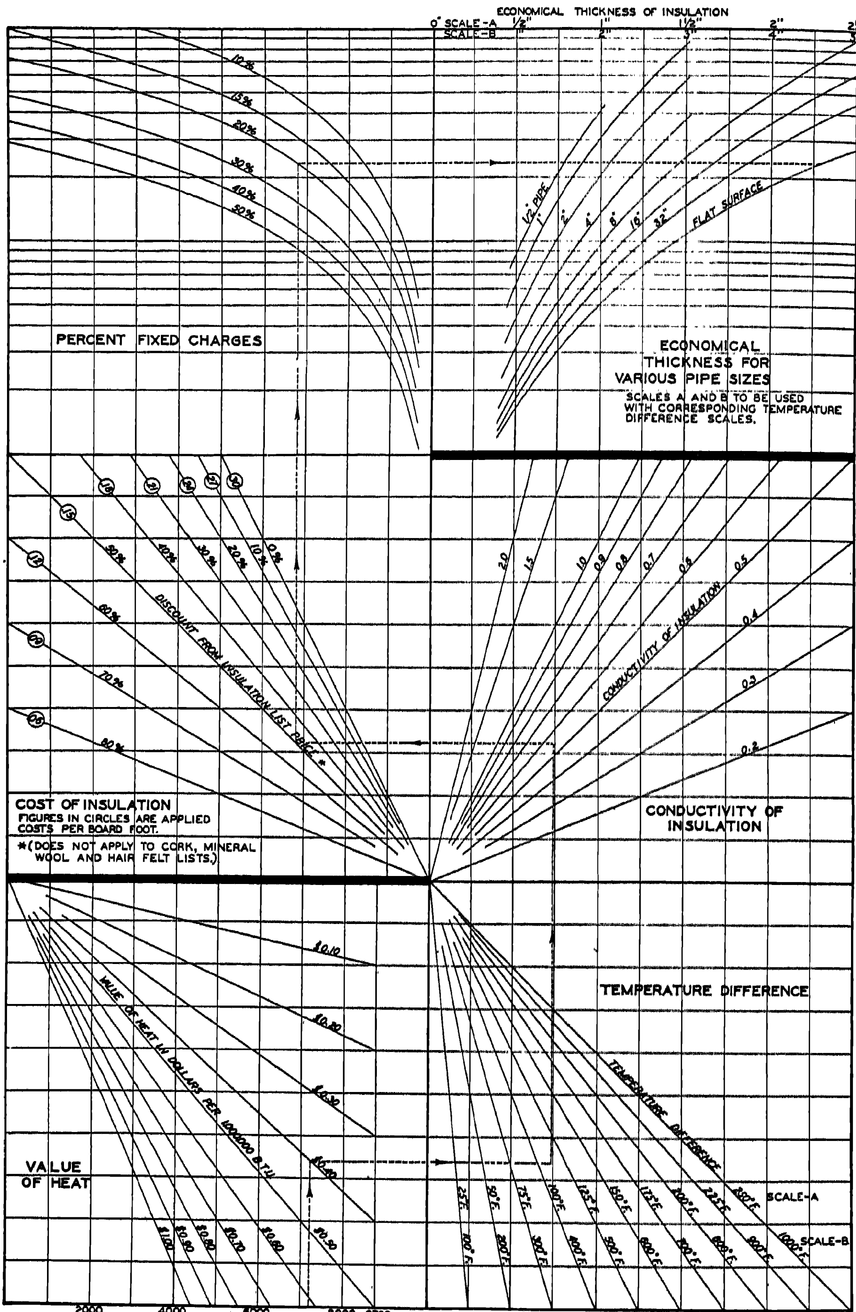
ECONOMICAL THICKNESS OF PIPE INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 18. Where a thorough analysis

TABLE 18. THICKNESSES OF INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

[a]All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.



(L. B. McMillan, *Proc. National Dist. Heating Assn.*, Vol. 18, p. 138).

FIG. 6. CHART FOR DETERMINING ECONOMICAL THICKNESS OF INSULATION

of economic thickness is desired this may be accomplished through the use of the chart, Fig. 6.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.

UNDERGROUND PIPE INSULATION

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes. (See Chapter 16). Detailed data on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[4].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

The economical thickness of insulation for underground work is difficult to determine accurately due to the many variables which have to be considered. As a result of theories[5] previously developed, together with

TABLE 19. THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In.	Pipes 4 In. to 10 In.	Pipes Larger than 12 In.	Pipes Less than 4 In.	Pipes 4 In. and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

[4]Handbook of the *National District Heating Association*, Second Edition, 1932.

[5]Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 335).

other experimental data which have been presented, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 19 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 6. The data in Fig. 6 are based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

Chapter 43

ELECTRIC HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Boilers, Electric Hot Water Heating, Heating Domestic Water Supply, Radiant Drying, Reversed Cycle Refrigeration, Auxiliary Electric Heating, Control, Calculating Capacities, Power Problems

ELECTRIC heating is steadily assuming a more important place in heating, ventilating and air conditioning installations, encouraged in many localities by reduced electric rates. Electric heating is flexible, clean, safe, convenient and easy to control. It has many basic principles in common with fuel heating, but there are also important differences. When heat is delivered by wire, no combustion process is necessary, either at a central plant or at the individual room units. The output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and it follows that the total load on an electric heating system is the total wattage of connected electric heaters, regardless of weather conditions. The main obstacle to the more general adoption of electric heating for buildings is the cost of the electricity itself.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first*, the conversion of energy into heat; *second*, the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, as 100 per cent of the energy applied to the resistor is always transformed into heat. In electric heating practice no concern need be given to efficiencies of heat production, but rather to efficiencies of heat utilization. The problem is to distribute the electrically produced heat units in such manner as to obtain conditions of maximum comfort with the minimum consumption of electricity.

DEFINITIONS

Definitions of general terms used in fuel heating are given in Chapter 46. Terms which apply particularly to electric heating are:

sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly upon the body. They are not satisfactory for general air heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. For a discussion of electrically heated panels as applied to radiant heating, see Chapter 44.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors. They generally have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from the floor line. The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water

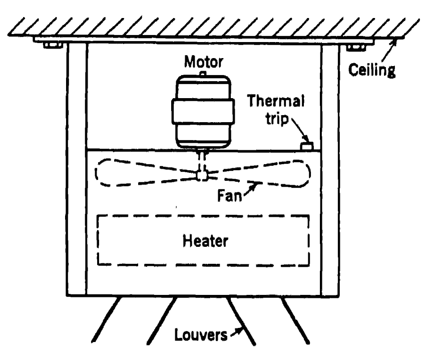


FIG. 1. CEILING MOUNTED UNIT HEATER

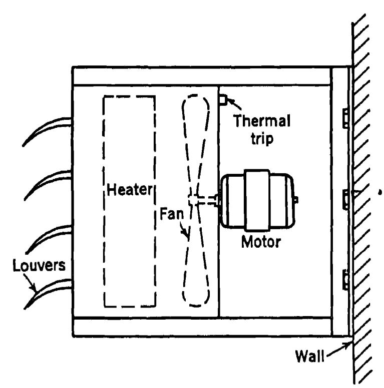


FIG. 2. WALL MOUNTED UNIT HEATER

and steam add nothing to the efficiency of an electric heater and entail expensive construction and maintenance.

UNIT HEATERS

Electric unit heaters include a built-in fan unit which circulates room air over the heating elements. Heaters of this type are manufactured in many designs and sizes, and can be located in the same manner as steam unit heaters.

Electric unit heaters are used in industrial plants, sub-stations, power houses, pumping stations, etc., where the power rate for electric heating is found to be favorable. In many large plants, such as flour mills, grain elevators, etc., in which there are a number of small offices, locker rooms, etc., scattered over wide areas, electric unit heaters are frequently economical in such locations. In small unattended stations, where freezing temperatures cannot be permitted, thermostatically-controlled electric unit heaters are frequently used to maintain a temperature above freezing. The best location for the heaters depends upon local circumstances as they can be mounted either on the ceiling to direct the air

downward, on the side wall about 7 ft from the floor, or near the floor line. Variations in design are necessary for different locations, but typical arrangements are indicated in Figs. 1 and 2.

The arrangement of the wiring circuits is very important for electric unit heaters. In principle they are all the same and include as essential elements an automatic control panel, a thermostat, and a master hand switch. All heaters should be designed with a safety thermal trip wired in series with the magnet coil of the control panel and with the hand switch and thermostat. A typical wiring diagram is shown in Fig. 3. This applies to a single phase power supply, but for 3 phase the only difference is to have a 3-pole panel and a heater arrangement for 3-phase connection.

Portable unit heaters are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction.

CENTRAL FAN HEATING

Electric heating elements can be used for the prime source of heat in a central fan electric heating system or in the heating phase of a complete

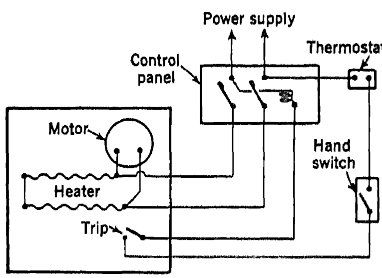


FIG. 3. WIRING DIAGRAM FOR UNIT HEATER

air conditioning system. They can be used in the same manner as steam heating units for tempering, preheating or reheating the air at the main supply fan location and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning electric heating elements can be used to provide moisture by the evaporation of water, or for controlling air washer dew-point temperatures when mounted as preheating units on the intake side of the air washer. (See Chapter 20.)

In coordinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature source of heat for any given pressure and a change in air volume flowing over steam coils does not greatly affect the temperatures of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, having a constant input of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered.

This occurs because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant unless some compensating action is performed by control. Automatic variation of the electrical heat input synchronized properly with the air flow can be successfully accomplished by various special methods of control.

Electric heaters are useful in balancing the heat distribution in central fan heating systems. Even in those instances where steam is the principal heat source, the temperature of individual rooms can be controlled locally by separate electric booster heaters. These heaters can be installed in branch ducts or behind the air outlet grilles in each room. With this arrangement, the central heating unit distributes air at an average temperature, controlled from a thermostat centrally located, such as in the main return duct. The electric booster heaters may be controlled by thermo-

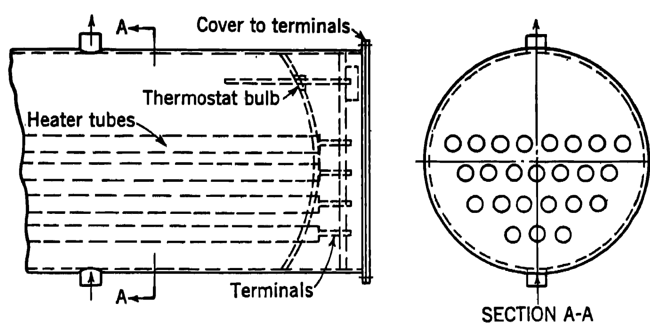


FIG. 4. RESISTANCE TYPE BOILER FOR STEAM OR HOT WATER

stats mounted in each individual room which permit the occupant to maintain any desired temperature independent of the rest of the building.

ELECTRIC BOILERS

Steam or hot water generating boilers using electric energy are entirely automatic and are well adapted to intermittent operation. Small electric boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used either with direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood of the heating elements burning out, they should be of substantial construction, with a low heat density per unit of surface area and provision should be made for cleaning off desposits of scale which restrict the heat flow. A typical resistance type of steam or hot water boiler is shown in Fig. 4.

Large electric boilers are usually of the type employing water as the resistor, using immersed electrodes. With this type only alternating current can be used, as direct current would cause electrolytic deterioration. Such a type of electrode boiler is shown in Fig. 5.

Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and also for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical to shut down the main plant fuel burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation. In general, electric steam heating is confined to auxiliary or other limited applications. If the heating system is designed to use electricity exclusively, steam generating or distributing equipment is superfluous.

ELECTRIC HOT WATER HEATING

Electric water heating, using an electric boiler in place of a fuel burning boiler, like electric steam heating, is generally confined to auxiliary or

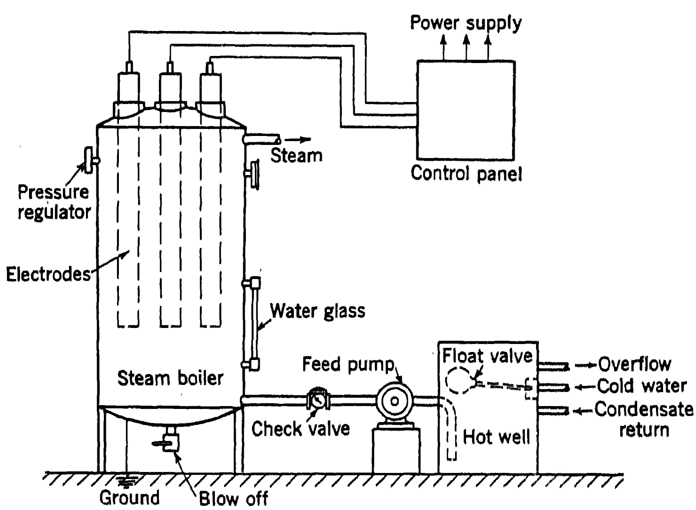


Fig. 5. Diagrammatic Arrangement of an Electrode Boiler

other limited applications. The use of insulated water storage tanks, in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which has some special applications.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with electric heating elements and automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the building may be of any standard individual radiator or fan-served indirect type or with provisions for the heating and humidification phases of an air conditioning system. A system of this kind requires very careful design to avoid excessive overall radiation losses during periods of low heat demand. It is also important to provide for sudden changes in heat demand. A typical hot water heating boiler is illustrated in Fig. 4.

HEATING DOMESTIC WATER BY ELECTRICITY[1]

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable. In many sections of the country low electric rates have been established by the electric utilities to secure this load. In some localities, electric rate schedules divide the current used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to carry over the periods when the off-peak element is timed out, without too frequent demands on the regular heating element which takes the higher domestic lighting service rate. Some utilities now offer

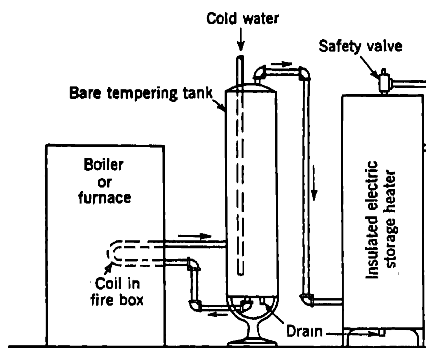


FIG. 6. PIPING ARRANGEMENT FOR CONNECTING ELECTRIC WATER HEATER TO FIRE-BOX COIL

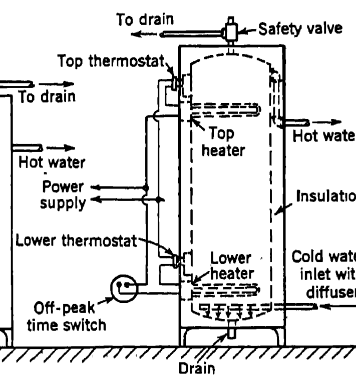


FIG. 7. DOMESTIC HOT WATER HEATER FOR OFF-PEAK SERVICE

a schedule which, beyond a stipulated minimum, lowers the rate for all electric service if an electric water heater is installed.

Competition with other fuels, especially gas, seems to be the major controlling factor in the use of electricity. The first cost of electric storage heaters is also greater than for gas, owing to the need for larger tank storage due to off-peak service and slower recuperating capacity.

In residential work, to effect a saving in the cost of operation, it is sometimes desirable to use a furnace coil or indirect heater in connection with an electric water heater. In this case it is important to make the proper connections in order to benefit by any heat obtained from the furnace and at the same time to prevent dangerous overheating. The proper piping connections are shown in Fig. 6, and in this case the electric heater will only furnish heat when insufficient heat is supplied from the furnace. This arrangement has a further advantage in the summertime in that the bare tank through which the cold water passes on its way to

[1]Test Results of Electric Water Heaters, by C. G. Hillier (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1936, p. 632). Fourteenth Range and Water Heater Survey (*Electric Light and Power*, August, 1940).

the electric heater serves as a tempering tank, absorbing heat from the basement air and requiring the use of less energy in the electric heater.

A typical domestic hot water heater as shown in Fig. 7 is arranged with upper and lower heating elements for the usual type of off-peak heating service. The lower heating element is under the control of the off-peak time switch. However, the upper heating element is usually connected to the line so that in case the supply of hot water in the tank becomes exhausted the top thermostat can turn on the top heater and heat a small supply of water. The top heater will not heat the water in the tank below its location, but when the off-peak period arrives the lower heater is turned on and the entire tank becomes heated.

RADIANT DRYING

Lacquers and similar surface films can be very effectively dried by radiation. Special electric lamp bulbs have been developed which give off a high percentage of infra-red and similar heat rays[2]. These are mounted in very efficient reflectors. For continuous manufacturing processes these reflectors are mounted in tunnels through which conveyors pass. For local applications, as for example paint drying in automobile repair shops, they may be mounted on portable racks.

In the application of this type of drying the composition of the paint or lacquer is important. In general, lacquers and those enamels using synthetic resins react most favorably. Other applications include the drying of ink, glue, and water, the softening of celluloid and bakelite for punching or shearing, and a wide variety of other uses.

REVERSED CYCLE REFRIGERATION

Reversed refrigeration is frequently referred to as a *heat pump* since the electric motor driving the refrigerating compressor furnishes the motive power to transfer heat from one temperature to a higher temperature level. The compressor acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and discharge it outdoors.

In normal use a refrigerating machine is arranged to remove heat and the heat removed is dissipated to the condenser cooling water. The driving energy is converted into heat, most of which is added to the heat removed and extracted. In so-called reversed refrigeration the heat removed together with the heat converted from the driving energy is utilized to heat the building. This conservation of the heat converted from the driving energy enables the reversed refrigeration to show a better performance in heating service than straight refrigeration can show in cooling service. In order to overcome the drop in capacity and in efficiency with lower outside temperatures, it is often desirable to use well-water instead of air as the source of heat. For a detailed description of this cycle see Chapter 24.

---

[2]Infra-Red Lamps Speed Up Drying Operations (*Automotive Industries* 82:376-7; April 15, 1940). Invisible Rays Build Visible Profits, by H. M. Archer (*Electric Light & Power*, May, 1940). Radiant Energy Drying and Baking for Organic Finishing (*Metal Industry* 38:294-6; May, 1940).

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electric heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a building comfortable. Likewise, such electric heating might be used on abnormally cold days to help out the main heating system and by this means reduce the necessary size of the system.

A few installations have been made using electric heating cable buried in the floors of bathrooms, etc., to provide auxiliary electric heating. At least one airplane hangar is heated in this manner.

Because of the feeling of comfort that a radiant type heater gives, bathrooms may be heated electrically with this type of heater while the rest of the house is cared for by some other system. Offices and rooms which require heat at periods when the main heating plant is shut down can be conveniently heated electrically.

CONTROL

Because the efficiency of electric heat production is the same for small and large units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Heaters are often controlled manually but thermostatic control is essential for economical operation. For duct systems having a variable volume of air flow the electric heater control must automatically vary the heat input in coordination with the changes in air volume and demand for heat.

CALCULATING CAPACITIES

The electric heating capacity required can be calculated from the heat requirement in Btu per hour by using the equation:

$$\frac{\text{Btu per hour}}{3413} = \text{kw rating of required electric heating} \qquad (1)$$

For comparison with steam radiation:

$$1 \text{ kw} = \frac{3413 \text{ Btu}}{240} = 14.2 \text{ sq ft of steam radiation} \qquad (2)$$

POWER PROBLEMS

The cost of electric energy varies because of several factors. Distribution costs differ for large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate electric plants at uniform loads; hence, even the time of use may affect the cost of electricity. Special low rates are sometimes available during certain prescribed hours of use.

Since the cost of production and distribution depends not only upon the quantity of energy used but also upon the maximum rate at which it is used, electric energy is often sold on a demand rate basis. In some

cases, the demand charge is based upon the rated connected load, in other cases, upon the maximum demand as indicated by a demand meter.

Homes are almost universally supplied with lighting current of 115 volts, which can only be used economically for small heaters. Usually the service lines will not permit more than plug-in devices. The Underwriters permit approved heaters of 1320 watts or less to be plugged into approved baseboard receptacles, but such heaters cannot be served on a circuit supplying much other load without overloading the fuses. There is an increasing trend toward supplying homes with three wire 115/230 volt service. Where homes have such service, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 220, 440 or 550 volts. All polyphase heaters should be balanced between phases.

REFERENCES

Electric Elements Well Adapted to the Air Conditioning Heating Cycle, by L. P. Hynes (*Heating, Piping and Air Conditioning*, January, 1940, p. 29).

Electric Air Heating Market (*Electrical West* 82:113; June, 1939).

Seven Jobs for Electric Heat (*Factory Management* 97:80-1; September, 1939).

Electric Heat for the Bathroom Floor (*Heating & Ventilating* 36:41; December, 1939).

Electric Heat Spreads in Sunny Climes (*Barron's National Financial Weekly* 20:5; January 8, 1940).

Electric Heating for Los Angeles Building (*Heating & Ventilating* 37:50-1; June, 1940).

Dracker Apartments, West Los Angeles, Calif. (*Heating & Ventilating* 37:32; July, 1940).

Electric Radiant Heat vs. Steam Convection (*Electrical World* 114:395-6; August 10, 1940).

Chapter 44

RADIANT HEATING

Physical and Physiological Factors, Control of Heat Losses, Rate of Heat Production, British Equivalent Temperature, Application Methods, Calculation Principles, Mean Radiant Temperature, Measurement of Radiant Heating

FOR health and comfort, the rate of heat loss from the human body must be controlled, by the aggregate effect of the conditions surrounding the body, so that the physiological reactions result in a feeling of comfort. Generally, any heating system serves not to add heat to the individual, but to reduce the net rate at which the body loses heat by radiation, convection and evaporation. In convection heating, the heating medium serves to maintain such an air temperature as will give comfort, under existing conditions of humidity and of surrounding surface temperatures. The primary object of radiant heating, on the other hand, is to maintain such an average temperature of the surrounding surfaces, as will give comfort without needlessly heating the air. The difference between convection heating and radiant heating is therefore partly physical and partly physiological.

On a cool spring day while standing in the sunshine, one may feel perfectly comfortable, but when a cloud passes over the sun, one will feel much cooler. A shielded thermometer will show no immediate reduction in air temperature, so one actually feels a cooling effect which an ordinary thermometer cannot register. This is because light and heat waves travel at the same speed and are both interrupted by a cloud, or other shield. This proves that radiant heat affects the comfort of the body as definitely as does air temperature.

Comfort requires that heat escape from the body at the same rate as it is generated, by the oxidation of food-stuffs in the body, and in a manner suitable to physiological requirements. Furthermore, the surrounding conditions, themselves, cause changes both in the rate of heat generation in the body, and in the operation of the several methods by which the body loses heat. The feeling of heat or cold results not only from the rate at which the body loses heat, but also from the manner in which the heat is abstracted from the body, and the ease with which the body's heat regulating mechanisms can operate.

CONTROL OF HEAT LOSSES

Heat is transferred from any warm dry object to cooler surroundings principally by convection and by radiation; the total loss is substantially the sum of these two. Where the surface is moist, as with the human

will result. This leaves about 110 Btu per hour to be lost by convection (or 5.65 Btu per hour per square foot of convecting body surface).

The mean surface temperature of the human body, including the whole area not only of exposed skin but also of clothing and hair, has been estimated variously at from 75 F (particularly in England) up to 83 F (in America). Further research and experience will be needed to finally derive the most suitable value for the American climate. The final figures will vary with sex, age, clothing, etc., but will probably come between these extremes.

The mean surface temperature of an inert body, which will cause given rates of heat loss by radiation and by convection in a uniform environment, having a given air temperature and a given mean wall temperature, may be calculated from fundamental equations for radiation and natural convection, but substituting comparable cylinders for the irregular human body.

Heilman[2] gives the following equations:

$$H_r = 0.1723\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \tag{1}$$

$$H_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \tag{2}$$

where

H_r = heat loss by radiation, Btu per square foot per hour.
H_c = heat loss by convection, Btu per square foot per hour.
T_s = absolute temperature of the body surface, degrees Fahrenheit.
T_w = absolute temperature of the walls, degrees Fahrenheit.
T_a = absolute temperature of the air, degrees Fahrenheit.
$T_m = \dfrac{T_s + T_a}{2}$
D = diameter of cylinder, inches.
e = the ratio of actual emission to black body emission.

If it is assumed that an average adult has a height of 5 ft 8 in. and a body surface of 19.5 sq ft for convection, and 15.5 sq ft for radiation, an equivalent effect can be worked out for two cylinders, 5 ft 8 in. high by 13.15 in. diameter and 10.45 in. diameter, respectively. However, while the effects on a cylinder (of a particular size and shape) may be used to estimate average similar effects on the human body, it should be remembered that the heat loss from the body varies greatly. Every movement alters not only its shape, but also the velocity of the air passing over it and the surface exposed to radiation. This fact renders the results of any such computation only approximate.

BRITISH EQUIVALENT TEMPERATURE

The British Equivalent Temperature (BET) is the mean temperature of the entire environment which is effective in controlling the rate of sensible heat loss from a *black body* in still air when this body has a surface temperature equal to that of the human body, and a size comparable

[2]Surface Heat Transmission, by R. H. Heilman (*A.S.M.E. Transactions, Fuels and Steam Power Section.* Vol. 51, No 22, September-December, 1929).

to the human body. The BET is, therefore, a function of both the air temperature and the mean radiant temperature of the surrounding objects. Its numerical value in a uniform environment (walls and air at the same temperature) is equal to the temperature of the walls and air. In a non-uniform environment (walls and air at different temperatures), the BET for America is at present considered to be equivalent to that of a uniform environment in which a body with an 83 F surface temperature will lose sensible heat at the same rate as in the given non-uniform environment. As originally defined (in England) the BET was based on a body surface temperature of 75 F, but 83 F has been accepted as more nearly conforming with American practice[3]. The most suitable temperatures depend in part on the clothes worn by the individual, which explains why ladies in evening dress require a higher BET, for comfort, than a man having only hands and head uncovered. The higher the BET, the less the heat loss from the body, as the rate of loss in still air is approximately proportional to the difference between the BET and the mean body surface temperature.

If the BET were 83 F, there could be no sensible heat loss from a surface at that temperature; so the temperature of a normal body surface would have to rise to a point where the heat generated in the tissues could be dissipated. With a BET of about 65 to 70 F, the sensible heat losses from the assumed average individual will approximate those stated on page 760. Actual experience in the United States has shown that with about half of the ceiling surface heated to about 115 F in midwinter weather, it is practical to maintain comfort with air temperatures of 65 F or less.

APPLICATION METHODS

There are several methods of applying radiant heating, as follows:

1. *By warming the interior surfaces of the building.* Pipe coils are embedded in the concrete or plaster of the walls or ceilings, the heating medium being hot water circulating through the pipe coils. These coils are generally constructed of small pipe spaced about 6 in. apart (Fig. 1). This has the effect of warming the entire concrete or plaster surface in which the pipes are embedded. Since the temperature of the heating medium should never exceed about 140 F (due to the possibility of cracking the plaster) the area of the panel must be sufficient to supply the requisite quantity of heat at this low temperature. When carefully designed, this method produces very comfortable results and great operating economy, but offers some slight obstacles when alterations or additions to the building are desirable. Normally the hot water circulation is maintained by means of a circulating pump and facilities have to be provided to eliminate all air at the top of the system. All of the pipes are welded together and tested after erection to a hydraulic pressure of 300 lb per square inch.

2. *By placing hot water or steam pipes under the floor.* With this arrangement the whole floor surface of a room is raised to a temperature sufficient to give comfortable conditions. This method is used for schools and hospitals where large quantities of outside air are desirable (Fig. 2). In some cases special floors are constructed in sections so that a whole floor can be lifted to examine the pipes. The floor surface may be of concrete, wood blocks, marble or any other material unaffected by heat. Pipes under the floor may be larger than those embedded in the plaster walls and ceilings.

3. *By circulating warm air through shallow ducts under the floor.* In this design the entire floor surface of a room is heated as in method 2. This method was used occasion-

[3]A.S.H.V.E. RESEARCH REPORT No. 962—Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperatures, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303).

ally in the Roman Empire, and while being more expensive in construction, is effective and quite suitable for cathedrals and large public buildings (Fig. 3). To provide a uniform floor temperature, special consideration should be given to the design of the air ducts so that equal heat distribution is obtained.

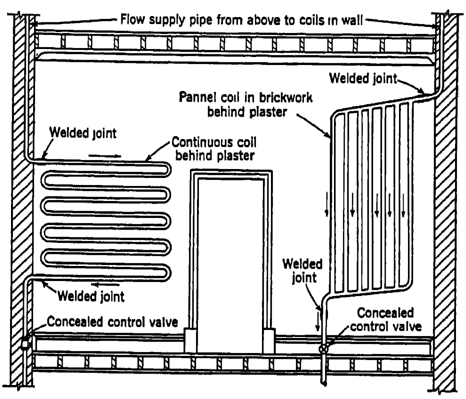


FIG. 1. PIPE COILS LOCATED IN INTERIOR WALL SURFACES

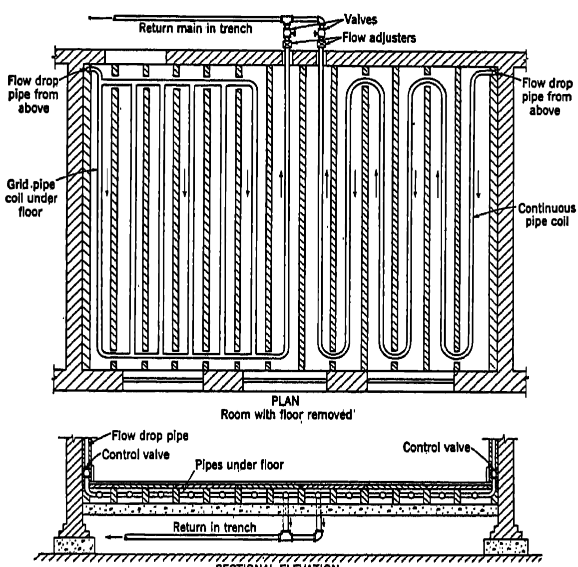


FIG. 2. ARRANGEMENT OF CONTINUOUS PIPE COIL IN FLOOR CONSTRUCTION

4. *By attaching separate heated metal plates or panels to the interior surfaces.* These plates or panels are placed either in an insulated recess so that the surface of the panel is flush with the surface of the walls or ceilings, or they may be secured to the face of the wall. They may be covered with wood veneers and decorated to harmonize with other parts of the room, or they can be cast into panels to imitate oak or other wood designs.

With flat plate panels it is common practice to use a frame of plaster, wood, metal or composition to allow for expansion. These plates may be heated with either hot water or steam and connected to an ordinary radiator system.

5. *By electric heated metal plates or panels.* These plates or panels are either placed in insulated recesses of walls or ceilings or fastened to the construction, as found desirable. They should not have a surface temperature much above 200 F; some have a much higher surface temperature but a lower temperature gives a more comfortable condition and is more efficient.

6. *By electrically heated tapestry mounted on screens and on the wall.* For this purpose the screen is woven with an electric continuous conductor. Such screens are useful to plug in at any position for emergency local heating without taking care of a large room or office.

Note. If all of a heating panel is installed at one end of a large room there may be a marked difference between the BET on the two sides of the body. It is usually desirable, therefore, that the heat be distributed at different parts of the walls and ceilings so that no uncomfortable effects will be felt from unequal heating.

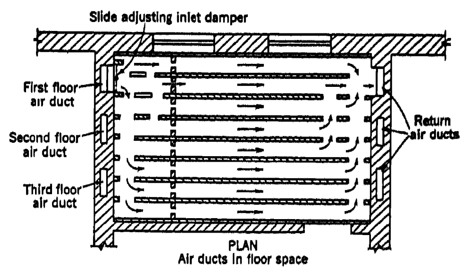


FIG. 3. DIAGRAM OF AIR DUCTS FOR FLOOR HEATING

CALCULATION PRINCIPLES

The calculations for radiant heating are entirely different from those for convective heating. The purpose of the latter is to determine, and compensate for the rate of heat loss from the room, when maintained in the desired condition; but radiant heating involves the regulation of the rate of heat loss from the human body.

The first step in the calculations for radiant heating of a given room is to ascertain the desired MRT; next, to decide at what temperature the heating surface shall operate; then, to compute the size and disposition of the heating surfaces required to produce this MRT; and last, to provide convected heat for the required number of air changes.

Mean Radiant Temperature

If the entire interior surface of a room were at the same temperature, this would be the MRT. Such a condition seldom exists, however, since in different parts of a room, with some surfaces exposed to the outer air, the actual surface temperature varies greatly with the construction and exposure of different sides of the enclosure. It is therefore necessary to calculate the thermal mean of these interior surface temperatures.

This is not the arithmetic average of the various actual surface temperatures, but the radiant temperature which corresponds to the average

of the several rates of heat emission (Btu per square foot) from the several surfaces. The emission at any given surface temperature, for any stated emissivity factor, and also the MRT corresponding to any average emission, can be obtained directly from Table 1. For example, if the emissivity of the surface is 0.9, 1 sq ft of surface at 50 F will emit 104.9 Btu per square foot per hour to surroundings at absolute zero.

TABLE 1. TOTAL BLACK BODY RADIATION TO SURROUNDINGS AT ABSOLUTE ZERO[a]

Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1.00	e 0.95	e 0.90	e 0.80		e 1.00	e 0.95	e 0.90	e 0.80
30	99.3	94.3	89.4	79.4	71	136.5	129.6	122.9	109.3
35	103.5	98.3	93.2	82.8	72	137.4	130.5	123.6	109.9
40	107.6	102.4	96.8	86.1	73	138.4	131.5	124.5	110.6
45	112.1	106.5	100.9	89.7	74	139.6	132.6	125.6	111.7
46	112.9	107.3	101.6	90.4	75	141.0	133.9	126.9	112.8
47	113.9	108.2	102.5	91.1	80	146.6	139.4	132.0	117.4
48	114.8	109.1	103.4	91.9	85	152.3	144.6	137.1	121.9
49	115.6	109.9	104.1	92.4	90	157.9	149.9	142.1	126.4
50	116.5	110.6	104.9	93.2	100	169.6	161.1	152.6	135.7
51	117.5	111.6	105.8	94.0	110	181.6	172.5	163.5	145.4
52	118.4	112.5	106.5	94.7	120	194.8	185.0	175.4	155.9
53	119.4	113.4	107.4	95.5	130	210.1	199.6	189.1	168.1
54	120.2	114.2	108.2	96.2	140	223.2	212.1	201.0	178.5
55	121.1	115.1	109.0	96.9	150	237.1	225.2	213.5	189.7
56	122.1	116.0	109.9	97.7	160	251.1	238.8	226.0	201.0
57	123.1	117.0	110.9	98.5	170	270.5	257.0	243.5	216.4
58	124.0	117.8	111.6	99.2	180	288.0	273.8	259.1	230.4
59	124.9	118.6	112.4	99.9	190	306.5	291.0	275.8	245.1
60	125.8	119.5	113.4	100.7	200	325.2	309.0	292.8	260.3
61	126.6	120.3	114.0	101.4	210	348.0	330.6	313.1	278.4
62	127.7	121.4	114.9	102.2	220	371.5	353.0	334.4	297.1
63	128.6	122.2	115.8	102.9	250	437.8	415.9	394.0	350.2
64	129.6	123.1	116.7	103.7	300	575.0	546.1	517.5	460.0
65	130.5	124.0	117.5	104.4	350	740.0	703.0	666.0	592.0
66	131.6	125.0	118.4	105.4	400	942.1	895.0	847.5	753.5
67	132.5	125.9	119.3	106.0	450	1176.0	1117.0	1059.0	941.0
68	133.5	126.8	120.1	106.8	500	1464.0	1390.0	1318.0	1171.0
69	134.5	127.8	121.1	107.6	550	1791.0	1701.0	1613.0	1434.0
70	135.5	128.8	121.9	108.4	600	2405.0	2284.0	2165.0	1925.0

[a]These factors are calculated from the formula

$$Q = e\left(\frac{0.1723 \times T^4}{100{,}000{,}000}\right)$$

where

Q = total black body radiation, Btu per square foot per hour.
e = emissivity.
T = absolute temperature, degrees Fahrenheit.

Such a determination of the amount of radiant heating surface needed in a room (to maintain a desired MRT), requires knowledge of the type of heating, and the surface temperatures of the unheated surfaces, which latter can only be estimated—but with a considerable degree of accuracy after some experience.

Detailed Computation Method

The area in square feet of each type or temperature of surface, horizontal or vertical, is multiplied by the emission value corresponding to its actual surface temperature. These products are added together to give the total radiant heat effect, inside the room from all surfaces.

The difference between the actual average radiant heat effect and 142 Btu per hour per square foot (the radiation from a surface at 83 F, with an emissivity of 0.95) is the Btu per hour per square foot which would be lost by radiation from a body at 83 F. If the rate, at which it is desired that heat be lost from the body by radiation, be assumed, the mean radiant emission from the walls required to give this desired rate can be determined from Table 1; and multiplying by the total surrounding area will give the desired total radiant heat effect.

The difference between the desired and the actual total radiant emission represents the additional heating effect which must be supplied by the hot surfaces to be installed. The temperature of the proposed hot surface must then be selected, and its emission per square foot at that temperature determined from Table 1. The difference between this emission, and that of the unheated surface, is divided into the total amount of additional heat needed, and the quotient will be the area of the required heating surfaces.

It is evident that this method of calculation depends for its accuracy on a correct estimate of the ultimate surface temperatures naturally attained by the actual wall, window, ceiling and floor surfaces.

Unless positive ventilation with tempered air is provided, the amount of heat given off by convection from the same heating surfaces should also be determined, checked against the tempering requirements of whatever outside air can in any way enter the room, and supplemented if necessary, by additional convection heating, to maintain the desired air temperature. As this air temperature will usually be from 6 to 8 F lower (for comfort) than with purely convective heating, the relative humidity will be appreciably higher at a given dew-point, and there will be a marked saving in fuel for tempering, humidification and conductive heat loss from the building.

Example 1. The surface areas, temperatures, and emissions for a room having a volume of 5760 cu ft are given in Table 2. The figures for temperatures are fairly

TABLE 2. SURFACE AREAS, TEMPERATURES AND EMISSIONS FOR A ROOM OF 5760 CU FT

	Area Sq Ft	Assumed Surface Temperature (Deg Fahr)	Heat Emission (Btu Per Sq Ft per Hour)	Total Heat Emission from Area (Btu per Hour)
External Wall	297	50	110.6	32,850
Glass	279	45	106.5	29,710
Inner Wall	480	55	115.1	55,250
Ceiling	480	55	115.1	55,250
Floor	480	55	115.1	55,250
Total	2016			228,310

representative of American practice with well-built walls, and the heat emission is based on an emissivity of 0.95 which approximates that of most paints and building materials.

The mean radiant emission of the room is 228,310 ÷ 2016 = 113.2 Btu per square foot per hour which, as seen from Table 1 corresponds to an MRT of 53 F for an average emissivity of 0.95.

For an average individual, having a surface area of 15.5 sq ft with an average surface temperature of 83 F, the heat given off by radiation (calculated by means of Equation 1) is 217 Btu per hour, or 14 Btu per square foot per hour. This corresponds to an environmental emission of 142 (the total radiation corrected to 83 deg) −14 = 128 Btu per square foot per hour, and (according to Table 1) to an MRT of 69.2 F, both of which are higher than the actual values available in the room.

In order to determine the amount of radiating surface necessary to maintain the MRT at 69.2 F, ceiling panels are assumed with a surface temperature of 160 F, which is approximately the mean temperature for metal plates heated by hot water.

The 2016 sq ft total area of the surfaces of the room, multiplied by 128 (the emission in Btu per square feet per hour, necessary to balance a body surface temperature of 83 F) gives a total desired emission of 258,048 Btu per hour. Enough radiant heating surface must be installed to increase the total radiant heat emission from 228,310 Btu per hour to 258,048 Btu per hour. The difference between these figures is 29,738 Btu per hour. From Table 1, the emission per square foot at 160 F is 238.8 Btu per square foot, or 123.7 Btu per square foot more than for the unheated ceiling surface (at 55 F). The required radiant heating surface is, therefore, 29,738 ÷ 123.7 = 240.4 sq ft. This surface, suitably placed, would raise the MRT to the degree required for comfort, and maintain it at that value.

Such calculations may be simplified, by preparing tables showing at the usual temperatures the area of hot surface required to bring each square foot of actual wall or other surface up to one or more desired standard MRT's.

MEASUREMENT OF RADIANT HEATING

Convection heating, intended to maintain a given air temperature, is best measured by thermometric methods, which indicate the air temperature, and not the rate of heat loss from the human body. Radiant heating, on the other hand, aims to control this rate of heat loss and can be measured only by calorimetric methods, that is, by determining directly the rate of heat loss from some object, maintained at the surface temperature of the body, irrespective of air temperature. Although a definite BET is needed, the MRT and the air temperatures may both vary in opposite directions provided the sensible heat loss by radiation and convection from a surface at 83 F is maintained constant within reasonable limits.

The apparatus for this purpose consists essentially of a hollow sphere or cylinder, maintained at the accepted mean surface temperature of the human body, together with an accurate means (usually electrical) of measuring the varying rate of heat supply required to maintain this exact temperature. This instrument, the *eupatheoscope*, is readily adapted to function like a thermostat so as to turn heat on or off, when the desired temperature of 83 F, or any other predetermined surface temperature of the vessel, decreases or increases as a result of changes in the BET.

Another instrument, at present available only for British practice as it is designed for a surface temperature of 75 F, consists of a blackened copper sphere of 6 in. diameter, in which a cylindrical sump contains a volatile liquid. A small electric heating coil creates in the sphere a

vapor pressure which is constant, as long as the total heat loss from the sphere is of the desired rate. If the BET becomes too high for comfort, a greater vapor pressure results from the smaller heat loss from the sphere, acts on a diaphragm, and turns down the supply of heat to the room. With falling BET, the reverse action occurs.

For testing work, the *globe thermometer* is a very useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 to 9 in. in diameter, usually made of thin copper and painted black. The temperature thus recorded is termed the *radiation-convection temperature.*

REFERENCES

Panel Warming, by L. J. Fowler (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 287).

Room Warming by Radiation, by A. H. Barker (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 331).

What Will be the Future Development of Heating and Air Conditioning, by W. H. Carrier (*Heating, Piping and Air Conditioning,* January, 1933, p. 16).

American Practice in Panel Heating, by L. L. Munier (*Heating, Piping and Air Conditioning,* June, 1937, p. 424).

Radiant Heating and Cooling, by F. E. Giesecke (*Heating, Piping and Air Conditioning,* June, July, August, September and October, 1940).

Panel Heating in the British Embassy, by Alfred L. Jaros, Jr. and Richard A. Wolff (*Heating and Ventilating,* May and June, 1930).

Calculations for Radiant Heating, by T. Napier Adlam (*Heating and Ventilating,* October, 1931).

Notes on Electric Warming with Special Reference to Low Temperature Panel Systems, by R. Grierson (*Proceedings of The Institution of Heating and Ventilating Engineers,* London, Vol. 28, 1929).

Notes on the Theory of Radiant Heating, by C. G. Heys Hallett (*Proceedings of The Institution of Heating and Ventilating Engineers,* London, Vol. 29, 1930).

Radiant Heat, by A. F. Dufton (*Proceedings of The Institution of Heating and Ventilating Engineers,* London, Vol. 30, 1931).

Panel Heating, by C. M. Oates (*Proceedings of The Institution of Heating and Ventilating Engineers,* London, Vol. 30, 1931).

Principles of Calculation of Low Temperature Radiant Heating, by A. H. Barker (*Proceedings of The Institution of Heating and Ventilating Engineers,* London, Vol. 30, 1931).

Radiant Heat, by A. F. Dufton (*Proceedings of The Institution of Heating and Ventilating Engineers,* London, Vol. 31, 1932).

Chapter 45

WATER SUPPLY PIPING AND WATER HEATING

Maximum Probable Flow, Factor of Usage, Kind of Pipe Used, Sizing of Risers, Sizing of Mains, Sizing of Systems, Hot Water Supply, Water Heating, Hot Water Storage

DOMESTIC water supply systems present the engineer with a design problem that requires combining the somewhat empirical rules and formulæ in use with the more or less exact hydraulic principles involved. Unlike heating and ventilating layouts, there are practically no definite data for estimating the quantity of water likely to be consumed or the probable rate of water flow at any particular moment.

Metered results in one building often show two or three times the metered amount in another building of the same size and with the same types of tenants. In hotels, one riser will often have an almost constant flow that may never be reached by another at peak load. In office buildings, the women's toilets show a far greater daily consumption than those of the men, yet at no time will they approach the hourly consumption of the men's toilet during the first hour of the day. This condition has led to a multiplicity of rules of practice which vary as much as the data used. All must of necessity be based on an assumed rate of consumption and on an assumed probability of simultaneous use, and while the formulæ employed may have been derived on sound technical basis the assumptions are often in error.

To arrive at a safe standard, the approximate rate of flow of each fixture to be supplied must be known and the probable number of fixtures in use at any one time must be assumed. Obviously, the maximum number of fixtures assumed to be in use must be taken at the peak of demand and the lines must be made adequate to supply such a peak regardless of the riser or branch on which the demand may occur. This means that all water piping under the usual conditions will be over-sized.

In tall buildings it is customary to divide the water supply systems, both hot and cold, into sections of 10 to 20 stories. Such *zoning*[1] or

[1]It is impractical to attempt to size piping so as to produce the proper pressure on fixtures at different levels by employing friction, owing to the fact that this friction will be built up to the amount desired only in times of maximum demand and at all other times the friction will be only a fraction of the maximum friction so that the fixtures by this method are subjected to a varying pressure on the water supply line. A much more practical method is to throttle the flow at the fixture, or to use flow regulators, so that the quantity of water delivered will approximate the fixture demands and so that this is accomplished without splashing or noise.

To obtain the maximum probable flow it is necessary to multiply the maximum possible flow by a factor of usage, and this factor varies with the installation and the number of fixtures in the installation. It is evident that with two fixtures it is quite possible that both will at some time be in operation simultaneously. With 200 fixtures, it is unlikely the entire 200 would ever operate at the same time. Consequently, the factor of usage reduces as the number of fixtures becomes greater, all other things being equal.

TABLE 1. APPROXIMATE FLOW FROM FIXTURES UNDER NORMAL WATER PRESSURES

Fixtures	Cold Water (Gallons per Minute)	Hot Water (Gallons per Minute)
Water-closets, flush valve	45[a]	0
Water-closets, flush tank	10	0
Urinals, flush valve	30[a]	0
Urinals, flush tank	10	0
Urinals, automatic tank	1	0
Urinals, perforated pipe per foot	10	0
Lavatories	3	3
Showers, 4 in. heads, ½ in. inlets	3	3
Showers, 6 in. heads or larger	6	6
Needle bath	30	30
Shampoo spray	1	1
Liver spray	2	2
Manicure table	1½	1½
Baths, tub	5	5
Kitchen sink	4	4
Pantry sink, ordinary	2	2
Pantry sink, large bibb	6	6
Slop sinks	6	6
Wash trays	3	3
Laundry tray	6	6
Garden hose bibb	10	0

[a]Actual tests on water-closet flush valves indicate 40 gpm as the maximum rate of flow with 30 lb pressure at the valve; this would increase to 60 gpm (about 50 per cent) at 90 lb pressure. The 45 gpm has been taken as an average flow; possibly, with very low pressures just sufficient to operate the flush valve, 30 gpm could be allowed with safety. Urinal flush valves would vary proportionately in the same manner.

In practice all the elements will vary according to conditions; in the case of flush valve closets the duration of flush with the kind and condition of supply apparatus, the interval between flushes with the number of people using the system and their habits; and the length of the rush period with the type of installation and its location. The effect of each of these time elements on the results should be considered in connection with any data on which it is based before passing judgment on the selection of the factor of usage. The longer the duration of the flush the greater is the probability of overlapping flow. In selecting the factor of usage shown in Fig. 1 for systems having flush valves, 10 seconds was chosen as the maximum duration of flush, a value that represents an approximate average as water-closets are installed.

While the curve has been calculated for systems composed of water closets alone, it is possible to calculate probabilities for mixed systems of water closets and other smaller fixtures. It has been found however that for two systems both having the same maximum possible flow, one composed entirely of water-closets and the other a mixed system of water

closets and smaller fixtures, the probability of a given rate of flow is greater for the system composed of water-closets than for the mixed system. The use of this chart then would produce results which would be on the safe side for mixed systems.

Table 2. Schedule of Sizes for Down-Feed Riser (See Fig. 2)

Portion of Riser	Allowable Drop per Lb per 100 Ft	Maximum Probable Flow, Gallons per Minute																
		5	10	15	20	25	30	40	50	60	70	80	90	100	125	150	200	250
T	3.5	¾	1	1¼	1¼	1½	1½	2	2	2	2½	2½	2½	3	3	3	3½	3½
S	20	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	2	2	2	2	2	2½	3
R	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
Q	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
P	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
O	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
N	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
M	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
L	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
K	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
J	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
I	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
H	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
G	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
F	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
E	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
D	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
C	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
B	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
A	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½

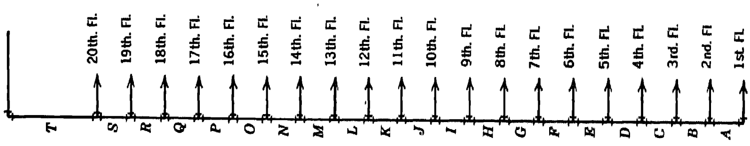


Fig. 2. Typical Riser for 20-Story Building

For systems composed entirely of fixtures other than flush valve fixtures the curve has been extended for smaller maximum possible flow values.

This chart applies to a normal building and not to installations where the inmates may all be required, for instance, to bathe on certain days of the week and at certain hours of those days; or in schools for example where all the showers in the gymnasium may be used simultaneously after instruction periods. In such special cases a new factor of usage must be developed based on the maximum probable usage under the conditions involved.

Example 1. Assume that in a normal building, such as a residential hotel or an apartment house, there are 50 flush valve water-closets, 50 lavatories, 50 sinks and 50 baths, and that it is desired to determine the maximum probable flow in a line supplying all of these fixtures with both hot and cold water.

Cold Water

50 W. C. x 45 gpm........2250 gpm
50 Lavs. x 3 gpm........ 150 gpm
50 Sinks x 4 gpm........ 200 gpm
50 Baths x 5 gpm........ 250 gpm

Maximum possible flow......2850 gpm

Fig. 1 shows a factor of usage of 9 per cent.

Maximum probable flow of cold water is 2850 × 0.09........ 257 gpm

Hot Water

50 Lavs. x 3 gpm........ 150 gpm
50 Sinks x 4 gpm........ 200 gpm
50 Baths x 5 gpm........ 250 gpm

Maximum possible flow...... 600 gpm

Fig. 1 shows a factor of usage of 23 per cent.

Maximum probable flow of hot water is 600 × 0.23........ 138 gpm

Total for main supplying cold and hot water (2850 + 600) × 0.08........ 276 gpm

It should be noted that this is a *rate of flow* or an *instantaneous demand.*

KIND OF PIPE USED

Before entering into the actual sizing of pipe, it is necessary to consider the kind of pipe to be used, and to make suitable allowance for corrosion and fouling during the lifetime of the system. For example, if brass, copper or alloy pipe is contemplated, it is probable that the quantities indicated in Example 1 are ample; if galvanized pipe is to be used, then it is quite likely that after a period of say 15 years the area may be decreased as much as 25 per cent and the quantitities of water assumed should be increased by 35 per cent to allow for this reduction of area; if the water contains lime it is possible that 50 per cent of the area may be lost and in such cases the flow should be doubled and no branch pipe connected to fixtures should be less than ¾ in. In all of the following calculations, the assumption is made that the water is fairly good and that a corrosion resistant type of pipe is to be used.

SIZING A DOWN-FEED RISER

Down-feed systems are commonly used for tall buildings. In sizing a riser arranged for down-feed, the gravity head permits a pressure drop that is almost prohibitive in an up-feed riser. There is a gain in riser head of 0.43 × 100 or 43 lb per 100 ft of run and hence it is quite permissible to size such a riser on the basis of a pressure drop of 30 lb per 100 ft of run, as the difference between the 43 lb generated and the 30 lb drop under maximum probable demand is ample to take care of the friction caused by the fittings. This method applied to the typical riser shown in Fig. 2 gives the schedule of sizes indicated in Table 2 for any flow from 5 to 250 gal.

SIZING AN UP-FEED RISER

When the riser is an up-feed, the opposite condition occurs; that is, there is a drop in pressure as the top of the riser is approached, due to the natural reduction in the gravity pressure, and to this must be added the pipe friction plus that introduced by the pipe fittings, all of which produce an excessive drop when compared to the conditions existing with a down-feed riser.

To size an up-feed riser the minimum pressure of the street main, or other source of supply, should be ascertained and from this should be subtracted the pressure to be maintained at the highest fixture, namely, 15 lb per square inch, plus the height in feet above the source of water pressure, multiplied by 0.43 to change from feet of head to pounds of pressure. The total length of run from the source of pressure to the farthest and highest fixture should be ascertained, and this should be changed to equivalent length of run to allow for the loss occasioned by

Table 3. Approximate Allowances for Fittings and Valves in Feet of Straight Pipe

Size of Pipe (Inches)	Type of Fitting or Valve					
	90-Deg Elbow	45-Deg Elbow	Return Bend	Gate Valve	Globe Valve	Angle Valve
½	4	3	8	2	48	8
¾	5	3	10	3	60	10
1	5	3	10	3	60	10
1¼	6	4	12	3	72	12
1½	7	5	14	4	84	14
2	7	5	14	4	84	14
2½	10	7	20	5	120	20
3	12	8	24	6	144	24
4	18	13	36	9	216	36
5	25	18	50	13	300	50
6	30	21	60	15	360	60

the pipe fittings. Table 3 gives the additional lengths necessary to allow for the various fittings and valves. The drop allowable in pressure per 100 ft of run may then be obtained by multiplying the surplus pressure (over that required for the gravity head and to supply 15 lb at the fixture) by 100 and by dividing this by the equivalent length of run to the farthest or highest fixture.

Where street water pressures are available the pressure drop through the meter and service pipe must be taken into consideration. Table 4 shows the pressure loss through meters. It also gives the minimum sizes of recommended service and maximum meter deliveries.

Example 2. Assume a street pressure of 60 lb, the height of the highest fixture 50 ft, and the length of the longest run 200 ft. Without knowing the additional length of pipe to be added for the fittings it will be assumed that this is about 100 ft. The surplus pressure which will be available for pressure drop will then be 60 lb − (15 lb + 50 ft × 0.43 lb) = 60 lb − (15 lb + 21.5 lb) = 23.5 lb.

To change this into drop per 100 ft: $\frac{23.5\text{ lb} \times 100}{200\text{ ft} + 100\text{ ft}} = 7.8$ lb per 100 ft.

The pipe may then be sized from the maximum probable flow by selecting a size that does not give a drop in excess of 7.8 lb per 100 ft.

It will be seen from Example 2 that it is impossible to size up-feed risers without determining the drop allowable in both the horizontal feed mains and the toilet room branches. Having once ascertained this allowable drop, it is simply a matter of applying it throughout the system.

TABLE 4. PRESSURE LOSS THROUGH WATER DISC METERS[a]

A. W. W. A. Standards

Rate of Flow Gpm	Approx. Pressure Loss Through Meters, Lb per Sq In Pipe Size (In.)							
	⅝	¾	1	1½	2	3	4	6
5	1.5	0.5	0.2					
10	6.0	2.0	1.0	0.2				
15	14.0	5.0	2.0	0.6	0.2			
20	25.0	9.0	3.5	1.0	0.4			
25		13.5	5.5	1.5	0.6			
30		19.5	8.0	2.0	0.9			
35			11.0	3.0	1.0			
40			14.0	4.0	1.5			
45			18.0	5.0	2.0			
50			22.0	6.0	2.5	0.7		
75				14.0	5.5	1.5		
100				25.0	10.0	2.8	1.0	
125					15.0	4.0	1.5	
150					22.0	6.0	2.2	
175						8.0	3.0	
200						10.4	4.0	1.0
250							16.0	1.5
300							23.0	2.2
350								3.0
400								4.0
500								6.5
600								9.0
800								16.0
1000								25.0

Minimum Size of Service Recommended

Rate of Flow Gpm	Approx. Minimum Pipe Size of Service, Main to Meter (In.) Maximum Length (Ft)				
	30	75	100	150	200
1–20	¾	¾	1	1	1
20–30	¾	1	1	1½	1½
30–50	1	1½	1½	1½	1½
50–100	1½	1½	2	2	2
100–150	1½	2	2	2½	2½

Safe Maximum Delivery of Meters

Meter Size In.	Capacity, Gpm Based on 25 Lb Loss Through Meter
⅝	20
¾	34
1	53
1½	100
2	160
3	315
6	500
8	1000

[a]Pressure loss through compound and current meters is less than shown in table. For exact information consult manufacturers.

HORIZONTAL SUPPLY MAINS

The horizontal mains supplying the risers at the top of a down-feed system must be liberally sized unless the house tank is set at a much higher elevation than usual. To provide a gravity head on the highest fixtures of 15 lb per square inch it is necessary for the water line in the house tank to be nearly 40 ft higher, and with the line loss considered this becomes about 45 ft. Such heights are not often practical and as a result the pressure on the highest fixtures either is reduced to 7 lb (which is sufficient to operate a flush valve), or flush tank water-closets are substituted, or a separate cold and hot water supply is installed with a small pneumatic tank to give the increase in pressure necessary. The chief objection to the use of a pneumatic tank is that a separate hot water heater is required and this heater must be located either sufficiently below the highest fixtures to obtain a gravity circulation, or it must be provided with a circulating pump in order to force the hot water to the top floor level.

The most common solution is to place the house tank as high as the structural and architectural conditions will permit and then to use liberally-sized lines between the house tank and the upper fixtures, say for the two top stories, below which the riser sizes may be reduced to those indicated in Fig. 2 and Table 2. Where the house tank is only one story above the top fixtures, flush tank water-closets must be used and the drop in the entire run from the house tank down to the farthest fixture should not exceed 1 lb; the less, the better. This means that if the total equivalent run to the farthest top fixtures supplied is 300 ft, the drop per 100 ft should not exceed $\frac{1 \text{ lb} \times 100}{300}$ or 0.33 lb per 100 ft. The friction curves shown in Fig. 3 may be used for quickly determining the proper size of pipe to give any desired drop in pounds per 100 ft of equivalent run.

OVERHEAD DISTRIBUTION MAIN

Example 3. Suppose an installation has a house tank in which the water line is 20 ft above the level of the top fixtures to be supplied and that the length of run to the farthest fixtures on this level is 400 ft with the pipe fittings adding another 200 ft, making an equivalent length of 600 ft. What would be the size of main coming out of the tank where a maximum flow rate of 400 gpm may be expected, of the horizontal main where a maximum flow rate of 200 gpm may be expected, and of the riser down to the fixture level where the maximum flow rate is approximately 100 gpm?

Here the level of the water in the house tank is 20 ft above the faucet of the highest fixture and the gravity pressure will be 0.43 lb × 20 ft = 8.6 lb and, if a total pressure drop of 1 lb is assumed, the pressure on the farthest fixture under times of peak load will be 8.6 lb − 1 lb = 7.6 lb while the drop per 100 ft of equivalent run will have to be

$$\frac{1 \text{ lb} \times 100}{600} = 0.1667 \text{ lb}.$$

Referring to Fig. 3 it will be noted that where the flow through the main is 400 gpm, an 8-in. pipe would be required; that where the flow is reduced to 200 gpm, a 6-in. pipe would be sufficient; and that where the flow is 100 gpm in the riser branch and riser, a 5-in. size would be correct. Of course these are somewhat excessive flows and the head from the tank is small so that large sizes are to be expected. It would be necessary to carry a 5-in. riser down to the branch to the top floor, then reduce to 4 in. for the branch to the floor below the top, and below this the sizes in Table 2 could be followed. In such a case, flush tank closets should doubtless be substituted.

Had the tank been set 10 ft higher, the head available to be used up in friction, but

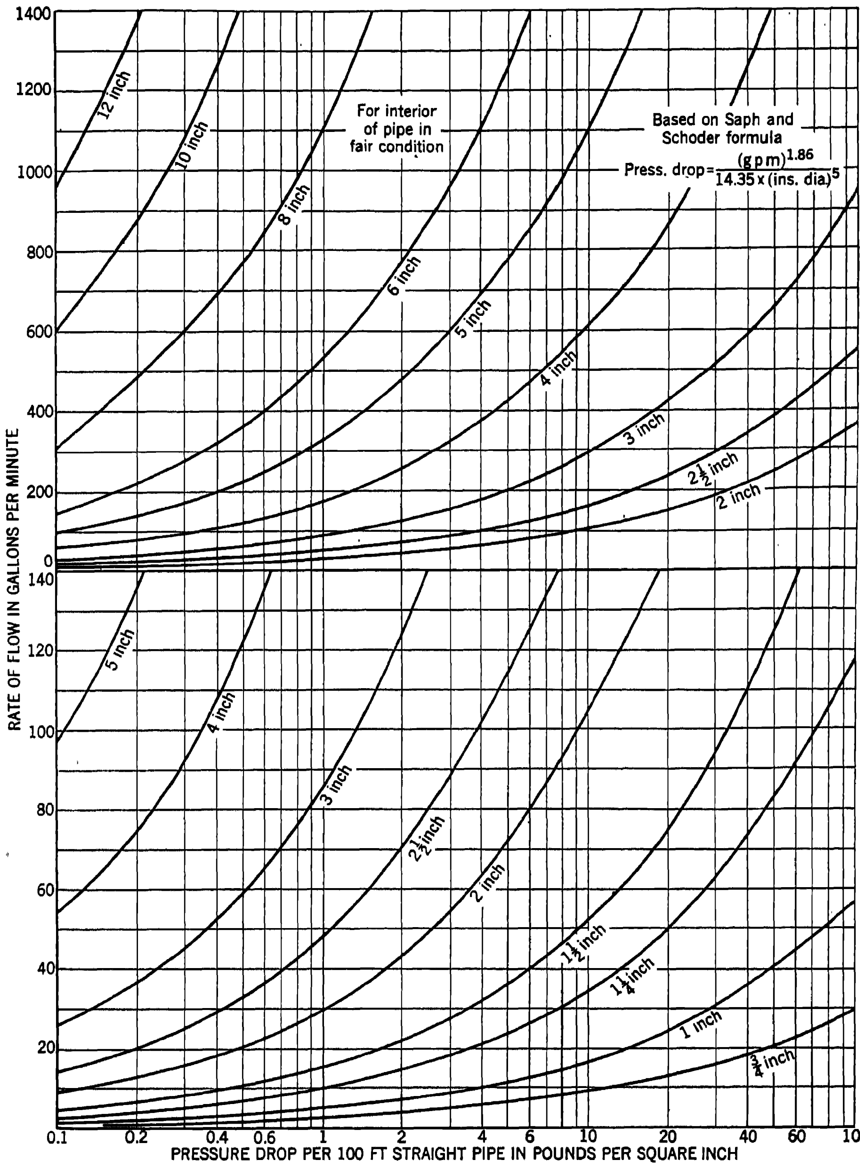


FIG. 3. CHART GIVING PRESSURE DROP FOR VARIOUS RATES OF FLOW OF WATER

still giving the same pressure at the top fixtures, would have been 0.43 lb × 10 ft or 4.3 lb greater and this, with the 1 lb drop used previously, would give a total allowable drop of 1 lb + 4.3 lb = 5.3 lb which, divided by the 600 ft equivalent run gives a drop per 100 ft of $\frac{5.3 \times 100}{600} = 0.9$ lb

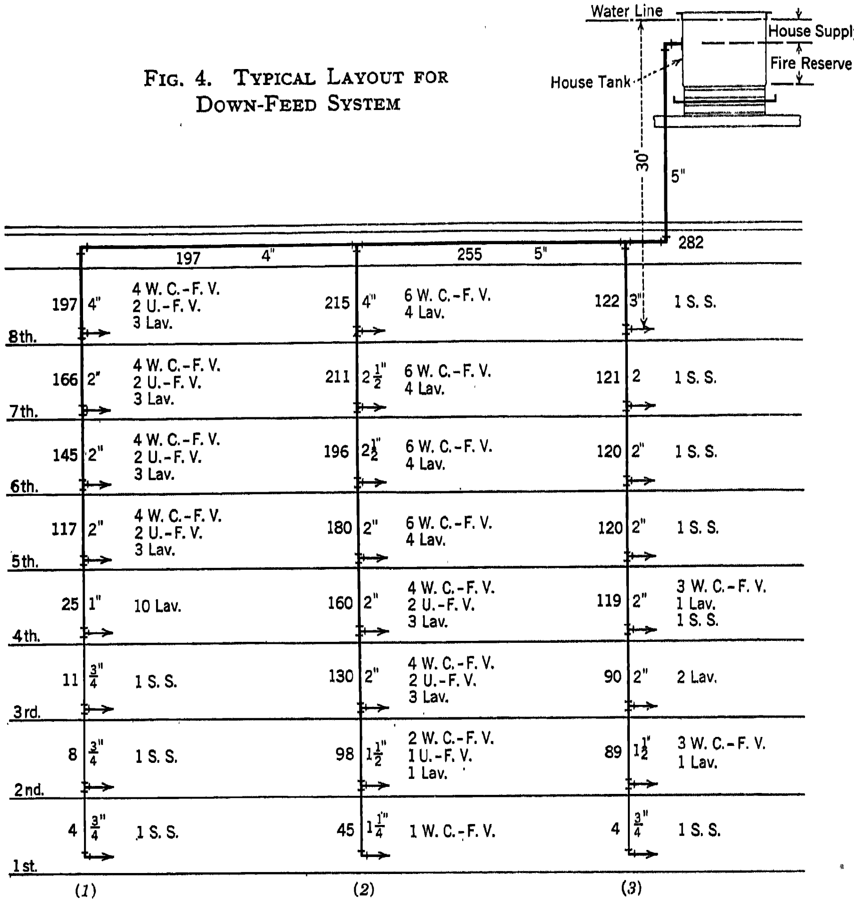


FIG. 4. TYPICAL LAYOUT FOR DOWN-FEED SYSTEM

and, with this drop, the sizes according to the chart (Fig. 3) are 6 in., 5 in., and 4 in., respectively, while if the run is reduced to 200 ft instead of 600 ft, the allowable drop will be $\frac{5.3 \text{ lb} \times 100}{200} = 2.7$ lb per 100 ft. This gives 5 in., 4 in., and 3 in., respectively, for the flows of 400, 200, and 100 gpm.

From Example 3 it is evident that, while the down-feed system possesses certain economies in size for the riser portion, it is quite likely to involve large distribution main sizes, especially when the tank is not elevated to a considerable degree.

SIZING A PIPING SYSTEM

Example 4. Fig. 4 shows a typical layout with three risers extending eight stories and with the fixtures noted on each floor. First this will be solved for a down-feed arrangement assuming that the level of the water in the house tank is 30 ft above the fixtures on the top floor, that the length of run from the tank to the farthest fixture is 200 ft, equivalent length of fittings 100 ft, and the pressure required at the fixture is 7 lb.

Table 5. Typical Calculation of Pipe Sizes on Down-Feed Riser with Flush Valve Water-Closets and Urinals

(Riser No. 1. Fig. 4)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
1st	1 S. S.	4	4	4	100	4	30	¾
2nd	1 S. S.	4	4	8	100	8	30	¾
3rd	1 S. S.	4	4	12	92	11	30	¾
4th	10 Lav.	3	30	42	58	25	30	1
5th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9					
			249	291	40	117	30	2
6th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9					
			249	540	27	145	30	2
7th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9					
			249	789	21	166	30	2
8th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9					
			249	1038	19	197	2	4

The 30-ft head is equal to a static pressure of 0.43 × 30 or 12.9 lb per square inch and to maintain a pressure of 7 lb at the highest fixtures the drop allowable in pressure is 12.9 − 7.0 lb or 5.9 lb. As the total equivalent run is 300 ft, this is a drop per 100 ft of 1.97 lb, or practically 2 lb. Therefore, all risers and mains from the top floor back to the tank must be sized on the basis of a drop of 2 lb per 100 ft. Tables 5, 6, 7 and 8 show the schedule for Risers Nos. 1, 2 and 3 with the maximum possible flow taken from Table 1, the percentage of use at the peak taken from Fig. 1, and the maximum probable flow at the peak worked out for each portion of the riser, the riser sizes being taken from Table 2 as far as possible and from Fig. 3 where the amounts exceed the values given in this table; a drop of 30 lb per 100 ft is used except on the riser from the top floor back to the tank where 2 lb per 100 ft is the allowable limit.

The reduction in pipe size which would occur if flush tank water-closets were used on the top floor and only 3 lb pressure used on the fixtures is given in Tables 9 and 10. This illustrates why flush tank closets so frequently are substituted on the uppermost floor when a house tank is the source of water pressure.

If it is now assumed that Riser No. 1 is to be fed from the bottom and the minimum street pressure is 75 lb with the top fixture of the riser 80 ft above the main, the problem would be solved by determining the maximum rate of flow in each portion of the riser as shown in Table 11 and then finding the allowable drop which can be used per 100 ft. The 80 ft of riser height will use up 0.43 lb × 80 = 34.4 lb and the pressure at the top of the required 15 lb will make the total reduction 49.4 lb, leaving a balance of 25.6 lb which may be used up in friction. If the distance from the street main to the bottom of the riser, which will be assumed to be the farthest one on the horizontal line, is 100 ft, and if the fittings are sufficient to add another 100 ft, as well as the 80 ft of vertical distance up the riser, the total equivalent run will be 280 ft, which will be taken as an even 300 ft.

TABLE 7. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(*Riser No. 3. Fig. 4*)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
1st	1 S. S.	4	4	4	100	4	30	¾
2nd	3 W. C. 1 Lav.	45 3	135 3					
			138	142	63	89	30	1½
3rd	2 Lav.	3	6	148	61	90	30	1½
4th	3 W. C. 1 Lav. 1 S. S.	45 3 4	135 3 4					
			142	290	41	119	30	2
5th	1 S. S.	4	4	294	41	120	30	2
6th	1 S. S.	4	4	298	40	120	30	2
7th	1 S. S.	4	4	302	40	121	30	2
8th	1 S. S.	4	4	306	40	122	2	3

required that a gravity circulation be kept up in such hot water lines and this often has a considerable influence on the size. There are three methods of arranging circulation lines, as follows:

1. By using the plain up-feed with a return carried back from the top of the riser and paralleling it.
2. By carrying a supply riser up in one location thus supplying fixtures on up-feed, then crossing over at the top and coming down past another collection of fixtures and supplying these by a down-feed.
3. By carrying all of the water to the top of the building and dropping risers wherever needed, feeding all hot water on a down-feed system.

TABLE 8. SIZE OF DISTRIBUTION MAIN FOR DOWN-FEED SYSTEMS (SEE FIG. 4)

Riser No.	Maximum Gpm Riser	Maximum Gpm Main	Probable Use (per cent)	Probable Gpm	Allowable Drop Lb per 100 Ft	Size of Main In.
1	1038	1038	18	187	2	4
2	1794	2832	9	255	2	4
3	306	3138	9	282	2	5

In the first instance the up-feed riser may be sized for the same pressure drop as used for the cold water riser and, from the top of the riser *just below the top fixture connection*, a return circulation line may be carried back to the main return line in the basement and connected through a check valve, set on a 45-deg angle, and a gate valve; these return circulation lines should never be less than ¾ in., and on the farther half of the risers, not less than 1 in. to favor circulation in the far end. Typical top and bottom connections for such risers are shown in Fig. 6.

TABLE 9. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISERS WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY (SEE FIG. 4)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
				Riser No. 1				
7th and below				789	21	166	30	2
8th	4 W. C. 2 U. 3 Lav.	10 10 3	40 20 9 69	858	20	172	3.3	4
				Riser No. 2				
7th and below				1512	14	211	30	2½
8th	6 W. C. 4 Lav.	10 3	60 12 72	1594	14	223	3.3	4
				Riser No. 3				
7th and below				302	40	121	30	2
8th	1 S. S.	4	4	306	40	122	3.3	3

For the second arrangement of hot water risers (Fig. 7*b*), circulation lines are run back from the last fixture supplied to the main return circulation line in the same manner as just described, using ¾ in. for the near risers and 1 in. for the far risers. The sizing is much more difficult, as it is necessary to start at the bottom floor of the return riser and work back to the top of this riser and then carry the maximum flow across on to the top of the corresponding supply riser and work down on this riser from the top floor to the bottom. Naturally this gives a much greater flow in the supply riser and aids circulation by reducing pipe friction. The allowable loss per 100 ft in such lines must be made about half that used for the cold water risers which do not have the combined up- and down-travel which the hot water must make.

In the third and most common arrangement (Fig. 7*c*) all of the water is carried from the tank or heater directly to the top of the building and is there distributed to the risers which are down-feed and may be sized in the

TABLE 10. SUMMARY OF RISER SIZES TO GIVEN MAIN SIZES WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY (SEE FIG. 4)

Riser No.	Maximum Gpm Riser	Maximum Gpm Main	Probable Use (per cent)	Probable Gpm	Allowable Drop Lb per 100 Ft	Size of Main In.
1	858	858	20	172	3.3	4
2	1594	2452	10	245	3.3	4
3	306	2758	9	248	3.3	4

regular down-feed manner if the total equivalent run either from the street main or house tank is taken into consideration. The return circulation lines from the bottom of each riser should be arranged in the manner already outlined and any riser not going to the basement to supply fixtures must have these returns carried down to the basement from the termination of the supply riser at whatever level it may end.

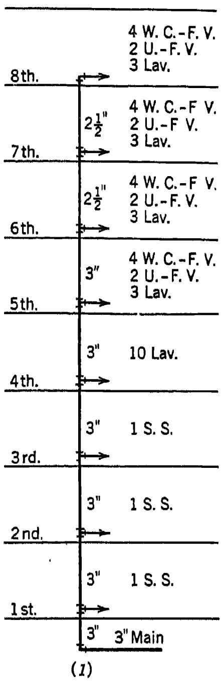


FIG. 5. UP-FEED SYSTEM

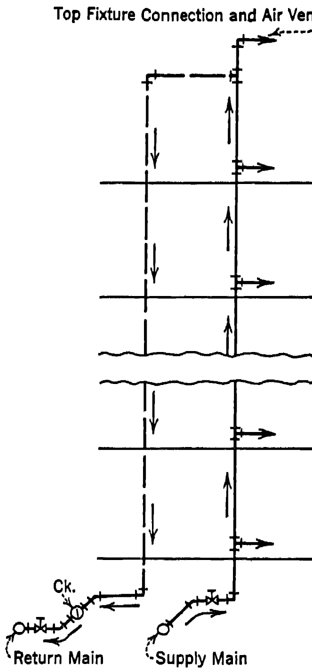


FIG. 6. SUPPLY AND RETURN MAIN CONNECTIONS FOR HOT WATER SUPPLY SYSTEM

All risers, both hot and cold, should be valved at the main with an extra check valve on the hot water return circulation so that the risers may be cut off and repaired when necessary without disturbing the service in the remainder of the system.

HOT WATER SUPPLY

Having designed the service hot water piping, the next step is to furnish some means of heating the water and in this respect it is necessary to pass from the maximum probable flow to the *maximum probable hourly demand*, which is quite different. If an instantaneous heater were used, it would require adequate capacity to provide for the heating of the water as fast as it is drawn and a heater of this type should be sized on the basis of the maximum probable flow with the accompanying heavy drafts on the heating device and with intervals of no draft at all. To balance these inequalities of flow the storage-type heater is often utilized so that the

TABLE 11. TYPICAL CALCULATION OF PIPE SIZES ON UP-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS (SEE FIG. 5)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
8th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	249	44	109	8.5	2½
7th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	498	28	139	8.5	2½
6th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	747	22	164	8.5	3
5th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9 249	996	18	179	8.5	3
4th	10 Lav.	3	30	1026	18	185	8.5	3
3rd	1 S. S.	4	4	1030	18	186	8.5	3
2nd	1 S. S.	4	4	1034	18	187	8.5	3
1st	1 S. S.	4	4	1038	18	188	8.5	3

TABLE 12. SUGGESTED STORAGE TANK SIZES FOR HOMES AND APARTMENTS

All Year Service Based on Boiler Water at 180 F				Service During Heating Season Based on Boiler Water at 215 F			
Tank Capacity Gal	Piping Connections		Number of Baths or Families	Tank Capacity Gal	Piping Connections		Number of Baths or Families
	Boiler, In.	Tank, In.			Boiler, In.	Tank, In.	
30	1	¾	1	30	1	¾	1
35	1¼	¾	1	40	1	¾	1
40	1¼	¾	1–2	52	1	¾	1
50	1¼	¾	1–2	66	1¼	1	1–2
60	1¼	1	1–2	82	1¼	1	2–3
72	1½	1	2–3	100	1¼	1	3
80	2	1	2–3	120	1¼	1	4
100	2	1¼	3–4	144	1½	1	5
125	2	1¼	4–5	160	2	1¼	6
150	2	1¼	5–6	200	2	1¼	6–7
200	2	1½	6–7	250	2	1½	7–9
250	2½	1½	7–9	300	2	1½	9–11
300	2½	1½	9–11	400	2½	2	11–15
400	3	2	11–15	500	2½	2	15–18
500	3	2	15–18	600	3	2½	18–21

water demand can be heated during periods of light demand and stored up for use during the periods of heavy demand. The total water consumption per person usually varies between 100 and 150 gal per day when laundry and culinary operations for the occupants are carried out on the same premises. The maximum hourly demand under these conditions will be found to be about one-tenth of the average daily consumption.

If one-third of the total water used is hot water and 125 gal per day are assumed as a fair average of consumption per person, it is apparent that each person uses about 40 gal of hot water per day. If one-tenth of this represents the peak hourly load, then 4 gph must be allowed per person for the heaviest demand. If the average occupancy of apartments is 3 persons, the peak hour demand per apartment will be about 12 gph. It is customary to allow 10 gph of heating capacity per apartment. Water in excess of this heating capacity drawn out during the peak hours is

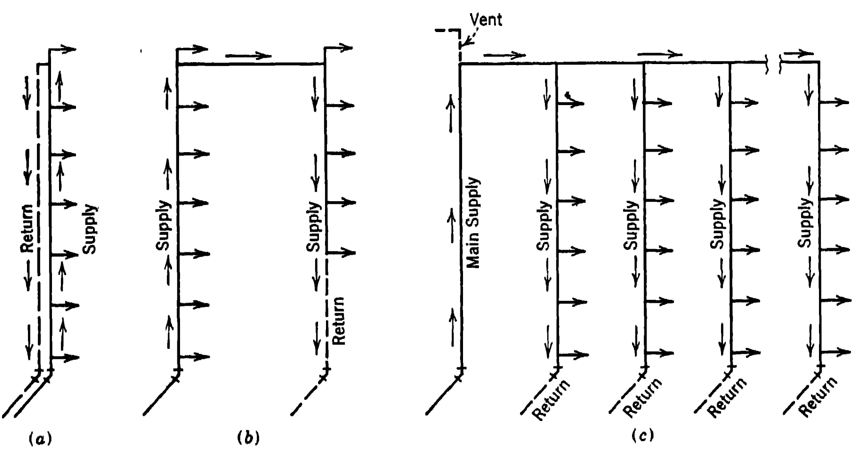


FIG. 7. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

provided for by storage in the hot water tank where this water is heated during hours when the demand is below the average. In single homes, hot water use averages 20 gal per day per person. Table 12 gives suggested storage tank sizes for homes and apartments based on the number of families or baths.

WATER HEATERS

Various types of heaters are available for supplying the hot water for domestic service in buildings. In any hot water supply system the water should be heated to a temperature between 150 and 180 F. Where the hot water requirements include supplies for kitchens, laundries or process work, the higher temperatures are used. In buildings where steam is available throughout the year, the hot water supply is usually taken from this source. In smaller domestic installations the fuel-burning device is generally automatically arranged so that hot water is supplied the entire year and not merely when the boiler is used for heating purposes.

Water is heated by various methods using heat exchangers arranged so that the boiler heating medium gives up its heat to the water in the hot water circulating system. These heat exchangers may be classified as follows:

1. Submerged steam heating coil in storage tank.
2. Submerged water heating coil in storage tank. (Fig. 8).
3. Indirect water heater, mounted on side of boiler below water line. (Fig. 9).
4. Submerged indirect water heater, placed in boiler below water line. (Fig. 10).

The efficiency of these heaters may be estimated as nearly 100 per cent as the heat loss from surface radiation of the heater and tank shell when covered with insulating material is generally reduced to a minimum. The capacities of these heaters are usually available from manufacturers'

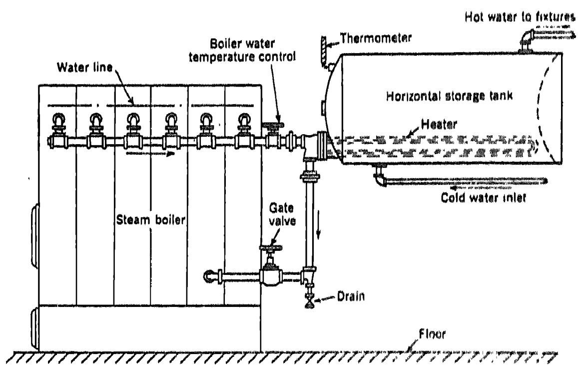


FIG. 8. WATER HEATING COIL SUBMERGED IN STORAGE TANK

rating tables. The area of the inside surface of a heating coil may be determined from the following equation:

$$A = \frac{Q \times 8.33\,(t_o - t_i)}{K_o \times t_m} \tag{1}$$

where

A = surface area of coil, square feet.

Q = quantity of water heated, gallons per hour.

t_o = hot water outlet temperature, degrees Fahrenheit.

t_i = cold water inlet temperature, degrees Fahrenheit.

K_o = coefficient of heat transmission, Btu per hour per square foot surface. For copper or brass coils K_o = 240 (steam) and 100 (hot water). For iron coils K_o = 160 (steam) and 67 (hot water).

t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature. t_m is approximately = $\left[t_s - \frac{(t_o + t_i)}{2}\right]$

Equation 1 may also be used for determining revised heating coil ratings under different temperature conditions as stated in the manufacturers' ratings. When selecting a water heater, the conditions of operation should be carefully considered, as well as the location of the storage tank and the piping arrangement between the boiler, heater and tank. It is generally good practice to allow a margin of safety when

selecting an indirect heater of the proper size to provide for loss of efficiency due to the accumulation of scaling on the coils and piping. Heat exchangers classified according to (3) and (4) may be used with or without a storage tank, but when tanks are omitted, the indirect water heaters should be increased in size so as to heat the water instantaneously as it is needed.

The storage tank should be installed as high as possible. Horizontal tanks are preferable for all medium size installations and absolutely essential on larger installations. Where possible the storage tank should be installed with the bottom of the tank at or above the boiler water line. Horizontal storage tanks smaller than 18 or 20 in. diameter are not recommended because of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn.

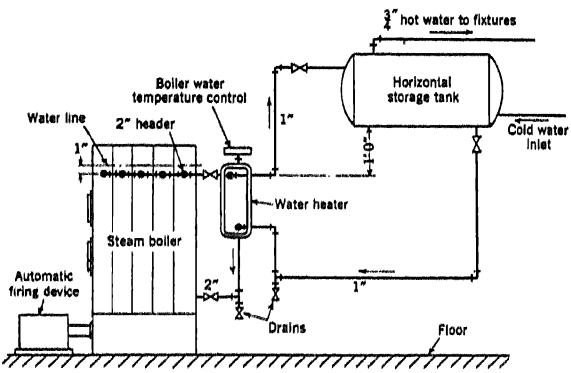


Fig. 9. Indirect Water Heater Mounted on Side of Boiler

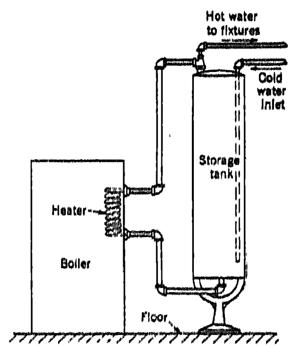


Fig. 10. Indirect Water Heater Placed in Boiler

Pipe sizes between the water heater and boiler should be full size of the heater tappings (Table 12). When a heater is connected to a horizontal sectional boiler, it is recommended that connections be made to all sections and joined together a few inches below the water line as shown in Fig. 8, so that steaming is prevented in those sections which are not connected to the header.

When a steam coil is used for heating the water, an automatic thermostatic valve may be installed in the steam supply to the coil. The operation of this automatic valve is controlled by a thermostat located in the storage tank which permits the proper amount of steam to enter the coil so as to maintain an even water temperature.

An indirect water heater may be used on either a steam or hot water system, and generally this type of heater is provided with a temperature control device located in the boiler water circulating connection to the water heater. The setting on this thermostatic valve may be as low as 140 F or as high as 180 F and may be readily adjusted to meet particular requirements. With this type of control it is impossible to overheat the hot water supply which is an important safety consideration in some installations. This type of system may also be conveniently used during

the non-heating season with the operation of the fuel burning device controlled by the water heater thermostat. (See Chapter 33.) During the heating season the water heater temperature control functions as a low limit control.

When an indirect water heater is applied to a gravity hot water system, it is necessary to provide a valve in the supply to the heating system to prevent the flow of hot water from the boiler when heat is not required in the house. This valve may be controlled from a room thermostat and the automatic fuel-burning device controlled from the water heater thermostat. To prevent circulation in a forced hot water heating system flow control valves may be installed in the flow and return lines which act merely as check valves when the circulating pump is not operating. In this arrangement the pump is controlled by the room thermostat and the automatic fuel-burning device is controlled from the water heater thermostat.

STORAGE CAPACITY AND BOILER ALLOWANCES

The amount of storage provided in the hot water tank or heater is somewhat a matter of choice but is usually made ample to carry over the peak shortage which is likely to occur and is based on the assumption that only 75 per cent of the storage capacity will be available, as it has been found that if more than this amount is withdrawn from storage, the tank is so cooled down as to make the balance useless. The general rule may be cited that the less the heating capacity the greater must be the storage, and the greater the storage the less may be the heating capacity down to a point where the heating capacity will fail to be sufficient to heat up the tank storage during the periods of small load.

Example 5. A heater to supply 500 persons will have an average daily use of about 500×40 gal = 20,000 gal and this is an average of $\frac{20{,}000 \text{ gal}}{24} = 833$ gph but the peak hour will require $\frac{1}{10}$ of 20,000 = 2000 gal and the shortage during the peak hour, if the heating capacity is made to suit the average hourly use of 833 gal, will be $2000 - 833 = 1167$ gal so that the storage capacity, based on 75 per cent being available from this capacity without cooling the tank excessively, will be $\frac{1167}{0.75} = 1556$ gal.

Should it be desired to reduce the size of storage tanks and to use a greater heating capacity, it is only necessary to increase the heating capacity to 1200 gph which then gives $2000 - 1200 = 800$ gal as the shortage during the peak hour, and the necessary storage will be $\frac{800 \text{ gal}}{0.75} = 1067$ gal; or the heating capacity can be increased to 1500 gal, leaving a shortage of $2000 - 1500 = 500$ gal.

Good design requires that the heating capacity be made as small as possible without introducing undesirable amounts of storage, as the heating capacity directly determines the load on the source of heat.

As indicated in Example 5, the heating load is proportional to the heating capacity and the boiler capacity must be increased for higher heating capacities and may be reduced for smaller heating capacities with greater storage. It may be assumed that a boiler capacity of about 4 sq ft of equivalent steam heating surface[2] (radiation) must be provided for every gallon of water heated 100 F or from 50 F to 150 F, which is

[2]Actual requirement for 100 deg temperature difference = $\frac{100 \times 8.33}{240} = 3.48$ sq ft per gallon of water heated.

TABLE 13. ORDINARY MAXIMUM HOURLY DEMAND FOR HOT WATER FOR VARIOUS FIXTURES IN GALLONS AND PROBABLE PERCENTAGE OF USAGE

TYPE OF BUILDING	LAVATORIES		BATHS	SHOWERS	SLOP SINKS	KITCHEN SINKS	PANTRY SINKS	FOOT BATHS	WASH TRAYS	AVG. MAX. USE[a]
	Private	Public								
MAXIMUM PROBABLE USAGE GPH	20	20	40	300	30	30	20	20	50	
	Probable Usage in Per Cent of Maximum Ordinary Use									
Apt. house	25	50	33	67	67	33	50	25	60	35
Club	25	75	50	67	67	67	100	25	80	60
Gymnasium	25	100	100	100	......	----	......	100	......	80
Hospital	25	75	50	33	67	67	100	25	80	45
Hotel	25	100	50	33	100	67	100	25	80	70
Industrial	25	150	100	100	67	67	......	100	......	90
Laundries	25	100	......	......	33	----	......	......	100	100
Office building	25	75	......	......	50	----	......	......	......	20
Baths	25	150	150	100	50	----	......	......	......	100
Residences	25	......	50	33	50	33	50	50	60	50
Schools	25	75	......	100	67	33	100	50	......	25
Y. M. C. A.	25	100	100	100	67	67	100	100	80	75

[a]Percentage of fixtures likely to be demanding maximum probable usage at any one time.

TABLE 14. HOT WATER CONSUMPTION IN VARIOUS TYPES OF BUILDINGS FOR DIFFERENT PURPOSES

TYPE OF BUILDING	CONDITIONS	GALLONS
Hotels	Room with basin only	10 (per day)
	Room with bath	
	(Transient)	40 (per day)
	(Men)	40 (per day)
	(Mixed)	60 (per day)
	(Women)	80 (per day)
	Two-room suite and bath	80 (per day)
	Three-room suite and bath	100 (per day)
Public Buildings	Public bath or lavatory	150 (per day per fixture)
	Public shower	200 (per day per fixture)
	Public lavatory with attendant	200 (per day per fixture)
Industrial Buildings	Per office employee	2 (per day)
	Per factory employee	5 (per day)
	Cleaning floors	3 (per 1000 sq ft per day)
Restaurants	$0.50 Meals	0.5 (per customer with hand washing)
		1.0 (per customer with machine washing)
	$1.00 Meals	1.0 (per customer with hand washing)
		2.0 (per customer with machine washing)
	$1.50 Meals	1.5 (per customer with hand washing)
		4.0 (per customer with machine washing)

Chapter 46

TERMINOLOGY

Glossary of Physical and Heating, Ventilating and Air Conditioning Terms Used in the Text, Standard Abbreviations, Conversion Equations, Drafting Symbols, Specific Heat Table

Absolute Humidity: *See Humidity.*

Absolute Pressure: The pressure referred to that of a perfect vacuum. It is the sum of gage pressure and barometric pressure.

Absolute Temperature: A reading on the absolute temperature scale. Absolute temperature is obtained by adding 459.70 degrees to the Fahrenheit temperature.

Absolute Zero: The zero point on the absolute scale 459.70 F below the zero of the Fahrenheit scale.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second. $a = V \div t$.

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body, the value of which varies with latitude and elevation. The international gravity standard has the value of 980.665 cm per second per second or 32.174 ft per second per second, which is the actual value of this acceleration at sea level and about 45 deg latitude.

Adiabatic: An adjective descriptive of a process in which no heat is added to or extracted from the system executing the process.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in greater or lesser degree human health or comfort. (See *Comfort Air Conditioning.*)

Air Washer: An enclosure in which air is forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of moving air.

Atmospheric Pressure: The pressure indicated by a barometer. *Standard atmospheric pressure* is a pressure of 76 cm mercury (density 13.5951 grams per cubic centimeter, gravity 980.665 cm per second per second). It is equivalent to 14.6959 lb per square inch or 29.921 in. of mercury at 32 F.

Baffle: A plate or wall for deflecting gases or fluids.

Blast: This word was formerly used to denote forced air circulation, particularly in connection with central fan systems using steam or hot water as the heating medium. As applied in this sense, the word *blast* is now obsolete.

Boiler: A closed vessel in which steam is generated or in which water is heated.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of $970.3 \times 34.5 = 33{,}475$ Btu per hour.

British Thermal Unit: A unit of energy defined in terms of the international steam-table calorie through the convenient relation 1 Btu per pound per degree Fahrenheit = 1 cal per gram per degree Centigrade. It is approximately the quantity of heat required to raise the temperature of 1 lb of liquid water from 63 to 64 F.

By-pass: A pipe or duct, usually controlled by valve or damper, for short-circuiting fluid flow.

Calorie: (large calorie or kilogram calorie) is equal to 1000 international steam-table calories = 1/860 international kilowatthour. For practical purposes it may be considered as 1/100 of the heat required to raise the temperature of 1 kilogram of water from 0 to 100 C.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distributing ducts. (See Chapter 20.)

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Transmission: The amount of heat (Btu) transmitted *from air to air* in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F *between the air on the inside and that on the outside of the wall, floor, roof or ceiling.*

Comfort Air Conditioning: The process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. (See *Air Conditioning.*)

Comfort Line: The effective temperature at which the largest percentage of adults feel comfortable.

Comfort Zone (*Average*): The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. *Comfort Zone* (*Extreme*): The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 2.)

Concealed Radiator: A heating device located within, adjacent to, or exterior to the room being heated but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, does not *see* the room. Such a device transfers its heat to the room largely by convection air currents.

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 F between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material *1 in. thick* for a difference in temperature of 1 F between the two surfaces of the material.

Conductor (*Heat*)*:* A material capable of readily conducting heat. The opposite of an insulator or insulation.

Constant Relative Humidity Line: Any line on the psychrometric chart representing a series of conditions which may be evaluated by one percentage of relative humidity; there are also *constant* dry-bulb lines, wet-bulb lines, effective temperature lines, vapor pressure lines, and lines showing other physical properties of air mixed with water vapor.

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. Convection may be *natural* or *forced.*

Convector: A heat transfer surface designed to transfer its heat to surrounding air largely or wholly by convection. Such a surface may or may not be enclosed or concealed. When concealed and enclosed the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator.*) (See also Chapter 12.)

Decibel: A unit commonly used for expressing sound or noise intensities referred to an arbitrary reference level. It is defined by the relation db = $10 \log_{10} \frac{P_1}{P_0}$, where P_1 is the unknown intensity, and P_0 is the reference level which is commonly taken as 10^{-16} watts per square centimeter.

Degree-Day: A unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exists as many degree-days as there are degrees Fahrenheit difference in temperature between the mean temperature for the day and 65 F.

Degree of Saturation or Per Cent Saturation: The ratio of actual humidity ratio W to the saturation humidity ratio W_s corresponding to the actual temperature and the observed pressure. $\mu = \frac{W}{W_s}$ (Approximately the same as but not identical with *relative humidity*. See Chapter 1).

Dehumidification: The condensation of water vapor from air by cooling below the dew-point.

Dehydration: The removal of water vapor from air by the use of adsorbing or absorbing materials.

Density: The weight of a unit volume, expressed in pounds per cubic foot. $d = W \div V$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *Radiator*.

Direct-Return System (*Hot Water*)*:* A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*)*:* A pipe which carries steam downward to the heating units and into which the condensation from the heating units drain.

Down-Feed System (*Steam*)*:* A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft Head (*Side Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening. (*Top Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe, considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry Air: In psychrometric work, *dry air* is defined as air without water vapor. This state, though not obtained practically, is used as the basis of calculations.

Dry-Bulb Temperature: The temperature indicated by a standardized thermometer after correction for radiation, etc.

Dry Return: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. (See *Wet Return.*)

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: Same as *Total Pressure.*

Effective Temperature: An arbitrary index which combines into a single value the effect of temperature, humidity, and movement of air on the degree of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation of warmth.

Enthalpy: A thermodynamic property which serves as a measure of the quantity of thermal energy *convected* by a fluid in steady flow. In a non-flow process the increase of enthalpy equals the quantity of heat absorbed *provided pressure is constant.* Enthalpy was formerly called *heat content*, sometimes *total heat.* Specific enthalpy is the ratio of total enthalpy to total weight, that is, enthalpy per unit weight of substance, Btu per pound.

Entropy: A thermodynamic property which, for practical purposes, is best defined by stating its principal functions: (1) during a reversible *adiabatic* change of state, entropy is constant; (2) during a reversible *isothermal* change of state, the heat absorbed is equal to absolute temperature times change of entropy. *Specific entropy* is the ratio of total entropy to total weight, that is, entropy per unit weight, Btu per degree Fahrenheit per pound.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and atmospheric pressure.

Estimated Design Load: The sum of the heat emission of the equivalent direct radiation to be installed plus the allowance for heat loss of the connecting piping plus the heat requirements of any auxiliary apparatus connected with the system.

Estimated Maximum Load: The load stated in Btu per hour or equivalent direct radiation that has been estimated to be the greatest or maximum load that the boiler will be called upon to carry.

Extended Heating Surface: See *Heating Surface.*

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being

soldered to the core or by both pressure and soldering. An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.

Fan Furnace System: See *Warm Air Heating System.*

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = (WV) \div (gt)$.

Free Enthalpy: A thermodynamic property which serves as a measure of the available energy of a system with respect to surroundings at the same temperature and same pressure as that of the system. No process involving an increase in available energy can occur spontaneously. (See example on *Free Enthalpy* in Chapter 1.)

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also, a fire-pot.

Furnace Volume (*Total*): The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i.e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929.)

Gage Pressure: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as previously defined.

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Heat: Heat is that form of energy which transfers from one system to a second system at lower temperature by virtue of the temperature difference, when the two are brought into communication.

Heating Medium: A substance such as water, steam, air, or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface* (*or extended surface*): Heating surface having air on both sides and heated by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *Boiler Heating Surface.*)

Heat of the Liquid: This can usually be interpreted as the specific enthalpy of saturated liquid.

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humid Heat: Ratio of increase of enthalpy per pound of dry air to rise of temperature under conditions of constant pressure and constant humidity ratio.

Humidify: To add water vapor to the atmosphere; to add water vapor or moisture to any material.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor when mixed with dry air or other dilutent gases. *Absolute humidity* is the weight of water vapor per unit volume of moist air, pounds per cubic foot. It can be calculated by dividing the humidity ratio, weight of water vapor per pound of dry air, by the volume of the mixture per pound of dry air. *Relative humidity* is the ratio of the partial pressure of the water vapor in the air to the saturation pressure of pure water corresponding to the actual temperature. (See Chapter 1.)

Humidity Ratio: Weight of water vapor per pound of dry air. (Formerly called specific humidity.)

Hygrostat: Same as *Humidistat.*

Inch of Water: The pressure due to a column of liquid water one inch high at a temperature of 60 F.

Insulation (*Heat*)**:** A material having a relatively high heat-resistance per unit of thickness.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Latent Heat: The most general interpretation is heat absorbed at constant temperature. More specifically the *latent heat of vaporization* is the difference between the specific enthalpies of saturated vapor and saturated liquid at the same temperature (and, for a pure substance, the same pressure). *Latent heat of sublimation* is the difference between the specific enthalpies of saturated vapor and saturated solid at the same temperature. *Latent heat of fusion* is the difference between the specific enthalpies of saturated liquid and saturated solid at the same temperature.

Laws of Thermodynamics: The Law of Conservation of Energy states that energy, in any of its forms, can neither be created nor destroyed. As a corollary to this, the *First Law* of Thermodynamics states that in any power cycle or refrigeration cycle the net heat absorbed by the working substance is exactly equal to the net work done. The *Second Law* of Thermodynamics states that a power cycle which absorbs heat at a single temperature and converts it wholly into work, as required by the First Law, is impossible; hence it is absolutely necessary to reject heat at some lower temperature if any work is to be done. The Second Law further prescribes the least possible quantity of heat that must be so rejected depending on the two temperatures involved.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second. $m = W \div g$.

Mb, Mbh: Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour.

Mechanical Equivalent of Heat: The conversion factor from Btu to foot pounds; $J = 778.26$ foot pounds per Btu. This is also referred to as Joule's Equivalent.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol (*Pound Mol*)**:** A weight in pounds numerically equal to the molecular weight of a substance. In the case of gases, and at not too high pressures, the volume of 1 mol is approximately the same for any gas at the same temperature and pressure. At 32 F and standard atmospheric pressure this volume is 358.65 cu ft.

One-Pipe Supply Riser (*Steam*)**:** A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System (*Hot Water*)**:** A hot water system in which the water flows through more than one heating unit before it returns to the boiler; consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

One-Pipe System (*Steam*)**:** A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe, usually in opposite directions. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

Overhead System: Any steam or hot water system in which the supply main is above the heating units. With a steam system the return must be below the heating units; with a water system, the return *may* be above the heating units.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

Panel Warming: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room, so that the heating process takes place mainly by radiation from the wall, floor or ceiling surfaces to the objects in the room.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Potentiometer: An instrument for measuring or comparing small electromotive forces.

Power: The rate of performing work; usually expressed in units of horsepower, Btu per hour, or watts.

Prime Surface: See *Heating Surface.*

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmosphere as to moisture. *Psychrometry:* The branch of physics that treats of the measurement of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects *it can see* and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called *radiator* is also a *convector* but the single term *radiator* has been established by long usage.

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Relative Humidity: See *Humidity;* also discussion relative humidity, Chapter 1.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reversed-Return System (*Hot Water*)*:* A hot water heating system in which the water from several heating units is returned along paths arranged so that all circuits composing the system or composing a major sub-division of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to facilitate egress of air.

Saturated Air: A mixture of dry air and saturated water vapor, all at the same dry-bulb temperature. It may also be considered as air containing the maximum possible amount of water vapor at a given temperature without becoming supersaturated.

Saturation: The condition for coexistence in stable equilibrium of two or more distinct phases, such as steam over the water from which it is being generated.

Saturation Pressure: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid or vapor and solid can coexist in stable equilibrium.

Sensible Heat: Heat which manifests itself by temperature change.

Smoke: Carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Specific Enthalpy: The ratio of total enthalpy to total weight. The specific enthalpy of air is its enthalpy, Btu per pound, measured above 0 F and 29.921 in. Hg as a reference point. The specific enthalpy of water is its enthalpy, Btu per pound, measured from the reference point of saturated liquid at 32 F. (See *Enthalpy.*)

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The ratio of heat absorbed per unit weight of substance to temperature rise. For gases, both specific heat at constant pressure, c_p, and specific heat at constant volume, c_v, are frequently given. In air conditioning, c_p is usually used.

Specific Volume: The volume, expressed in cubic feet, of one pound of a substance. $v = 1 \div d = V \div W$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point.

Square Foot of Heating Surface (*Equivalent*)*:* Equivalent Direct Radiation (EDR). That amount of heating surface which will give off 240 Btu per hour. The *equivalent* square feet of heating surface may have no direct relation to the actual surface area.

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Standard Air: Air weighing 0.07488 lb per cubic foot, which is air at 68 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.92 in. of mercury. (Most engineering tables and formulae involving the weight of air are based on air weighing 0.07492 lb per cubic foot, which is dry air at 70 F dry-bulb with a barometric pressure of 29.921 in. of mercury. The error involved in disregarding the difference between these two weights is very slight and in most instances may be neglected.)

Static Pressure: The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practi-

cally, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances created by inserting the tube cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Steam: Water in the vapor phase. *Dry Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing no water in suspension. *Wet Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing water particles in suspension. *Superheated Steam* is steam at a temperature higher than the saturation temperature corresponding to the pressure.

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of steam to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate and preventing the passage of steam, or for allowing the passage of air as well as condensate.

Superheated Steam: See *Steam.*

Supply Mains (*Steam*)*:* The pipes through which the steam flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction, and convection *from a surface to the air or liquid surrounding it*, or vice versa, in one hour per square foot of surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid.

Therm: 100,000 Btu. (Used in the gas industry.)

Thermal Resistance: The reciprocal of *conductance.*

Thermal Resistivity: The reciprocal of *conductivity.*

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Ton of Refrigeration: The removal of 12,000 Btu of heat per hour at a low temperature.

Ton Day of Refrigeration: The removal of 288,000 Btu of heat at a low temperature.

Total Heat: This can usually be interpreted as increase of enthalpy at constant pressure. It is often regarded as snynonymous with enthalpy.

Total Pressure: In the theory of the flow of fluids; the sum of the static pressure and the velocity pressure at the point of measurement.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

Two-Pipe System (*Steam or Water*)*:* A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Underfeed Distribution System (*Hot Water*)*:* A hot water heating system in which the main flow pipe is below the heating unit.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit: As applied to heating, ventilating and air conditioning equipment this word means a factory-built and assembled equipment with apparatus for accomplishing some specified function or combination of functions. (See Chapters 21 and 22.)

It is loosely applied to a great variety of equipment. Usually the function is included in the name, and hence come terms like Unit Heater, Unit Ventilator, Humidifying Unit, and Air Conditioning Unit.

Units are said to be ***direct*** or ***room***, when intended for location, or located in, the treated space; ***indirect*** or ***remote***, when outside or adjacent to the treated space. They are ***ceiling*** units when suspended from above, and ***floor*** when supported from below. Other descriptive words include ***free delivery*** when the unit is not intended to be attached to ducts or similar resistance-producing devices, and ***pressure*** when for use with such ducts. Complete description requires the use of several of these qualifying words or phrases. (See Chapter 22.)

Up-Feed System (*Steam*)*:* A steam heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vapor: Any substance in the gaseous state.

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Vapor Pressure: Synonymous with saturation pressure in the case of a pure substance.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Pressure: The difference due to velocity between total pressure and static pressure. It is supposed to equal the kinetic energy per unit volume of the fluid at the point of measurement.

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning.*)

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers and filters.

Wet-Bulb Temperature: *Thermodynamic wet-bulb temperature* is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. *Wet-bulb temperature* (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications. (*A.S.M.E.* Power Test Codes, Series 1932, Instruments and Apparatus, Part 18.)

Wet Return: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. (See *Dry Return.*)

ABBREVIATIONS[1]

Absolute	abs
Acceleration, due to gravity	*g*
Acceleration, linear	*a*
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Area	*A*
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiler pressure	bp
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
Calorie	cal
Centigram	cg
Centimeter	cm

[1]From compilations of abbreviations approved by the *American Standards Association*, Z, 10 a, c, f, and As a general rule the period is omitted in all abbreviations except where the omission results in the formation of an English word.

Centimeter-gram-second (system)....cgs
Change in specific volume during vaporization....v_{fg}
Cubic....cu
Cubic foot....cu ft
Cubic feet per minute....cfm
Cubic feet per second....cfs
Decibel....db
Degree[2]....deg or °
Degree centigrade....C
Degree Fahrenheit....F
Degree Kelvin....K
Degree Réaumur....R
Density, Weight per unit volume, Specific weight....d or ρ (rho)

$$d = \frac{1}{v}$$

Diameter....D or diam
Direct-current (as adjective)....d-c
Distance, linear....s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, vapor in contact with liquid....*Subscript* g
Entropy. (The capital should be used for any weight, and the small letter for unit weight)....S or s
Feet per minute....fpm
Feet per second....fps
Foot....ft
Foot-pound....ft-lb
Foot-pound-second (system)....fps
Force, total load....F
Freezing point....fp
Gallon....gal
Gallons per minute....gpm
Gallons per second....gps
Gram....g
Gram-calorie....g-cal
Head....H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight)....H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid....h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor....h_g
Heat of vaporization at constant pressure....L or h_{fg}
Horsepower....hp
Horsepower-hour....hp-hr
Hour....hr
Inch....in.
Inch-pound....in.-lb
Indicated horsepower....ihp
Indicated horsepower-hour....ihp-hr
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight)....U or u
Kilogram....kg
Kilowatt....kw
Kilowatthour....kwhr
Length of path of heat flow, thickness....L
Load, total....W
Mass....mass
Mechanical efficiency....e_m
Mechanical equivalent of heat....J
Melting point....mp
Meter....m

[2]It is recommended that the abbreviation for the temperature scale, F, C, K, be included in expressions for numerical temperatures but, wherever feasible, the abbreviations for *degree* be omitted; as 68 F.

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degrees per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* $_{fg}$
Velocity V
Volume (total) V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight .... q
Watt .... w
Watthour .... whr
Weight of a major item, Total weight .... W
Weight rate, Weight per unit of power, Weight per unit of time .... w
Work (total) .... W

CONVERSION EQUATIONS

Heat, Power and Work

1 ton refrigeration	= 12,000 Btu per hour 200 Btu per minute
Latent heat of ice	= 143.4 Btu per pound
1 Btu	= 778.26 ft-lb 0.293 whr 252 mean calories
1 watthour	= 2,655 ft-lb 3.413 Btu 3600 joules 860 mean calories
1 kilowatthour	= 3,413 Btu 3.517 lb water evaporated from and at 212 F
1 kilowatt (1000 watts)	= 1.341 hp 56.88 Btu per minute 44,253 ft-lb per minute
1000 mean calorie 1 kilogram calorie	= 3.969 Btu 3087 ft-lb 1.1627 whr
1 horsepower	= 0.746 kw 42.42 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,475 Btu per hour 9.808 kw

Weight and Volume

1 gal (U. S.)	= 231 cu in. 0.1337 cu ft
1 British or Imperial gallon	= 277.42 cu in.
1 cu ft	= 7.48 gal 1728 cu in.
1 cu ft water at 60 F	= 62.37 lb
1 cu ft water at 212 F	= 59.83 lb
1 gal water at 60 F	= 8.34 lb
1 gal water at 212 F	= 7.998 lb
1 lb (avdp)	= 16 oz 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb

Pressure

1 lb per square inch	=	144 lb per square foot 2.0421 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	=	0.1276 in. mercury at 62 F 1.732 in. water at 62 F
1 atmosphere	=	14.6959 lb per square inch 2117 lb per square foot 33.9 ft water at 62 F 30 in. mercury at 62 F 29.921 in. mercury at 32 F
1 in. water at 62 F	=	0.03609 lb per square inch 0.5774 oz per square inch 5.196 lb per square foot
1 ft water at 62 F	=	0.433 lb per square inch 62.35 lb per square foot
1 in. mercury at 62 F	=	0.491 lb per square inch 7.84 oz per square inch 1.131 ft water at 62 F 13.58 in. water at 62 F

Metric Units

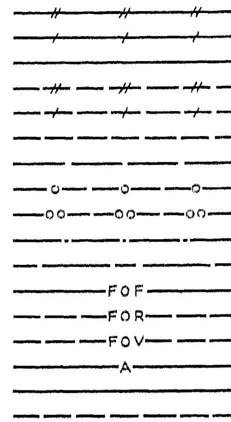
1 cm	=	0.3937 in.
1 in.	=	2.540 cm
1 m	=	3.281 ft
1 ft	=	0.3048 m
1 sq cm	=	0.155 sq in.
1 sq in.	=	6.452 sq cm
1 sq m	=	10.76 sq ft
1 sq ft	=	0.0929 sq m
1 cu cm	=	0.06102 cu in.
1 cu in.	=	16.39 cu cm
1 cu m	=	35.31 cu ft
1 cu ft	=	0.02831 cu m
1 liter	=	1000 cu cm = 0.2642 gal
1 kg	=	2.205 lb (avdp)
1 lb	=	0.4536 kg
1 metric ton	=	2205 lb (avdp)
1 gram	=	0.002205 lb (avdp)
1 kilometer per hour	=	0.6214 mph
1 gram per square centimeter	=	0.02905 in. mercury at 62 F 0.3944 in. water at 62 F
1 kg per sq cm (metric atmosphere)	=	14.22 lb per square inch
1 gram per cubic centimeter	=	0.03613 lb per cubic inch 62.43 lb per cubic foot
1 dyne	=	0.00007233 poundals
1 joule	=	10,000,000 ergs 0.7376 ft-lb
1 metric horsepower	=	75 kg-m per second 0.986 hp (U. S.)
1 kilogram-calorie per kilogram	=	1.8 Btu per pound
1 gram-calorie per square centimeter	=	3.687 Btu per square foot
1 gram-calorie per square centimeter per centimeter	=	1.452 Btu per sq ft per inch
1 gram-calorie per second per square centimeter for a temperature gradient of 1 deg C per centimeter.	=	2903 Btu per hour per square foot for a temperature gradient of 1 deg F per inch of thickness.

Graphical Symbols for Drawings

Piping

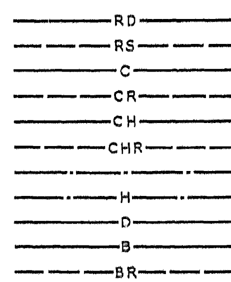
Heating

1. High Pressure Steam
2. Medium Pressure Steam
3. Low Pressure Steam
4. High Pressure Return
5. Medium Pressure Return
6. Low Pressure Return
7. Boiler Blow Off
8. Condensate or Vacuum Pump Discharge
9. Feedwater Pump Discharge
10. Make Up Water
11. Air Relief Line
12. Fuel Oil Flow
13. Fuel Oil Return
14. Fuel Oil Tank Vent
15. Compressed Air
16. Hot Water Heating Supply
17. Hot Water Heating Return



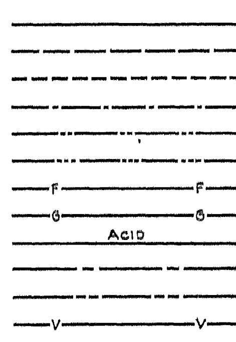
Air Conditioning

18. Refrigerant Discharge
19. Refrigerant Suction
20. Condenser Water Flow
21. Condenser Water Return
22. Circulating Chilled or Hot Water Flow
23. Circulating Chilled or Hot Water Return
24. Make Up Water
25. Humidification Line
26. Drain
27. Brine Supply
28. Brine Return



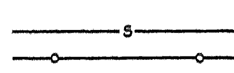
Plumbing

29. Soil, Waste or Leader (Above Grade)
30. Soil, Waste or Leader (Below Grade)
31. Vent
32. Cold Water
33. Hot Water
34. Hot Water Return
35. Fire Line
36. Gas
37. Acid Waste
38. Drinking Water Flow
39. Drinking Water Return
40. Vacuum Cleaning
41. Compressed Air



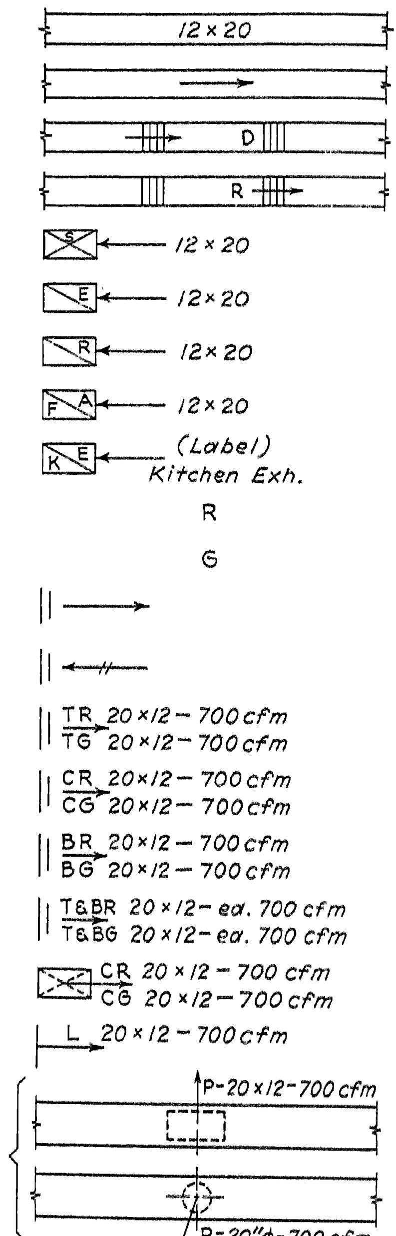
Sprinklers

42. Main Supplies
43. Branch and Head
44. Drain



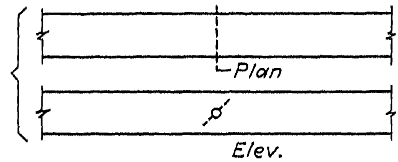
GRAPHICAL SYMBOLS FOR DRAWINGS *Ductwork*

45. Duct (1st Figure, Width; 2nd, Depth)
46. Direction of Flow
47. Inclined Drop in Respect to Air Flow
48. Inclined Rise in Respect to Air Flow
49. Supply Duct Section
50. Exhaust Duct Section
51. Recirculation Duct Section
52. Fresh Air Duct Section
53. Other Duct Sections
54. Register
55. Grille
56. Supply Outlet
57. Exhaust Inlet
58. Top Register or Grille
59. Center Register or Grille
60. Bottom Register or Grille
61. Top and Bottom Register or Grille
62. Ceiling Register or Grille
63. Louver Opening
64. Adjustable Plaque

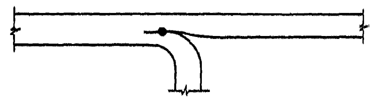


GRAPHICAL SYMBOLS FOR DRAWINGS *Ductwork*

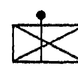
65. Volume Damper



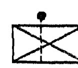
66. Deflecting Damper



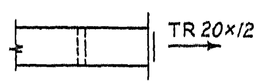
67. Deflecting Damper, Up



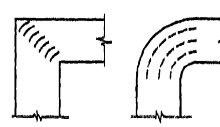
68. Deflecting Damper, Down



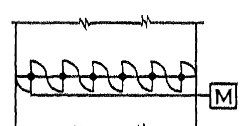
69. Adjustable Blank Off



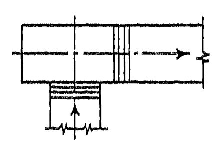
70. Turning Vanes



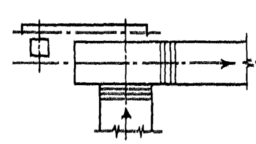
71. Automatic Dampers



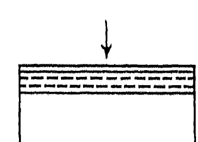
72. Canvas Connections



73. Fan and Motor With Guard



74. Intake Louvers and Screen



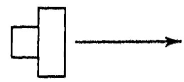
Graphical Symbols for Drawings

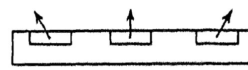
Heating and Ventilating

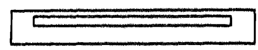
75. Heat Transfer Surface, Plan


76. Wall Radiator, Plan

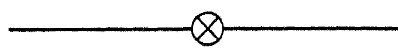

77. Wall Radiator on Ceiling, Plan


78. Unit Heater (Propeller), Plan


79. Unit Heater (Centrifugal Fan), Plan


80. Unit Ventilator, Plan


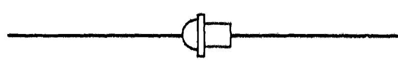
Traps

81. Thermostatic


82. Blast Thermostatic

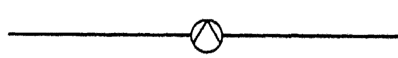

83. Float and Thermostatic


84. Float

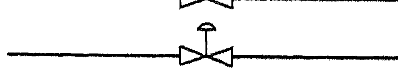

85. Boiler Return


Valves

86. Reducing Pressure


87. Air Line


88. Lock and Shield


89. Diaphragm


90. Air Eliminator


91. Strainer


92. Thermometer


93. Thermostat


Table 1. Specific Heat of Solids

Materials	Temperature F	Specific Heat	Authority
Alloys			
Brass, Red	32	0.0899	S
Brass, Yellow	32	0.0883	S
Bronze (80*Cu*, 20*Sn*)	57–208	0.0862	S
Monel Metal	68–2370	0.127	S
Aluminum	80–212	0.212	S
Asbestos	68–208	0.195	S
Brickwork	....	0.195	H
Carbon (Graphite)	104–1637	0.314	I
Coal	....	0.278	H
Coke	....	0.201	H
Concrete	....	0.270	H
Copper	64–212	0.0928	S
Fire Clay Brick	77–1832	0.258	I
Glass			
Crown	50–122	0.161	S
Flint	50–122	0.117	S
Gold	64	0.0312	S
Gypsum	....	0.259	H
Ice	32	0.487	S
Ice	−40	0.434	S
Iron, Pure	32	0.1043	S
Iron, Pure	32–600	0.127	M
Iron, Cast	68–212	0.1189	H
Iron, Wrought	59–212	0.1152	H
Lead	32	0.0297	S
Nickel	32	0.1032	S
Masonry	....	0.2159	H
Plaster	....	0.2	H
Platinum	58–212	0.0319	S
Rocks			
Gneiss	63–210	0.196	S
Granite	54–212	0.192	S
Limestone	59–212	0.216	S
Marble	32–212	0.21	S
Sandstone	....	0.22	S
Silver	32	0.0536	S
Steel	....	0.1175	H
Sulphur	240–320	0.220	S
Silica Brick	77–1832	0.263	I
Tin	77	0.0548	S
Woods (Average)	68	0.327	S
Zinc	32	0.0913	S

Table 2. Specific Heat of Liquids

Liquid	Temperature F	Specific Heat	Authority
Alcohol, Ethyl	32	0.548	S
Alcohol, Methyl	50–68	0.601	S
Glycerine	59–122	0.576	S
Lead (Molten)	360	0.041	H
Mercury	68	0.03325	S
Petroleum	70–136	0.511	S
Sea Water			
Sp. Gr. 1.0043	64	0.980	S
Sp. Gr. 1.0463	64	0.903	S
Water	59	1.000	S

Table 3. Specific Heat of Gases and Vapors

Substance	Temperature F	Specific Heat at Constant Pressure	Ratio of Specific Heat C_p/C_v	Specific Heat at Constant Volume (Computed)	Authority
Air	32–392	0.2375	1.405	0.169	S
Ammonia	80–392	0.5356	1.277	0.419	S
Carbon Dioxide	52–417	0.2169	1.3003	0.1668	S
Carbon Monoxide	79–388	0.2426	1.395	0.1736	S
Coal Gas	68–1900	0.3145	....	....	S
Flue Gas	....	0.24 (Approx.)	....	....	H
Hydrogen	70–212	3.41	1.419	2.402	S
Nitrogen	32–392	0.2438	1.41	0.1729	S
Oxygen	55–404	0.2175	1.3977	0.155	S
Water Vapor	212	0.421	1.305	0.322	S
Water Vapor	356	0.51	....	....	S

Notes: When one temperature is given the true specific heat is given, otherwise the value is the mean specific heat between the given limits.

Authorities: S—Smithsonian Physical Tables, 1933; I—International Critical Tables; H—Heating, Ventilation and Air Conditioning, by L. A. Harding and A. C. Willard; M—Engineers' Handbook, by Lionel S. Marks.

Table 4. Circumferences and Areas of Circles

Diameter in Inches	Area Sq In.	Area Sq Ft	Circumference Inches	Circumference Feet	Diameter in Inches	Area Sq In.	Area Sq Ft	Circumference Inches	Circumference Feet
¼	0.049	0.0003	0.785	0.0652	28	615.8	4.276	87.97	7.330
½	0.196	0.0014	1.571	0.1309	28½	637.9	4.430	89.54	7.462
¾	0.442	0.0031	2.356	0.1964	29	660.52	4.587	91.11	7.592
1	0.785	0.0054	3.142	0.2618	29½	683.5	4.747	92.63	7.725
1¼	1.227	0.0085	3.927	0.3273	30	706.8	4.909	94.25	7.854
1½	1.767	0.0123	4.712	0.3927	31	754.8	5.241	97.39	8.116
1¾	2.405	0.0167	5.498	0.4582	32	804.3	5.585	100.5	8.378
2	3.142	0.0218	6.283	0.5236	33	855.3	5.940	103.7	8.639
2¼	3.976	0.0276	7.069	0.5891	34	907.9	6.305	106.8	8.901
2½	4.909	0.0341	7.854	0.6546	35	962.1	6.681	109.9	9.163
2¾	5.939	0.0412	8.639	0.7200	36	1018.0	7.069	113.1	9.425
3	7.069	0.0491	9.425	0.7854	37	1075.0	7.467	116.2	9.686
3¼	8.296	0.0576	10.21	0.8510	38	1134.0	7.876	119.4	9.948
3½	9.621	0.0668	10.99	0.9160	39	1195.0	8.296	122.5	10.21
3¾	11.04	0.0767	11.78	0.9818	40	1256.0	8.727	125.6	10.47
4	12.57	0.0873	12.57	1.047	41	1320.0	9.168	128.8	10.73
4¼	14.19	0.0986	13.35	1.113	42	1385.0	9.621	131.9	10.99
4½	15.90	0.1104	14.14	1.178	43	1452.0	10.08	135.1	11.26
4¾	17.72	0.1231	14.92	1.243	44	1521.0	10.56	138.2	11.52
5	19.64	0.1364	15.71	1.309	45	1590.0	11.04	141.4	11.78
5¼	21.65	0.1504	16.49	1.374	46	1662.0	11.54	144.5	12.04
5½	23.76	0.1650	17.28	1.440	47	1735.0	12.05	147.7	12.30
5¾	25.97	0.1840	18.06	1.505	48	1810.0	12.51	150.8	12.57
6	28.27	0.1964	18.85	1.571	49	1886.0	13.09	153.9	12.83
6¼	30.68	0.2131	19.64	1.637	50	1963.0	13.64	157.1	13.09
6½	33.18	0.2304	20.42	1.702	51	2043.0	14.19	160.2	13.35
6¾	35.79	0.2486	21.21	1.768	52	2124.0	14.75	163.4	13.61
7	38.49	0.2673	21.99	1.833	53	2206.0	15.32	166.5	13.88
7¼	41.28	0.2867	22.78	1.899	54	2290.0	15.90	169.6	14.14
7½	44.18	0.3068	23.56	1.964	55	2376.0	16.50	172.8	14.40
7¾	47.17	0.3276	24.35	2.029	56	2463.0	17.10	175.9	14.66
8	50.27	0.3491	25.13	2.094	57	2552.0	17.72	179.1	14.92
8¼	53.46	0.3713	25.92	2.160	58	2642.0	18.35	182.2	15.18
8½	56.75	0 3942	26.70	2.225	59	2734.0	18.99	185.4	15.45
8¾	60.13	0.4175	27.49	2.291	60	2827.0	19.63	188.5	15.71
9	63.62	0.4418	28.27	2.356	61	2922.0	20.29	191.6	15.97
9¼	67.20	0.4668	29.06	2.422	62	3019.0	20.97	194.8	16.23
9½	70.88	0.4923	29.85	2.488	63	3117.0	21.65	197.9	16.49
9¾	74.66	0.5185	30.63	2.553	64	3217.0	22.34	201.1	16.76
10	78.54	0.5454	31.42	2.618	65	3318.0	23.04	204.2	17.02
10½	86.59	0.6010	32.99	2.750	66	3421.0	23.76	207.3	17.28
11	95.03	0.6600	34.56	2.880	67	3526.0	24.48	210.5	17.54
11½	103.9	0.7215	36.13	3.011	68	3632.0	25.22	213.6	17.80
12	113.1	0.7854	37.70	3.142	69	3739.0	25.97	216.8	18.06
12½	122.7	0.8520	39.27	3.273	70	3848.0	26.73	219.9	18.33
13	132.7	0.9218	40.84	3.403	71	3959.0	27.49	223.1	18.59
13½	143.1	0.9937	42.41	3.535	72	4072.0	28.27	226.2	18.85
14	153.9	1.069	43.98	3.665	73	4185.0	29.07	229.3	19.11
14½	165.1	1.146	45.55	3.796	74	4301.0	29.87	232.5	19.37
15	176.7	1.227	47.12	3.927	75	4418.0	30.68	235.6	19.63
15½	188.7	1.310	48.69	4.058	76	4536.0	31.50	238.8	19.90
16	201.1	1.396	50.27	4.189	77	4657.0	32.34	241.9	20.16
16½	213.8	1.485	51.84	4.321	78	4778.0	33.18	245.0	20.42
17	226.9	1.576	53.41	4.451	79	4902.0	34.04	248.2	20.68
17½	240.5	1.670	54.98	4.582	80	5027.0	34.91	251.3	20.94
18	254.5	1.767	56.55	4.712	81	5153.0	35.78	254.5	21.21
18½	268.8	1.867	58.12	4.845	82	5281.0	36.67	257.6	21.47
19	283.5	1.969	59.69	4.974	83	5411.0	37.57	260.8	21.73
19½	298.6	2.074	61.26	5.105	84	5542.0	38.48	263.9	21.99
20	314.2	2.182	62.83	5.236	85	5675.0	39.41	267.0	22.25
20½	330.1	2.293	64.40	5.367	86	5809.0	40.34	270.2	22.51
21	346.4	2.405	65.97	5.498	87	5945.0	41.28	273.3	22.78
21½	361.1	2.508	67.54	5.629	88	6082.0	42.24	276.5	23.04
22	380.1	2.640	69.12	5.760	89	6221.0	43.20	279.6	23.30
22½	397.6	2.761	70.69	5.891	90	6362.0	44.18	282.7	23.56
23	415.5	2.885	72.26	6.021	91	6504.0	45.17	285.9	23.82
23½	433.7	3.012	73.83	6.153	92	6648.0	46.16	289.0	24.09
24	452.4	3.142	75.40	6.283	93	6793.0	47.17	292.2	24.35
24½	471.4	3.274	76.97	6.415	94	6940.0	48.19	295.3	24.61
25	490.9	3.409	78.54	6.545	95	7088.0	49.22	298.4	24.87
25½	510.7	3.547	80.11	6.676	96	7238.0	50.27	301.6	25.13
26	530.9	3.687	81.68	6.807	97	7390.0	51.32	304.7	25.39
26½	551.6	3.832	83.25	6.938	98	7543.0	52.38	307.9	25.66
27	572.6	3.976	84.82	7.069	99	7698.0	53.46	311.0	25.92
27½	593.9	4.125	86.39	7.199	100	7854.0	54.54	314.2	26.18

CATALOG DATA SECTION

The HEATING VENTILATING AIR CONDITIONING GUIDE 1941

INDEX TO ADVERTISERS

PAGE 811

INDEX TO MODERN EQUIPMENT

PAGE 1097

ALPHABETICAL INDEX TO ADVERTISERS

HEATING VENTILATING AIR CONDITIONING GUIDE, 1941

A

B

C

D

D

E

F

G

H

T

U

V

W

Y

AIR CONDITIONING

•

Equipment for complete air conditioning systems, consisting of an assembly of apparatus for air circulation, air cleaning and heat transfer, with control apparatus for maintaining temperature and humidity within prescribed limits, has many commercial, comfort, and industrial applications. Systems for all-year, winter and summer service, and special processing work are presented in four divisions . . . Pages 818-860.

CENTRAL SYSTEMS (p. 818-829)

Complete assembly of supply and return ducts serving one or more spaces, connected with some or all of the following equipment: fans, motors, heat transfer surfaces, humidifiers, dehumidifiers, refrigeration machinery, air cleaning devices and control equipment.

An outline of the design procedure generally used to create a modern central air conditioning system is given in Chapter 20 of the Technical Data Section.

DIRECT FIRED UNITS (p. 830-836)

Automatic heating and comfort air conditioning apparatus suitable for residential and small commercial applications designed to give results similar to the larger central systems provide direct fired oil, gas or coal heating units, filtration, fan controls, etc.

The Technical Data Section, Chapters 9, 11 and 19 cover this type of equipment.

FAN-FURNACE SYSTEMS (p. 837-840)

Winter air conditioning and summer ventilation for residences are provided by Automatic fired fan-furnace systems. As in the larger central systems these installations clean, heat and humidify the air, and if desired, auxiliary units will provide cooling.

In Chapter 19 on Mechanical Warm Air Furnace Systems will be found details of the design of this type of system.

UNIT HEATERS, COOLERS (p. 841-860)

For complete or partial air conditioning there are a variety of self-contained units. Such units may be complete in themselves, employing their own direct means of air cleaning, heating distribution and source of refrigeration.

The various functional elements of unitary equipment are given in Chapters 21 and 22, for Unit Heaters, Ventilators, Humidifiers, Conditioning and Cooling Units and Attic Fans.

Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

American Blower Corporation

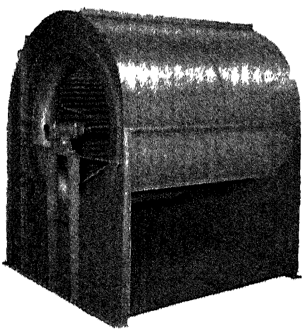
Division of American Radiator and Standard Sanitary Corporation

General Offices and Factory

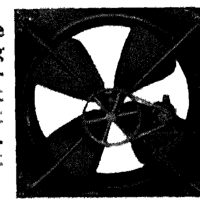
Detroit, Mich.

Branches in All Principal Cities

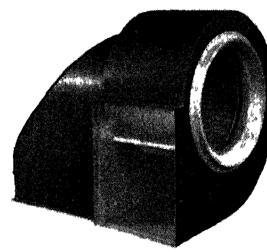
AIR CONDITIONING — HUMIDIFYING — DEHUMIDIFYING — COOLING — VENTILATING — HEATING — VAPOR-ABSORPTION — DRYING — AIR WASHING AND PURIFICATION — EXHAUSTING EQUIPMENT AND MECHANICAL DRAFT APPARATUS.



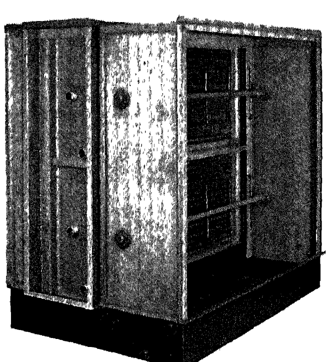
Double Inlet "ABC" Multiblade Fan—above, is a heavy duty ventilating fan. Its wheel has narrow, forward pitched blades. Low tip speeds assure quiet operation. Request Bulletin A-701. Write for Bulletin A-403 for backwardly inclined, non-overloading H. S. Fan.



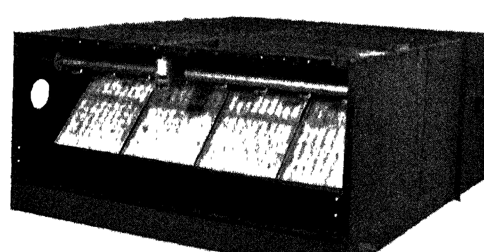
Commercial V-Belt Drive Ventura Fans—for ventilating applications without duct systems, where extremely quiet operation is desired. Inlet-outlet streamlined for high efficiency. Also direct connected. Request Bulletin A-31029.



"ABC" Utility Sets—right, complete packaged units, direct connected or V-Belt short coupled drive, for duct applications. Famous "ABC" Multiblade Wheel operates at low tip speeds. Quiet, compact. Bulletin A-31029.



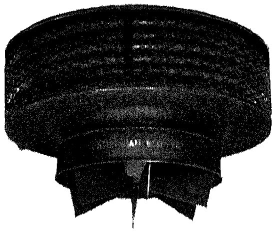
American Blower Air Washer—above, cleans, purifies and freshens the air, removes dust, odors and bacteria, cools if desired and provides an effective method of controlling humidity. Bulletin 3623.



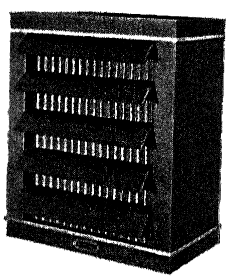
American Blower Capillary Air Washers—above, for high efficiency in cleaning, humidification, cooling and dehumidification of air. A highly efficient surface contact mechanism, the capillary cell, is used. Air is forced at low resistance through long, irregular passages of small size formed by a large amount of thoroughly wetted glass surface. Unit includes a substantial metal casing and tank of air washer design, capillary cells, improved low head sprays, metal or glass fibre low resistance moisture eliminators, non-ferrous, extended surface cooling or heating coils. Write for Bulletin 3723.

TYPES OF AMERICAN BLOWER CORPORATION AIR HANDLING AND CONDITIONING EQUIPMENT

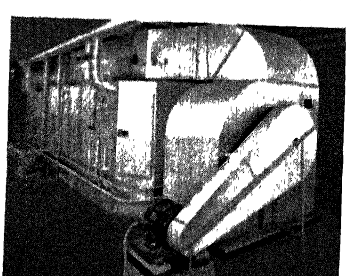
All types of air handling and air conditioning equipment for industrial applications, process work, drying, cooling; also equipment for stores, offices, shops, public buildings, power plants, etc., and attic ventilation for homes.



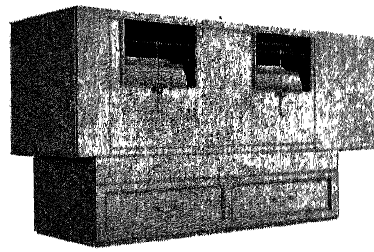
"ABC" Vertical Heaters—for ceiling applications, give an even, wide floor area distribution of heat. For either steam or hot water heating systems. Variable speed, 2-speed and constant speed models. Write for Bulletin A-9418.



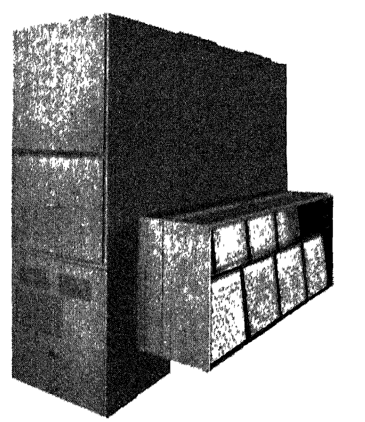
Venturafin Unit Heaters—for many general purpose heating jobs. Wall or ceiling mounting. Streamline construction, rugged heating elements. Steam or hot water. Write for Bulletin A-8218.



Air Conditioning Central Systems—provide an effective way of cooling, heating, humidifying, dehumidifying and purifying air in all classes of business and public buildings where a dust system is desirable. Write for Special Data.



"HV" General Purpose Units—with air filters and Aileron control. Ideal wherever attractive, quiet and economical heating and ventilating units are required. Wall, floor or ceiling mounting. Offer great flexibility of design and arrangement to meet specific needs. Write for Bulletin 5927.



American Blower Series "H" Air Conditioners with Sprayed Coils—are usually applied for industrial uses where air washing and evaporative cooling are required. Sprayed coils give cleaner air, cut coil maintenance and refrigeration costs, reduce necessary air volumes, permit use of smaller ducts and grilles. Horizontal or floor types (as shown). Aileron control provides simple method of regulating flow of air from the fans. Write for Bulletin 6027.

Clarage Fan Company

Kalamazoo, Michigan

Sales Engineering Offices in All Principal Cities

(Consult Telephone Directory)

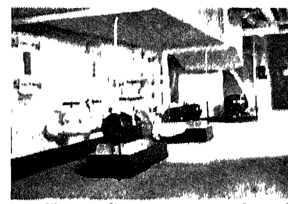
CLARAGE AIR HANDLING AND CONDITIONING EQUIPMENT

For Over a Quarter-Century Clarage has been a leading manufacturer of air handling and conditioning equipment. There is a Clarage fan or blower, conditioning unit or system to meet every need, from the simplest ventilating or cooling job to the most exacting temperature and humidity control installation.

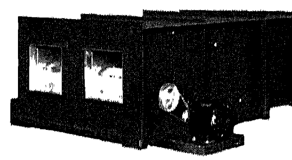
Whatever your ventilating, unit heating, cooling, drying, air cleaning, humidifying, dehumidifying or complete air conditioning problem, we can meet your requirements successfully and economically.

Clarage Experience covers every conceivable type of installation, commercial, industrial and public building. Clarage equipment is used in the largest industrial plants, office buildings, auditoriums, theatres, hotels, restaurants, retail stores, hospitals, churches and schools.

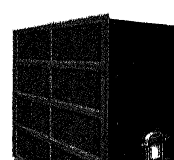
Architects, Engineers and Contractors find our service specially helpful. This Company is an independent manufacturer selling through regular trade channels, and cooperating fully with those who specify and those who install. Your inquiry for data on any Clarage product is invited. Write for Bulletins.



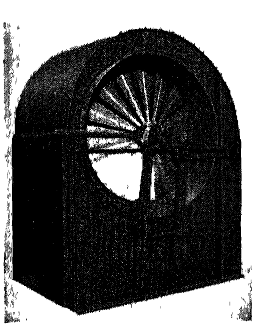
Clarage Systems for complete air conditioning in public buildings and industrial plants.



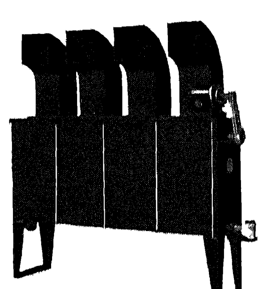
Multitherm Units for complete conditioning, summer cooling, or winter heating.



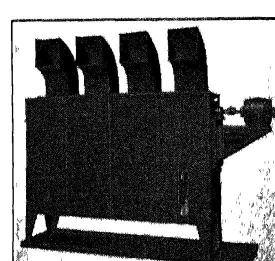
Unicoil Units used in conditioning systems for air cleaning, cooling, heating and humidity control.



Clarage Fan with Vortex (constant speed) Volume Control for ventilation and air conditioning.



Unitherm Unit Heaters with Syncrotherm Temperature Control for factory heating.



Unitherm Unit Coolers for product cooling and refrigeration.

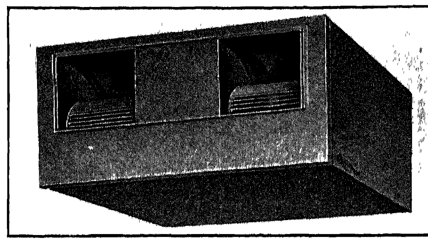
Refrigeration Economics Co., Inc.

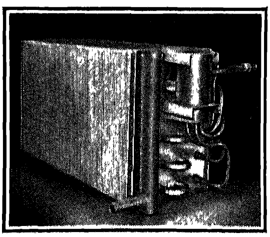
Canton, Ohio

RECOY PRODUCTS

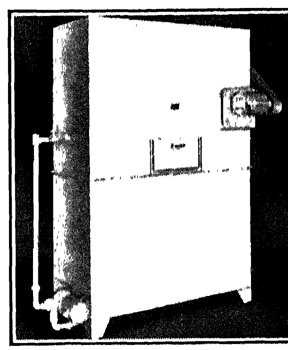
RECOY AIR CONDITIONING UNITS of the suspended type as shown, or vertical floor type, are made for all season purposes, also for summer cooling or winter heating and humidifying.

Capacities range from one ton up to any size required. Cooling and heating surface, and filter area are liberally proportioned and blowers are of moderate speed, all to insure the highest efficiency and quiet, satisfactory performance. Bulletin "E".



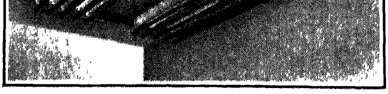


RECOY CONTINUOUS FIN BLAST COILS for cooling or heating are constructed of copper tubing with aluminum fins, or all steel hot dip galvanized after fabrication and are suitable for use with any cooling or heating medium. Bulletin "F".



RECOY EVAPORATIVE CONDENSERS are cooling towers and condensers combined into one efficient unit for use indoors or out. They reduce the water consumption 95 per cent and are used with no water at all in cold weather. Made in sizes from one to one hundred tons. Ceiling Type 2 to 12½ tons. Bulletin "G".

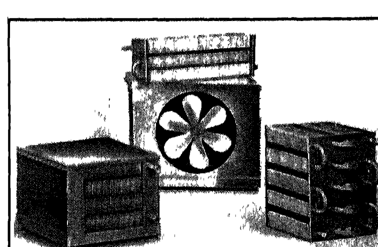
RECOY DOWN DRAFT FIN COILS are used for all refrigerating purposes for temperatures above 34 F. Coils defrost at every cutout period and are equipped with aluminum troughs that catch the water.



No coil bunkers or pans required. Air circulation unimpeded and very rapid and humidities of 85 to 90 per cent are easily maintained.

Coils are made of aluminum, copper, or steel hot dipped galvanized after fabrication. They require only about 12 in. of head room and may be hung up to the ceiling between meat rails, etc., thus conserving space and reducing the cost of cold storage rooms. Bulletin "F".

RECOY REFRIGERATING FINNED COILS and unit coolers are made up on continuous copper, or aluminum tubing (no joints so no leaks), with aluminum fins, or all steel, hot dipped galvanized after fabrication. Fins are corrugated to hold moisture in the air stream during off cycle to re-evaporate and maintain high humidities. They are applicable to both large and small work. Bulletin "F".



Surface Combustion Corporation TOLEDO OHIO

Largest and Oldest Manufacturers of Exclusively Gas-Fired Equipment

Kathabar Air Conditioning Systems | **Janitrol Winter Air Conditioners**
Kathabar Humidity Control Units | **Janitrol Boilers, Gravity Heaters**
Janitrol Unit Heaters

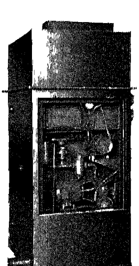
KATHABAR

- **FOR COMFORT**—in hotels, restaurants, public buildings, office buildings, factories, stores, theatres and recreation places.
- **FOR HEALTHFUL COMFORT**—in hospitals and institutions.
- **FOR CONTROLLED DRYING**—in factories where materials and their by-products must be dried under controlled conditions.
- **FOR SPECIAL ATMOSPHERES**—in industries where air conditions affect materials, processes and products.
- **FOR DRY BLAST**—in cupola and blast furnaces where conditioned air is required in combustion.

KATHABAR OPERATING PRINCIPLE

Kathabar solves difficult problems in air conditioning by a radically different operating principle. Kathabar applies the unique qualities of a liquid chemical called Kathene, the vapor pressure of which is readily changed and accurately controlled. Air passing through the Contactor Cell encountering the Kathene automatically gives up or takes on moisture until its vapor pressure approaches that of the Kathene. Thus any desired humidity is created in the treated air with absolute precision regardless of outside weather. No by-passing is necessary.

Treated air leaving the Contactor Cell at precisely controlled humidity passes over the conventional finned coils, utilizing cool well water, chilled water, or direct refrigerant as the cooling medium, or to conventional heating coils. This permits close individual control of temperature as well as humidity.



Kathabar operates in a continuous cycle, for the Kathene liquid flows from the Contactor Cell to the regenerator sump. Here moisture lost in the contactor is replaced by water automatically added, or excess moisture is expelled in a very simple process involving the heating of only a portion of the solution. Low cost exhaust steam may be utilized for regeneration. The proper vapor pressure being restored, the Kathene continues through the operating cycle.

Kathene also cleans and deodorizes the air treated. It is non-inflammable, non-explosive, odorless, and inherently harmless.

Kathabar is compact, readily accommodated in practically any space, and is so adaptable that it may easily be included in system specifications for comfort or industrial process installations. Its low operating costs, and its extreme simplicity both in control and operation recommend it for trouble-free satisfaction. Complete engineering data sent on request.

STANDARD KATHABAR EQUIPMENT

Contactor cells are available in two sizes and two types. The No. 3 and No. 4 cells will handle up to 3,000 and 4,000 cfm respectively. The type No. 1 cells have one section of contact surface and are for normal use where water at normal temperatures is available for KATHENE cooling. Where higher temperature city or C. T. water must be used, the type No. 2 cells are available. These cells have two sections of contact surface and therefore operate at higher KATHENE temperature than the type No. 1 cells for the same moisture removal. The size and type numbers are combined to give the cell numbers 31, 41, 32 and 42.

KATHENE coolers and heaters are available in a variety of shell sizes and tube lengths for the use of water at various temperatures and steam at various pressures. These cells may be banked together to handle air quantities as large as 100,000 cfm.

Regenerators are made in standard sizes from 1 to 16. The size number designates the number of regenerator contact sections in the unit.

REGENERATORS

Each section of regenerator surface will remove approximately 43 pounds of water per hour on 5 pound steam, 58 pounds per hour on 12 pound steam and 65 pounds per hour on 25 pound steam. Each section of regenerator requires about 340 cfm of fresh air.

Table D1. Kathabar Contractor Cell Data

Cell Size	Max. CFM	Overall Dimensions—Inches		
		Width	Length	Height
No. 31	3,000	27	48	49
No. 41	4,000	27	48	61
No. 32	3,000	27	66	54
No. 42	4,000	27	66	66

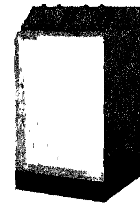


Table D6. Type No. 2 Cell Ratings Dehumidifying

Max. Water Temp...	90	80	70	60	55
Inlet Air—Gr./No...	Outlet Air—Gr./No.				
130	51	38	28	21	18.5
110	50	37	27	20	18.0
90	49	36	26.5	20	17.5
70	48	36	26.5	19.5	17.5
50	46	35	26.0	19.5	17.5

Table D7. Outlet D. B. Type No. 2 Cells for Inlet D. B. of 85° F.

Max. Water Temp...	90	80	70	60	55
Inlet Air—Gr./No...	Outlet Dry-Bulb Temp. °F.				
130	106	99	92	85	82
110	104	97	90	83	79
90	101	95	88	81	77
70	99	93	86	78	74
50	97	91	84	76	73

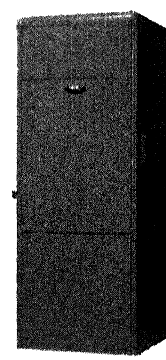
Janitrol Gas-Fired Winter Air Conditioners, Gravity Heaters, Boilers, Unit Heaters

For any type home or individual apartment heating, this compact, gas-fired unit, completely assembled in the most popular sizes, will satisfy most exacting clients. New features include Multi-Thermex method of heat transfer, Amplifire burner, patented control system. Send for Janitrol FAC Series Specification Sheets.

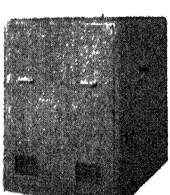
Engineering Data—FAC-14 Series Conditioners

Model No.	Rating Btu—Hr.		CFM Delivery for °F. Temp. Rise			Gas Cap. Cu. Ft. Hr. with Btu Content of		
	Input	*Output at Register	70°	80°	90°	550	800	1000
FAC 60-14	60,000	43,200	600	525	465	109	75	60
FAC 90-14	90,000	64,800	900	790	700	164	113	90
FAC120-14	120,000	86,400	1200	1050	935	218	150	120
FAC150-14	150,000	108,000	1500	1310	1170	273	188	150
FAC180-14	180,000	129,600	1800	1575	1400	328	225	180

*"Hourly Btu output at register" capacities allow for 10 per cent heat loss from warm air ducts between bonnet and register—normal loss for average installations. Longer than average ducts, type of duct covering, duct exposure, whether or not basement is heated, etc., should be considered and conditioner sized accordingly.

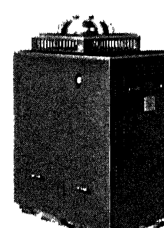


GAS-FIRED GRAVITY FURNACE

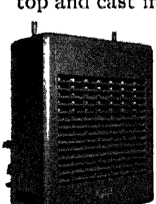


Ideal for projects, where heating costs must be kept in line with the $4000 home, but where you want satisfactory long-life equipment. Five sizes, from 66,000 to 154,000 Btu input (A.G.A.). Cast iron Multi-Thermex heat exchanger, Amplifire Burner. Janitrol GAC Series Specification Sheets give complete information.

GAS BOILER



Janitrol Gas Boilers are built in two types, Type S for steam or vapor, Type W for Hot Water. Capacities range from 40,800 to 540,000 Btu available output per hour. Cast iron flue top and cast iron sections exemplify rugged construction. Controls comply with A.S.M.E. Boiler Construction Code and A.G.A Approval Requirements. A.G.A. ratings, Bonded Load Ratings, complete data given in Janitrol Gas Boiler Specifications.



GAS-FIRED UNIT HEATERS FOR INDUSTRIAL AND COMMERCIAL USE

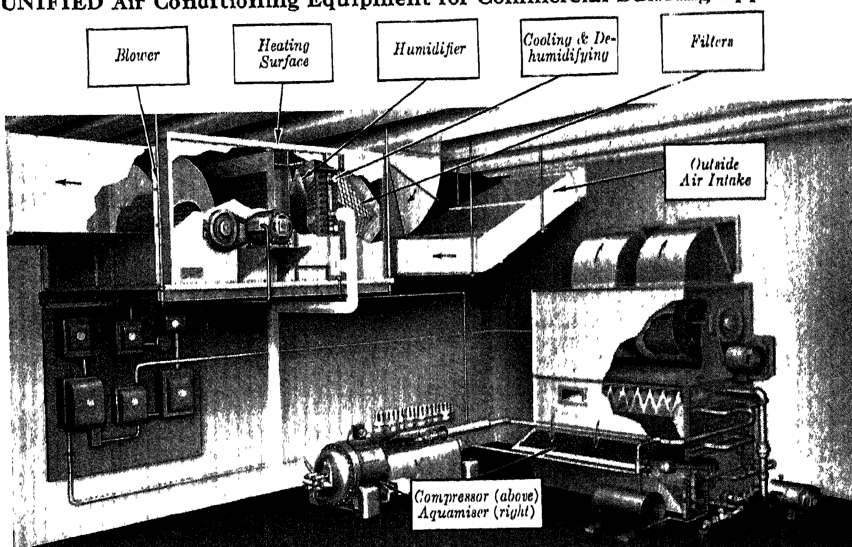
Sizes and types for every space heating need, from small stores to large industrial plants. Capacities range from 50,000 to 1,250,000 Btu hourly input. Can be operated in multiple. Send for Specification Sheets on blower, propeller and floor types.

Westinghouse Electric & Manufacturing Co.

653 Page Blvd., **Springfield, Mass.**

Sales, engineering and service available through Authorized Engineering Contractors in all principal cities

UNIFIED Air Conditioning Equipment for Commercial Building Applications



APPLICATION—Westinghouse equipment provides air conditioning for every building application. Every unit is engineered by Westinghouse and applied in carefully engineered installations.

COMPRESSORS AND CONDENSING UNITS—Hermetically-sealed to conserve power and efficiency. Dust, dirt and trouble sealed out. No seal leaks. All operating parts accessible. Lightweight and compact.

WATER CHILLING UNITS—Available in five sizes from 5 to 110 tons of refrigeration. Highly efficient vertical, semi-flooded type, designed for air conditioning with chilled water.

AIR CONDITIONING UNITS—Incorporate blower, heat transfer surfaces, humidifiers and filters. Horizontal and vertical types.

AQUAMISERS (Evaporative Condensers)—Reduce water consumption 90 per cent to 95 per cent. Sizes to match all compressors in which water consumption is a factor.

Hermetically-Sealed Compressors and Condensing Units

Type	Hp	Capacity Btu per Hour		Dimensions Inches‡			‡Approx. Net Weight Lb
		*Nominal Rating	†Maximum	Length	Width	Height	
CLD- 45	1	12000	15700	23½	18	16⅛	235
CLD- 90	2½	26900	41000	21	20½	36½	440
CLD- 135	3¾	41000	60800	21	20½	36½	460
CLD- 205	5	69600	94400	32	22	36	572
CLD- 275	7½	90000	124000	32¼	21¾	32¼	590
CLD- 415	10	120000	163000	34½	22¼	35¼	630
CLS- 550	15	181000	255000	64	19½	40½	1720
CLS- 640	20	222000	302000	80¾	19½	40½	2080
CLS- 850	25	294000	393000	80¾	19½	40½	2080
CLS-1320	40	455000	615000	92⅞	26¼	49¼	3760
CLS-1980	60	653000	896000	92⅞	26¼	49¼	3775
CLS-2550	75	915000	1155000	97	34	64	7300
CLS-3400	100	1212000	1529000	95¼	34	70	8100

*Rating at 37 lb suction, 65 F suction gas temperature, 75 F entering water, 95 F leaving water.
†Rating at 50 F evaporating temperature, 25 F superheat and 100 lb per square inch condensing pressure.
‡Dimensions and weights are for complete water-cooled condensing units.

(*See also Page 876*)

Air Conditioning Units

Type	Nominal Capacity Air Delivery Cfm	Dimensions—Inches Length	Width	Height	Approx. Net Weight Lb
AF- 16	480- 1120	$58\frac{5}{8}$	29	21	300
AF- 27	810- 1890	50	46	19	400
AF- 37	1110- 2590	50	58	21	525
AH- 27	1500- 2250	$55\frac{1}{2}$	$66\frac{5}{8}$	$22\frac{1}{2}$	350
AV- 27	1500- 2250	$38\frac{1}{16}$	$51\frac{5}{8}$	$45\frac{3}{16}$	360
AH- 55	2580- 3870	$72\frac{13}{16}$	$79\frac{1}{2}$	$27\frac{1}{4}$	612
AV- 55	2580- 3870	$48\frac{1}{16}$	$79\frac{1}{2}$	$62\frac{1}{4}$	615
AH- 83	4080- 6120	$81\frac{9}{16}$	$88\frac{15}{16}$	$33\frac{1}{4}$	794
AV- 83	4080- 6120	$56\frac{13}{16}$	$88\frac{15}{16}$	$74\frac{1}{4}$	826
AH-103	4930- 7394	$75\frac{3}{4}$	$102\frac{9}{16}$	$33\frac{1}{4}$	915
AV-103	4930- 7394	51	$102\frac{9}{16}$	$74\frac{1}{4}$	951
AH-124	5800- 8700	$82\frac{3}{4}$	$111\frac{3}{4}$	$35\frac{1}{4}$	1127
AV-124	5800- 8700	58	$111\frac{3}{4}$	$78\frac{1}{4}$	1177
AH-154	7874-11812	$93\frac{5}{8}$	$112\frac{5}{16}$	$42\frac{1}{2}$	1450
AV-154	7874-11812	$68\frac{3}{4}$	$112\frac{5}{16}$	$92\frac{3}{4}$	1550

Aquamisers (Evaporative Condensers)

Type	*Capacity Btu per Hour	Dimensions—Inches Length	Width	Height	Approx. Net Weight Lb
EV - 205	99200	$68\frac{1}{8}$	$31\frac{1}{4}$	$62\frac{1}{2}$	719
EV - 275	117800	$68\frac{1}{8}$	$31\frac{1}{4}$	$62\frac{1}{2}$	832
EV - 415	187000	$89\frac{1}{8}$	$38\frac{1}{4}$	$69\frac{1}{2}$	1266
EV - 550	247000	$89\frac{1}{8}$	$38\frac{1}{4}$	$69\frac{1}{2}$	1376
EV - 640	312600	$115\frac{1}{8}$	$42\frac{7}{8}$	$74\frac{1}{2}$	1863
EV - 850	399600	$115\frac{1}{8}$	$43\frac{1}{4}$	$74\frac{1}{2}$	2065
EV -1320	572700	$117\frac{3}{8}$	$55\frac{1}{4}$	$88\frac{1}{2}$	2931
EV -1700	799200	$117\frac{3}{8}$	$83\frac{9}{16}$	104	3902

*Net refrigerant capacity at 75 F WB entering air and 110 F condensing temperature.

HOW TO SELECT—Fit equipment to meet the total Btu load.

WHERE TO BUY—Consult classified telephone directory or nearest Westinghouse district office for name of Authorized Contractor.

SELF CONTAINED SYSTEMS

Mobilaire

Compact, self-contained units for individual room cooling in homes, offices, hotels, apartments. Attractive cabinets of modern design and finish. Details available on request.

Unitaire

Compact, self-contained units for retail stores, restaurants, and other businesses, also for home use. Available in either "within-the-space" or "central plant" types. Quick economical installation either singly or in combination. Nine sizes.

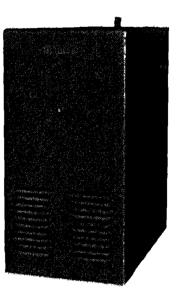
Unitaire Specifications

Type	*Capacity Btu per Hour	Dimensions—Inches Depth	Width	Height	Approx. Net Weight Lb
CU- 45	12000	24	38	26	450
SU- 20	24000	24	36	45	750
SU- 30	36000	24	40	57	900
SU- 50	64500	$23\frac{3}{8}$	$46\frac{1}{2}$	$92\frac{7}{8}$	1455
LU-275	85000	†34	54	$69\frac{3}{4}$	1970
LU-415	115000	†34	$70\frac{1}{2}$	72	2300
LU-550	168100	†34	$82\frac{1}{2}$	$66\frac{1}{2}$	3350
LU-640	204000	*$34\frac{5}{16}$	100	$73\frac{1}{4}$	4000
LU-850	256000	‡$34\frac{5}{16}$	100	$73\frac{1}{4}$	4200

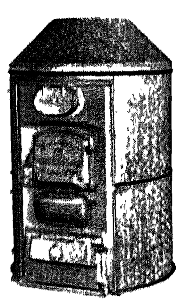
*Net capacity with normal air flow, entering 80 deg DB, 67 deg WB and condenser water inlet 75 F, outlet 95 F.
†Add $14\frac{1}{2}$ in. for overall dimensions with filters.
*Add $21\frac{1}{4}$ in. for overall dimensions with filters.
‡Add $25\frac{1}{2}$ in. for overall dimensions with filters.

HOME HEATING AND AIR CONDITIONING

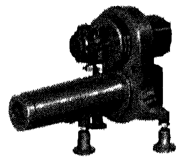
A complete line of attractive, efficient heating and air conditioning equipment for oil, coal or gas, in a full range of capacities for all residential requirements.



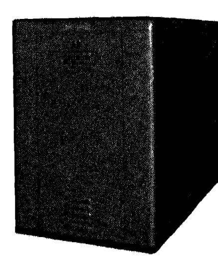
Boiler-burner units for steam or hot water systems, oil, coal, or gas-fired.



Steel and cast-iron gravity warm air furnaces available in 20 models and sizes, for coal, oil or gas firing.



Spiralaire conversion oil burner, capacities from 1 to 12 gph.



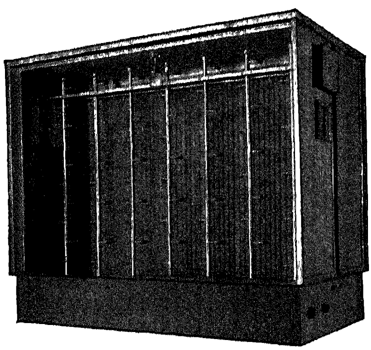
Oil, gas and coal-fired winter air conditioning unit, capacities from 80,000 to 285,000 Btu at register.

United States Air Conditioning Corporation

A Complete Line of Air Conditioning Equipment

2101 Kennedy St., N.E.
Minneapolis, Minn.
Branch Offices or Agents in Principal Cities

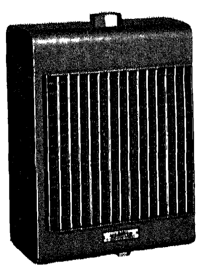
U. S. AIRCO Air Washers



Single and double stage air washers, all sizes, from 2500 cfm to 100,000 cfm.

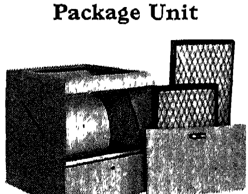
U. S. AIRCO Unit Heaters

Series 39 Unit Heater with U. S. AIRCO patented Deflecto-Grille, horizontal and vertical blades both adjustable for perfect control of air volume and air distribution.



Also Standard Model Unit Heaters with adjustable horizontal louvres.

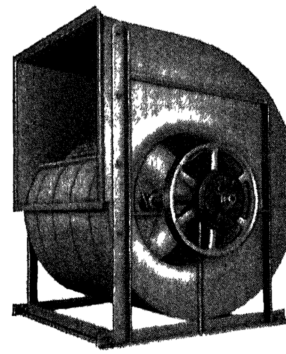
U. S. AIRCO—Blower—Filter Package Unit



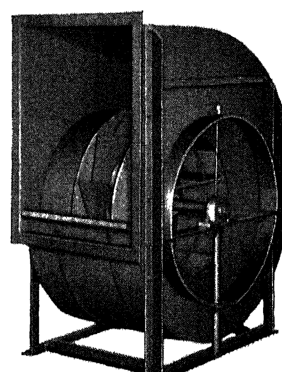
Complete units with blower, drive, motors, filters, furnacestat. Also Blower Assemblies, wheels and scroll housings.

U. S. AIRCO Blowers

Single inlet single width and double inlet double width Blowers for both supply and exhaust. Sizes from 300 cfm to 100,000 cfm.



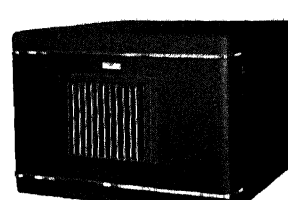
Type A Blower, with backwardly curved blade impeller. Both single and double inlet. Sizes from 1,000 to 70,000 cfm.



Also light duty Blowers and Blower-Filter Units for furnaces and self-contained air conditioners.
Also Propeller (Exhaust) Fans.

U. S. AIRCO Unit Coolers

Unit Coolers for cold water or direct expansion. Range of sizes.



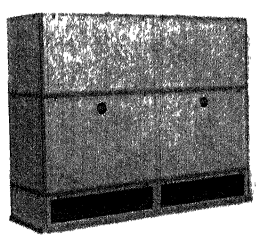
Send for catalog showing complete line of U. S. AIRCO Equipment.

York Ice Machinery Corporation

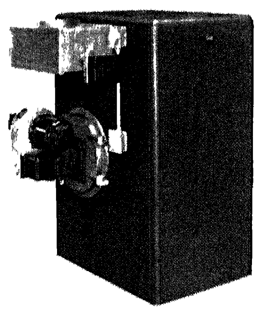
York, Pennsylvania

Factory Branches and Distributor Engineering and Sales Offices throughout the World.

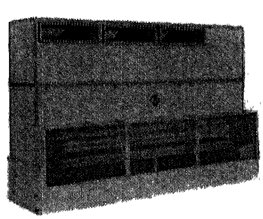
Air Conditioning and Refrigeration for maintaining proper atmospheric conditions for human comfort and industrial processes. Installations of unit and central systems in a complete range of capacities and types for every design requirement.



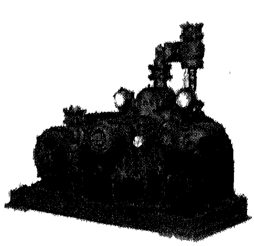
York Sectional Air Conditioner



Yorkaire Heating Unit



York Sectional Economizer



York V-W Condensing Unit

Air Conditioning Units: A complete line of finned coil, dry coil, wetted surface and spray type sectional air conditioners for horizontal or vertical applications, designed to facilitate installation and the distribution of air. Standard units can be equipped with by-pass feature and arranged for cooling and dehumidifying, heating and humidifying, for year-round comfort.

Yorkaire Heat and Winter Air Conditioning—A complete line of York equipment is available for residential or small commercial heating and winter air conditioning installations. Direct fired furnaces, and boilers for steam or hot water can be furnished for burning oil, gas, or coal. Stokers and conversion oil burners complete the catalog. Related apparatus for use with YORKAIRE HEAT units provides complete equipment for year-round air conditioning systems for homes and small business establishments.

Dehumidifiers—For central station systems where a large volume of air is to be handled and where control of humidity is an essential requirement, the York dehumidifier is especially applicable. Construction features insure a minimum space demand and maximum performance conditions. Standard washers are available in a full range of capacities for human comfort or industrial installation. Air washers can be furnished also for use as indoor condensing water cooling towers when specified.

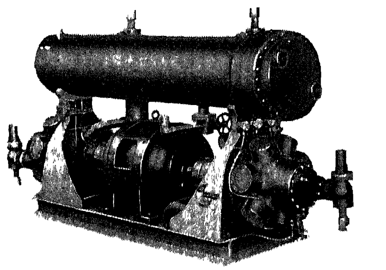
The York Economizer—A combined forced-draft cooling tower and refrigerant condenser, is available for installations where prohibitive water costs or inadequate drainage facilities preclude the use of a water cooled condenser. Standard factory constructed and built-up units may be used singly or in multiple for applications of any specified capacity. Economizers for use with Freon as the refrigerant are furnished, as standard, with a liquid sub-cooling coil.

Condensing and Water Cooling Systems—Standard systems are available for every application requirement up to 1,000 hp capacity using a single compressor. Self-contained units up to 300 hp feature the YORK line. These units are furnished with water cooled condensers or without condensers for economizer applications.

Automatic or manual capacity reduction by-pass valves can be provided for economical operation at reduced load.

All materials and manufacturing methods employed in the construction of the York Freon Condensing and Water Cooling Systems conform to the high standards and efficient operating characteristics of all YORK products and carry performance guarantees based on *Refrigeration Manufacturers' Association* and *Air Conditioning Manufacturers' Association* ratings.

AIRTEMP
DIVISION OF
CHRYSLER CORPORATION • DAYTON, OHIO



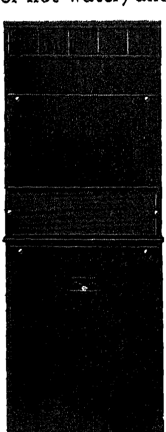
RADIAL COMPRESSOR

Sizes range from 10 Hp. to 75 Hp. capacity, for direct expansion with Freon or chilled water. For use with city water, cooling towers or evaporative condensers.

Airtemp Radial Compressors are engineered for air conditioning duty. Direct connected. Force-feed lubrication. Automatic starting unloader and automatic capacity-reduction unloader. Removable cylinder liners. Vital parts are "Superfinished" to reduce wear to a minimum. Exclusive cylinder unloader gives high operating efficiency. Oil separator. Practically no vibration. Shipped ready to run —easy to install—no foundation necessary.

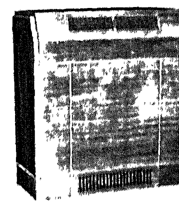
"All-In-One" AIR CONDITIONER

These compact units cool, dehumidify, clean, and circulate the air—free discharge or duct distribution. Heating coil (steam or hot water) and humidifier can be added.

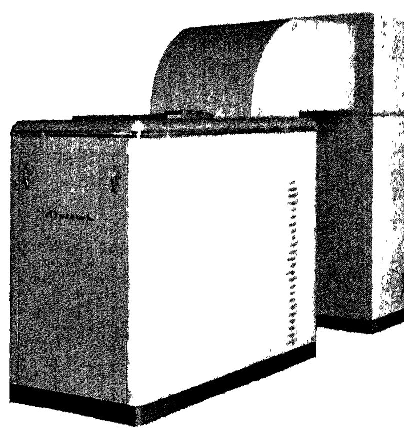


Sealed radial compressor with removable cylinder liners, force-feed lubrication, and automatic starting unloader. Vital parts "Superfinished" to reduce wear. Compressor suspended from single rubber mounting. Water-cooled condenser for use with city water or cooling tower. Shipped tested and with all controls—ready to install. Uses Freon, the safe refrigerant. "Bonderized" cabinet. Approved by Underwriters' Laboratories, Inc.

COOL-BREEZE SUMMER AIR CONDITIONER



Larger models cool, dehumidify, circulate, and filter the air introduce fresh air, take out stale, room air. Shut out street noises. One window model (⅓ Hp.) and two floor models available (½ and ¾ Hp.).

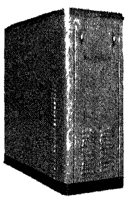


YEAR-'ROUND AIR CONDITIONER

A combination heating and cooling unit for homes consisting of an Airtemp Forced-air Furnace and a self-contained refrigerating mechanism in 3 Hp. capacity. It gives complete year-'round air conditioning through the same system of ducts in homes of average construction and size (6 to 8 rooms).

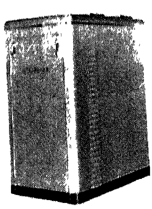
The heating unit can be any Airtemp Winter Air Conditioner shown on the opposite page, except the smallest sizes. The cooling unit is identical with the All-In-One Conditioner pictured at the left, except that it uses the furnace blower and the evaporator is in a vertical position instead of horizontal.

AIRTEMP
DIVISION OF
CHRYSLER CORPORATION • DAYTON, OHIO



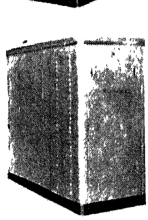
OIL-FIRED WINTER AIR CONDITIONER

Heats, humidifies, filters and circulates the air. Five models from 70,000 to 160,000 B.t.u. output. "Bonderized" and insulated jacket. Buff with red top, stainless steel trim. Chrome steel combustion chamber, seam-welded firebox of copper-bearing steel, large, slow-speed, rubber-mounted fan. Airtemp conventional or Twin Airflow Oil Burners.



GAS-FIRED WINTER AIR CONDITIONER

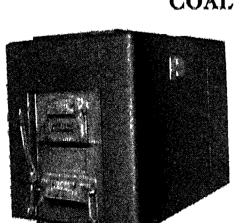
Heats, humidifies, filters and circulates the air. Steel firebox models from 70,000 to 160,000 B.t.u. output. Cast iron firebox models from 40,000 to 160,000 B.t.u. output. "Bonderized" and insulated jacket in two tones with stainless steel trim. Airtemp exclusive "Silent Flame" Gas Burner. Efficient, quiet while it starts, operates, and stops. Approved by A.G.A. Laboratories.

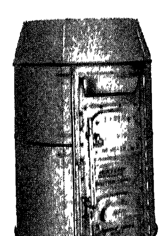


BOILERS—Oil-Fired—Gas-Fired

Steam or hot water. Six oil-fired models from 400 to 1,700 E.D.R. (steam). Four gas-fired models from 400 to 1,050 E.D.R. (steam). "Bonderized" and insulated jacket in two tones of green with stainless steel trim. Fire chamber surrounded with water on all sides and bottom. Internal heater for domestic hot water. Burner and all controls enclosed. Oil-fired models use conventional or Twin Airflow Oil Burner. Gas-fired models use Airtemp exclusive "Silent Flame" Gas Burner. Efficient and quiet.

COAL-FIRED FURNACES—Forced-Air—Gravity

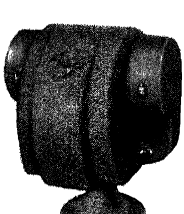




Forced-air models in cast iron and steel. Sizes from 226 square inch grate area to 346 square inch grate area in steel—and from 258 square inch grate area to 384 square inch grate area in cast iron. Heavy "Bonderized" jacket. Large, slow-speed blower rests in rubber for quietness. Oversize blower motor, also mounted in rubber and has automatic overload and low voltage protection.

Gravity models in cast iron and steel. Sizes from 226 square inch grate area to 452 square inch grate area in steel—and from 204 square inch grate area to 452 square inch grat area in cast iron.

OIL BURNERS



Airtemp Oil Burners are approved by Underwriters' Laboratories, Inc., and bear the seal of the Official Inspection Agency of the Oil Burner Industry as evidencing compliance with commercial Standard CS75-39, as issued by the National Bureau of Standards of the U. S. Dept. of Commerce

Model A-8: A conventional oil burner. .75 to 1.35 gallons No. 3 Furnace Oil per hour. Quiet, efficient.

Model B-9: The exclusive Airtemp Twin Airflow Oil Burner. 1.35 to 3.0 gallons No. 3 Furnace Oil per hour. Air for combustion is furnished by two fans instead of one, and adjusted at the outlet, not the inlet of the fans. Quiet, efficient.

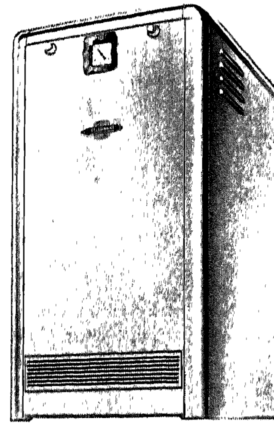
Model B and C10: A conventional pressure atomizing oil burner. 1.35 to 4.5 gallons No. 3 Furnace Oil per hour. Finished in gray and blue with stainless steel trim.

DELCO AUTOMATIC HEAT

BOILERS

Delco Automatic Boilers coordinate the Delco Oil Burner with a boiler of special design and construction for application to hot water, steam, or vapor-vacuum heating systems. Most oil-fired models incorporate the famous Delco Oil Burner with the Rotopower Unit. Boiler sections are dotted with heat-absorbing fins. When sections are fitted together, fins form a series of passes, exposing a maximum of water-backed surface to the heated gases.

Three models, the DSS, the DB-3 and DB-4 incorporate the exclusive Quik-Action Heat Transmitter which provides quick, radiant heating and renders the previous slow heating fire clay refractory lining of the combustion chamber unnecessary. Provision is made in all models for incorporating a built-in domestic water heater. Six oil-fired automatic boilers, with capacities ranging from 300 to 1,335 sq ft of steam, EDR, are available.

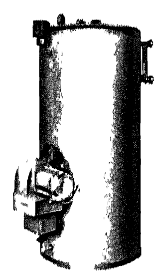


The DB-3 Boiler

DSS-DSW BOILER

The latest addition to the Delco line of automatic boilers is Model DS, a new small steel boiler for either steam or hot water with a capacity of 300 sq ft of steam EDR. The Delco Automatic Water Heater, Model DS, resembling the steel boiler in construction and operation, is also available.

Model DS employs the Quik-Action Heat Transmitter which heats 9 times faster than refractory combustion chambers using an open type flame; the simplified Rotopower Unit that contains all moving parts within a single, easily removed unit on an integrated shaft; the Thin Mix Fuel Control which meters fuel oil economically and prevents waste; and the Heat Trap, a special baffle that conserves heat ordinarily lost. Water chambers of the DS Boiler are of boiler-plate steel tested to 300 lb pressure. They are insulated with a thick overcoat of rock-wool.

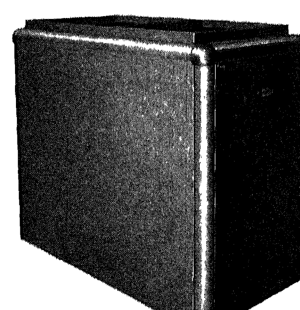


The DSS Boiler

CONDITIONAIRS

The Delco Conditionair is a compact, completely automatic unit, oil or gas-fired, which provides true winter air conditioning by circulating cleaned, humidified and properly heated air. Model DAOO, DAO and DA1 Delco Oil Conditionairs incorporate the new, exclusive Quik-Action Heat Transmitter. Air flow resistance is reduced to a minimum by tear drop design. Heat is transferred to the flowing air from a large heating surface dotted with heat projectors, and moisture is then added by a pan, cascade or spray type humidifier.

In Delco Gas Conditionairs there is a sufficient range in sizes to permit selection of the proper unit for applications ranging from 55,000 Btu heat loss to 120,000 Btu heat loss. Delco Conditionairs, oil-fired, range in size from 85,000 Btu heat loss to 200,000 Btu heat loss.



The Delco DAO Conditionair

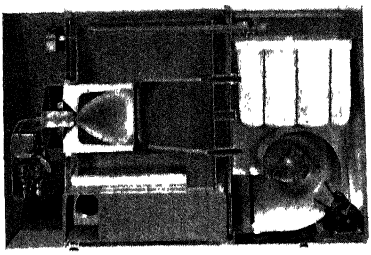
Gar Wood Industries, Inc.

AIR CONDITIONING DIVISION

Gar Wood

7924 Riopelle St., **Detroit, Mich.**

Licensed Distributors in All Principal Cities

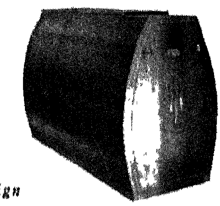


Cross-Sectional View of Tempered-Aire Unit

TEMPERED-AIRE UNITS

Made in six (6) capacities, with filters, blower, burner, humidifier, oil furnace and economizer engineered into one compact, coordinated system. Clean, humidified, filtered, warmed air is delivered from the unit to the duct system and then uniformly distributed to all parts of the building. Air is cleaned by washable cloth filters and circulated by a multiple-blade ball bearing blower, rubber mounted. The heating unit is a counterflow, down-draft furnace and economizer unit, equipped with an integral fire bowl and pressure atomizing oil burner. The humidifier is a flash type, steam tube, located within the firebox, assuring prompt humidification.

The 1941 line of Tempered-Aire units covers a wide range (Models 001, 101, 201, 301, 401, 501), and has many improvements. The perfected Model "O" oil burner starts, operates and cuts off smoothly even with widely varying draft conditions. Combustion is clean, quiet and efficient, due to the improved manner in which the air and oil are mixed and burned in the horizontal, cylindrical fire bowl to produce the efficient, sunburst-shaped flame.



Improved Functional Design

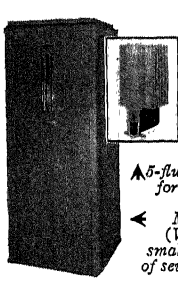
Tempered-Aire Ratings and Dimensions	No. 001	No. 101	No. 201	No. 301	No. 401	No. 501
Btu 1 Hour at Bonnet	85,000	100,000	135,000	200,000	300,000	400,000
Btu 1 Hour at Grilles	68,000	80,000	110,000	160,000	240,000	320,000
*Air Delivery CFM	850	1,000	1,350	2,000	3,000	4,000
Oil Nozzle Size G.P.H.	.85	1.00	1.35	2.00	3.00	4.00
Heating Surface, Sq. Ft.	40	54½	64	86	124	164
Filter Area, Sq. Ft.	30	30	48	52	78	78
Motor HP—Burner	1/6	1/6	1/6	1/6	1/6	1/6
Motor HP—Blower	1/6	1/8	¼	[illegible]	¼	1
Overall Length, Inches	71	77	83	89	116	131
Overall Width, Inches	29¼	32	32	36⅝	45	45
Overall Height, Inches	48	54	54	57	65	65
Dim.—Supply Opening	28¾"x14¾"	35"x16¾"	36½"x16¾"	36¾"x16¾"	50"x21¼"	65"x21¼"
Dim.—Return Opening	28½"x14¾"	28½"x16¾"	32½"x16¾"	36"x16¾"	47½"x21¼"	47½"x21¼"
Blower Wheel, Dim.	10"x10"	12"x12"	14"x14"	16"x16"	20"x20"	20"x20"
Shipping Weight, lbs.	760	900	1,000	1,200	1,500	1,800
†Stack Connection, dia.	7"	7"	8"	9"	2-10"	2-10"
Recommended Chimney Size	8"x12"x30'	8"x12"x30'	8"x12"x30'	12"x12"x35'	12"x16"x40'	12"x16"x40'

*Ratings based on 160 deg bonnet temperature and 68 deg return temperature. Greater air deliveries with correspondingly lower bonnet temperatures may be had by the use of larger blower motors.

†On Models 401 and 501 connect both stack outlets to chimney.

GAS-FIRED AIR CONDITIONING UNITS

Seven units: 60V, 84H&V, 120H&V, 168H, 240H, 360H, 480H (V, vertical type; H, horizontal). The model numbers designate the heat inputs in thousands of BTU's per hour. Three standardized basic heating sections (5-, 7-, and 10-fluted) used singly and in multiple produce the seven sizes and heating capacities. The horizontal types—approximately 4½ feet in height—are ideal for basements with unusually low ceilings. The vertical types (while also designed for basements) occupy small space and are suitable for installation in heater-utility rooms—located on the first floor—where space is definitely limited.

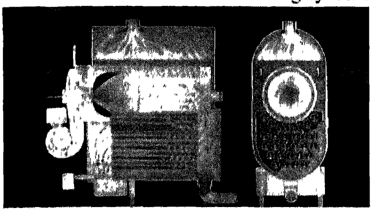


▲ *5-fluted heating section for MODEL 60V*

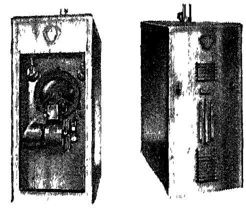
◀ *MODEL 60V (Vertical Type) smallest in 1941 line of seven gas-fired units*

BOILER-BURNER UNITS

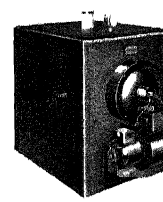
For steam and hot water heating systems. Eight sizes: Series "B" 5 sizes; Series "R" 3 sizes.



Cross-sectional view of Models 135, 180 and 270



Models 135, 180 and 270

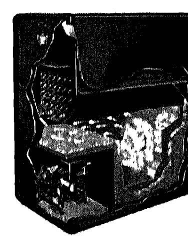


Models 120 and 165

Series "B"	B-120-W	B-135-S or W	B-165-W	B-180-S or W	B-270-S or W
Boiler Nozzle Output B.T.U./Hr.	120,000	135,000	165,000	180,000	270,000
Net Load Rating B.T.U./Hr.	80,000	90,000	110,000	120,000	180,000
Boiler Nozzle Output Sq. Ft. 240 B.T.U./Hr. Steam Radiation	—	562	—	750	1125
Boiler Nozzle Output, Sq. Ft. 150 B.T.U./Hr. Hot Water Radiation	800	900	1100	1200	1800
Net Load Capacity 240 B.T.U./Hr. Steam Radiation	—	375	—	500	50
Net Load Capacity 200 B.T.U./Hr. Hot Water Radiation	400	450	550	600	900
Net Load Capacity 150 B.T.U./Hr Hot Water Radiation	533	600	733	800	1200
Heating Surface, Sq. Ft.	30	34	44	46	64
Oil Jet Size, G.P.H.	1.20	1.35	1.65	1.80	2.70
Burner Type	"O"	"O"	"O"	"O"	"O"
Size Steam Nozzle	1-3"	1-3"	1-3"	1-3"	1-4"
Size Return Connection	1-3"	1-3"	1-3"	1-3"	1-4"
Stack Size	7"	8"	8"	8"	9"
Recommended Chimney Size	8"x12"x30'	8"x12"x30'	8"x12"x35'	12"x12"x35'	12"x16"x40'
Water Capacity, Gallons—Steam	—	40	—	46	64
Water	14	51	20	64	88
Internal Tankless Water Heater (Dual Coil)	—	—	—	200*	200*
Internal Tank Type Water Heater (Single Coil)	—	30*	—	30*	30*
Shipping Weight—Pounds	885	1000	1100	1300	1585

*Gallons per Hour 80° Temperature Rise at 180° Boiler Temperature.

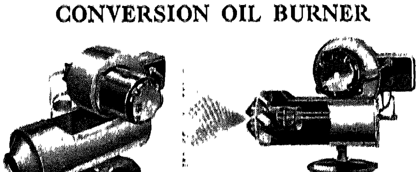
Series "R"	R1000	R1400	R1800
Burner Model	D	D	D
Oil Jet Size—Gal. per Hr.	3.50	5.00	6.50
Max. Net Steam Load, Sq. Ft.	1000	1400	1800
Max. Net Hot Water Load, Sq. Ft.	1600	2240	2880
Max. Gross Steam Load, Sq. Ft.	1500	2100	2700
Max. Gross Hot Water Load, Sq. Ft.	2400	3360	4320
Heating Surface, Sq. Ft.	84	118	154
Size Steam Nozzle	4"	2-4"	2-4"
Size Return Connections	$2\text{-}2\frac{1}{2}$"	$2\text{-}2\frac{1}{2}$"	2-3"
A—Overall Height, Jacket	$68\frac{5}{8}$	$68\frac{5}{8}$	$68\frac{5}{8}$
B—Overall Width, Jacket	$37\frac{13}{16}$	$37\frac{13}{16}$	$37\frac{13}{16}$
C—Overall Length, Jacket	$59\frac{3}{4}$	66	80
D—Height Steam Nozzle	$68\frac{3}{8}$	$68\frac{3}{8}$	$68\frac{3}{8}$
E—Height Return Connections	$29\frac{7}{8}$	$29\frac{7}{8}$	$29\frac{7}{8}$
F—Height External Heater Connection	$43\frac{5}{8}$	$43\frac{5}{8}$	$43\frac{5}{8}$
G—Height Mean Water Line	$47\frac{3}{8}$	$49\frac{9}{16}$	$49\frac{9}{16}$
H—Height Smoke Pipe Adapter	40	40	40
I—Hor. Distance Between Returns	7	7	7
J—Hor. Distance Between Steam Nozzles		18	18
Stack Size	12"	12"	12"
Recommended Chimney Size	12"x16"x40'	12"x16"x45'	16"x16"x45'
Water Capacity, Gals.	52	69	91
Shipping Weight, Lbs.	2200	2500	2950



SERIES "R"

The series "R" Boiler-Burner Units are designed for larger installations where their separate boiler shells and fireboxes facilitate rigging and erection.

CONVERSION OIL BURNER

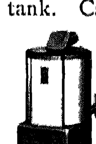


Model "OC"

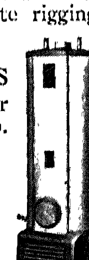
For installation in existing heating plants.

DOMESTIC WATER HEATERS

Model S50 - coil type heater for separate storage tank. Cap. 50 gal. per hour 100 F rise. Model S40 has storage tank integral. Cap. 40 gal. per hour 100 F rise. Vertical rotary burner.



Model S50



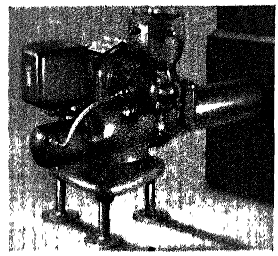
Model S40

Williams Oil-O-Matic Heating Corporation

Manufacturers of Automatic and Manually Controlled Fuel Oil Burners, Year 'Round Air Conditioning Systems, Williams Ice-O-Matic Compressors

Bloomington, Illinois

New York, N. Y., 2014 Graybar Building



WILLIAMS OIL-O-MATIC HEATING

A Complete Line

Williams Oil-O-Matic offers six Oil-O-Matic burner models—a genuine Williams Oil-O-Matic for every size and type of house, for every apartment, public building or commercial structure.

Complete boiler burner and furnace-burner units; a new type of oil-burning automatic water heater; an oil-burning range burner for heavy duty ranges are available.

Heating Capacities of Oil-O-Matic Burners

Model	Domestic Shipping Weight, Lb	Length, in.	Width, in.	Height, in.	HP	RPM	Gals. Fuel Oil per Operating Hour: Minimum	Maximum
K-150	125	29	15¼	19¾	1/10	1800	½	1½
K-3	145	30	15⅝	20¾	1/10	1800	1	3
K-4.5	170	33½	20¼	22¼	1/5	1800	2½	4½
K-7	175	32½	22¼	21¾	1/5	1800	4	7
J-1800	255	45	20	24	1/2	1800	8	15
JJ	295	50	32	24	1	1800	12	25

Standard draft pipe 11 in. 17-in. length draft pipe optional.

Standard electric current is 110-volt, 60-cycle.

Water Heaters

WHA*	600	57	22½	28	1/10	1800	....	½
WHB†	885	75	23	28	1/10	1800	½	1
WHC‡	1385	90	29	34	1/10	1800	1	1¾

*Output: 90 F rise, 60 gal per hour. †Output: 90 F rise, 120 gal per hour. ‡Output: 90 F rise, 210 gal per hour.

How to Decide Size of Burner

For low pressure domestic boilers, 1 gal of fuel oil per hour (140,000 Btu) is required for approximately:

300 sq ft of steam radiation or its equivalent.

480 sq ft of hot water radiation or its equivalent.

70,000 Btu when using hot air furnace ratings.

24 sq ft steam boiler heating surface (or 2.2 hp).

For exact detail data, see Oil-O-Matic Installation and Service Manual.

Oil-O-Matic Water Heaters

3 sizes cover ordinary home use, large home and heavy duty requirements. The Oil-O-Matic oil burner, combustion chamber, water reservoir, and automatic controls combined in one compact unit.

Complete Boiler-Burner and Furnace-Burner Units

A complete line of boiler-burner and furnace-burner units equipped with OIL-O-MATIC burners are available.

Boiler-burner units automatically provide domestic hot water supply the year 'round. Indirect heating coils may be included. Furnace-burner units provide winter air conditioning.

Underwriters' Listing

By Underwriters' Laboratories, Inc.

Fuel Oil Range Burners

Williams Oil-O-Matic fuel oil burners as an integral part of Oil-O-Matic South Bend heavy duty ranges.

Year 'Round Air Conditioning

Williams Air-O-Matic provides cooling and dehumidification in summer, heating and humidification in winter, together with ventilation, air circulation and cleansing throughout the entire year. This complete year 'round air conditioning is provided by one integrated system.

Refrigeration is furnished by a new type of absorption unit, using low pressure steam as its source of energy, developed specifically to better meet the unique requirements of air conditioning. Both refrigerant and solvent are essentially non-toxic, non-explosive, and non-combustible under normal atmospheric or normal operating conditions. Air-O-Matic is an automatic-fired complete year around temperature and humidity control unit.

Air-O-Matic offers these principal advantages: quiet operation; low power requirement; negligible deterioration; economy of operation and maintenance; and mechanical simplicity. Low pressures prevailing in the absorption unit minimize the possibility of losing a refrigerant charge either when the unit is in operation or when idle.

Engineering Service

Available to architects. See *A. I. A.* File No. 30 G-1.

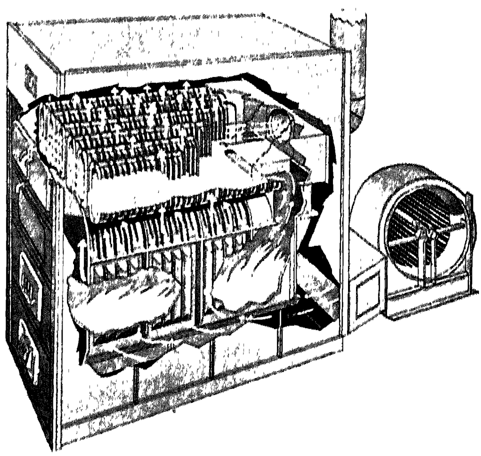
Acme Heating & Ventilating Co., Inc.

4224 Lowe Avenue, **Chicago, Ill.**

THE ACME HEATER—"It's in the Fins"

The Acme Heater has been designed by experienced heating engineers, and is constructed by expert craftsmen. It combines the best practice in the design of direct transmission heaters, with marked improvements resulting from practical experience.

Burns Any Kind of Fuel: A direct-transmission heater, such as the Acme, is not dependent upon the kind of fuel used—any type of fuel may be burned. Suitable grates may be provided so that bituminous, semi-bituminous, anthracite coal, or other solid fuels may be used with equal efficiency. Replacement of grates and linings by proper refractory material permits the use of automatic stokers or oil burning equipment.



Phantom View of Acme Heater Showing Flow of Gases and Air Travel.

Physical Data—Large Series

Size No.	Dimensions Lgth	Width	Ht	Grate sq ft	Heat Surf. sq ft	Free Area sq ft Min.	Free Area sq ft Max.	Wt. Lb	Max. Capacity Btu
7	6′-6″	4′-0″	7′-0″	10.31	260	6.55	10.25	5900	900,000
8	8′-1″	4′-0″	7′-0″	11.91	340	7.73	12.50	7000	1,100,000
9	9′-8″	4′-0″	7′-0″	13.06	430	8.91	14.75	8000	1,300,000
10	11′-3″	4′-0″	7′-0″	14.43	500	15.82	22.62	9300	1,500,000

Junior Series

2	4′-6″	3′-6″	5′-8″	3.9	136	4.7	4.7	3200	350,000
3	6′-0″	3′-6″	5′-8″	6.1	183	5.9	6.9	4800	527,000
4	7′-6″	3′-6″	5′-8″	7.2	230	7.1	9.1	5000	634,000
5	9′-0″	3′-6″	5′-8″	9.3	280	8.3	11.3	6000	800,000

Note: For automatic firing add 10% to ratings given.

Large Combustion Chamber: Provides ample space for ignition of gases of combustion, regardless of kind of fuel used. The unusually large combustion chamber, acting as "primary" heating surface, effects efficient transfer of heat, because of the great temperature difference between the burning gases inside the chamber and the air passing over the outside surface.

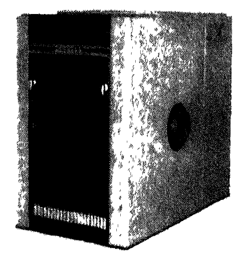
Efficient Radiator Section: Although the heating surface of the combustion chamber is large and efficient, still more heat must be extracted to obtain satisfactory overall efficiency. The "phantom view" as shown reveals how the gases of combustion enter the rear smoke chamber, flow to the front of the heater, and return again to the smoke-box. The gases are held in intimate contact with the heating surface, six times the length of the heater, before they are permitted to escape.

High Ratio of Heating Surface to Grate Area: The radiator tubes are covered with extended surfaces, or fins, typical of those used on indirect heating coils. The long, oval tubes of the radiator provide an exceptionally large heating surface which, combined with the surface of the combustion chamber, affords a remarkably high ratio of heating surface to grate area.

Balanced Construction of the Acme Heater provides ample free area and allows proper velocity of the air to be heated. Moreover, this air is brought into direct contact with as much heating surface as possible, resulting in the Acme of Efficiency.

SERIES 50 OIL-FIRED AIR CONDITIONING UNIT

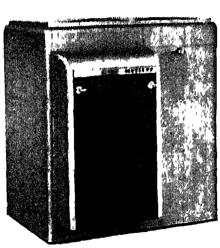
Designed and constructed to meet the needs and purse of the moderate-sized home, this unit automatically heats, filters, humidifies and circulates the air within the home efficiently and economically. The heating drum and radiator are made of heavy gauge steel, all electric welded, with no joints. Uniform distribution of the conditioned air is secured by the quiet, efficient Mueller fan. Filters furnished are of ample area, and with large dirt-holding capacity. Series 50 unit is available with a Mueller Vaporizing or Pressure Atomizing type oil burner. If desired, any standard burner may be used. Three sizes, from 100,000 to 225,000 Btu per hour.



MUELLER CLIMATOR FAN-FILTER UNITS

For positive distribution of air in old or new construction, wherever air is to be moved, there is a Mueller Climator Fan-filter unit available. These units are designed to provide better, cleaner, and healthier home heating. Fans are designed in accordance with latest aerodynamic principles, providing uniform air distribution from top to bottom of the fan outlet. Filter area is adequate to handle the air requirements. Fans have ample air delivery capacity for the addition of cooling, if desired. Smaller size fans are equipped with top-mounted motors. Motor mounting is designed and cushioned so as to absorb motor vibration, thereby eliminating possibility of transmission of sound to the duct work. These units are available in a range of sizes capable of handling practically any requirement.

SERIES "EPS" GAS-FIRED AIR CONDITIONING UNIT

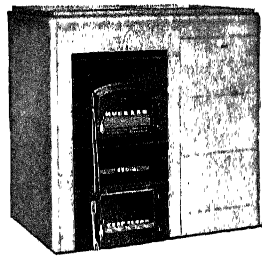


Designed and styled for the modern home, this Mueller unit meets every requirement for an automatic Winter air conditioning unit. Provides balanced distribution of filtered, humidified warm air in ample volume to every room. Heating unit consists of Mueller steel Heatspeeder sections, providing quick heat in desired volume. The fan operates quietly and efficiently, with ample capacity for any requirement. Filters thoroughly clean the air. Humidity is supplied automatically. Available in three sizes with AGA input ratings from 90,000 to 180,000 Btu per hour.

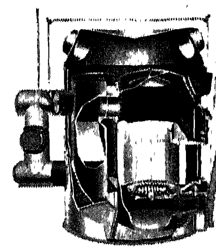
Mueller Heaters For All Fuels

A Complete Line for All Purposes

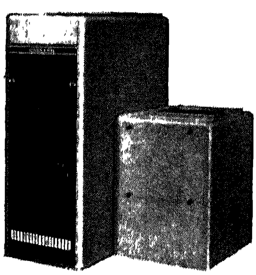
Return Flue all-cast Furnace. 18 in. to 30 in. firepots, single and double firedoor styles. Available in round, galvanized or square, lacquered casings.



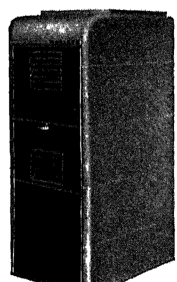
Series P-400 Steel coal-fired fan-filter-furnace unit. Also available with round casing for gravity operation. Four sizes, 20 in. to 27 in. drums.



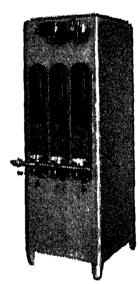
Series "200" Steel coal-fired furnace. Sizes, 20 in. to 34 in. drums. Available also with square lacquered casings for forced air installations.



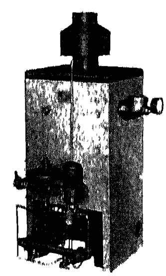
Gas Era Cast Iron air conditioning furnace. *AGA* input rating, 65,000 Btu per hour, per section. Wide range of sizes and air delivery capacities.



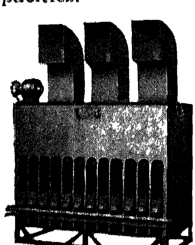
Gas-fired air conditioning furnace. A.G.A. input ratings from 60,000 to 100,000 Btu per hour. Wide range of air deliveries.



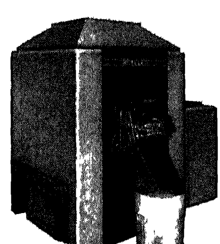
Muelleraire unit. Gas-fired with fan, filters, and humidifier. 5 sizes—*AGA* input ratings, 72,000 to 180,000 Btu per hour.



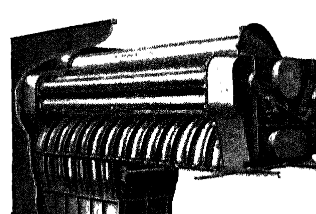
Series "AE" Gas Boiler. *AGA* ratings, 180 to 1,260 sq ft steam; 290 to 2,015 sq ft hot water.



Gas Era Unit Heater. Sizes from six to twenty sections, with A.G.A. ratings from 216,000 to 720,000 Btu per hour.



Series "SA" stoker-fired furnace, with fan-filter unit. Any stoker may be used. Capacities, 110,000 and 175,000 Btu.



Horizontal Tubular Heaters, for schools, churches and other large buildings. Three sizes, with capacity range from 1,188,000 to 1,390,000 Btu per hour.

Complete catalogs on each of above units available upon request.

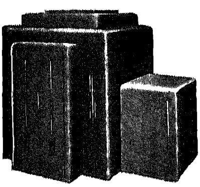
The Meyer Furnace Company

Peoria, Illinois

Manufacturers of Heating and Air Conditioning Equipment for Coal, Gas and Oil Burning

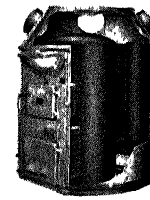
Branches and Distributors

Kansas City, Mo.
Omaha, Neb.
Green Bay, Wis.
Pittsburgh, Pa.
Philadelphia, Pa.
St. Louis, Mo.
Columbus, O.
Minneapolis, Minn.
New York, N. Y.
Atlanta, Ga.



WEIR Conditioned Air Unit

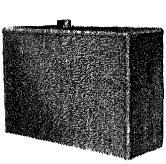
The WEIR Air Conditioner for solid fuels (hand or stoker fired) embodying the famous WEIR Steel Furnace is a complete unit for winter air conditioning, including air cleansing, humidifying and forced circulation as well as heating. Encased in furniture steel with baked enamel finish. Automatic controls for dampers, blower, humidifier.



WEIR Gravity Heater

No.	Grate Surface (Sq Ft)	Ratio Htg. to Grate Surface	Smoke Outlet Diam. (In.)	Gravity Circulation: Casing Dimen. Round (In.)	Gravity Circulation: Casing Dimen. Rect'lar (In.)	Gravity Circulation: Rated Output At Reg. (Btu.Hour)	Gravity Circulation: Rated Output Pipe Area (Sq In.)	Fan Circulation: Casing Dimen. (In.)	Fan Circulation: Air Delivery (CFM)	Fan Circulation: Rated Output at register (Btu.Hour)
621	1.26	41.2	9	48	.....	54,400	400	......	......	....
624	1.78	33.9	10	52	47x50	73,600	541	47x90	1200	92,000
628	2.32	29.2	10	54	50x52	94,100	692	50x99	1600	118,000
630	3.08	26.4	10	58	54x56	119,000	875	54x103	2000	148,000
633	3.82	22.7	10	65	56x64	138,000	1015	56x110	2300	172,000
636	4.74	19.4	10	67	56x66	160,000	1180	56x118	2700	200,000
540	6.25	19.3	12	..	.....	....	....	54x100	6100	360,000
544	7.60	18.5	12	..	.....	....	....	58x108	7500	440,000

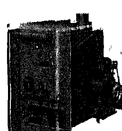
The WEIR Oil-Fired Air Conditioner does a complete job of winter air conditioning. Designed for oil fuel and forced circulation. Completely self-contained in *low* compact casing enclosing burner and blower as well as heater and all controls, yet with everything easily accessible.



WEIR Oil Fired Air Conditioner

The MEYER Gas-Fired Air Conditioner automatically provides completely controlled winter air conditioning. Efficient performance, compact design, modern appearance. Heavy gauge welded steel heating section; die-formed furniture steel casing. *A.G.A.* approved.

No.	Input at Burner (Btu/Hour)	Output at Bonnet (Btu/Hour)	Vent Diam. (In.)	Dimensions W. (In.)	Dimensions L. (In.)	Dimensions H. (In.)	Air Delivery 1/4 In. S.P. (CFM)	Motor Size (HP)
WEIR Oil-Fired Air Conditioner								
A-100-M	140,000	110,000	6	25	71	48	1300	1/4
A-150-M	212,000	165,000	8	32	78	48	1950	1/3
A-175-M	260,000	195,000	8	56	68	48	2300	1/3
A-225-M	335,000	250,000	8	63	68	48	3000	1/2
MEYER Gas-Fired Air Conditioner								
F-10	110,000	88,000	5	32	69	48	1200	1/4
F-15	165,000	132,000	6	41	69	48	1800	1/4
F-20	220,000	176,000	7	53	69	48	2400	1/3
F-30	330,000	264,000	9	69	69	48	3600	1/2
F-45	495,000	396,000	10	97	69	48	5400	3/4



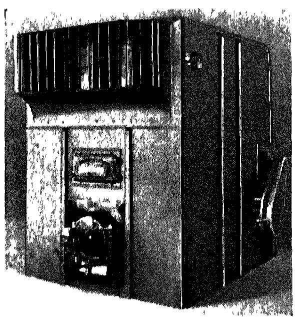
MEYER Gas Fired Air Conditioner

Airtherm Manufacturing Company

710 S. Spring Ave., **St. Louis, Mo.**

THE ENGINEERED LINE OF UNIT HEATERS

DIRECTHERM WARM AIR HEATERS FOR OIL, GAS OR COAL are available in six standard sizes with capacity from 300,000 to 1,500,000 Btu. They are made of heavy gage steel plate, with major sections all welded and flue gas headers are readily cleanable.



VENTILATION AND AIR RE-CIRCULATION. The Directherm can be easily hooked up for outside air intake. Air filters may be used in the intake box when desired.

INSTALLATION. These units, when assembled, require nothing more than a stack and an electrical connection (plus a gas or oil fuel hook-up) and can be made fully automatic in operation when using oil, gas or stoker.

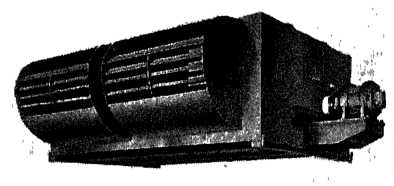
PORTABILITY. Directherm Heaters may be readily removed from one location to another or from one plant to another when building expansion programs demand such alterations in the heating system.

DUCT WORK. While the Directherm Heater will provide thorough heat distribution without duct work, where necessity requires it may be hooked up with a system for further heat distribution. The fan equipment is of ample capacity to overcome duct resistances.

ENGINEERING SERVICE

The Airtherm Mfg. Co. Engineering Department and District Representatives are at all times available for consultation. At your request we will place experienced engineering aid at your disposal. *Representatives in all principal cities.*

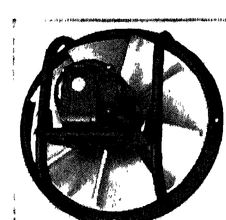
AIRTHERM LINE OF UNIT HEATERS represent a full range of capacities in all types.



THE AIRBLANKET. A revolutionary type of heating unit which holds the heat in the working zone through the use of the over-riding cold air blanket. This unit is available either in the centrifugal fan type as illustrated above or in the propeller fan type. They are designed for wall or ceiling mounting and are especially recommended for high ceiling jobs. Bulletin No. 210 contains complete details of the Airblanket method of heating.

THE AIRHEATOR. A highly efficient, large capacity, centrifugal fan heater for all types of installations, available for floor, wall or ceiling mounting.

THE AIRVECTOR. A newly redesigned propeller fan type unit heater backed by 30 years of manufacturing experience. The Airvector is available for ceiling suspension or mounting from the floor on a recirculating stack.



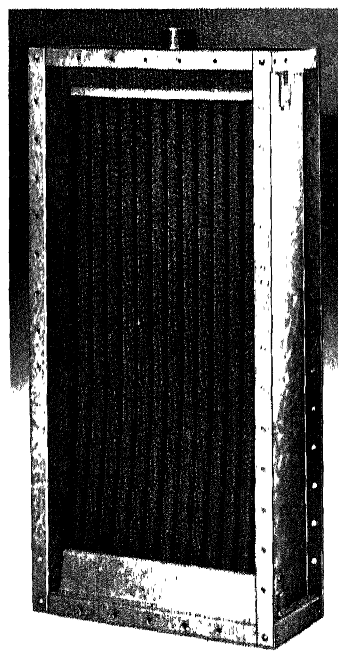
AIRTHERM EXHAUST FANS have been expressly designed to meet industrial and general requirements for rugged, heavy duty type fan of simplified construction that would minimize both installation and maintenance problems. Capacity charts and literature will be forwarded immediately on request.

FEDDERS
MANUFACTURING COMPANY, INC.
85 TONAWANDA ST.
BUFFALO, NEW YORK

TYPE K HEATING COILS

Strong, rigid casings . . . large cylindrical headers . . . full-floating protection against overall expansion . . . top header tri-point supported by center anchorage brackets and drop forged bronze trunnions . . . knee action relief of differential expansion among tubes . . . scale breaker-tube orifices . . . floating type tube supports . . . permanently bonded fins and tubes.

7 Standard Face Widths 12¾ in. to 36⅛ in.
18 Standard Face Lengths 1½ ft to 10 ft.
3 Fin Spacings.
1 or 2 Row Deep Coils of all 3 Fin Spacings.



BASIC TEMPERATURE RISES

Coil Type	K15	K16	K18	K25	K26	K28
Air Temp. Rise° F*	36.1	52.3	62.5	67.0	93.0	108.0

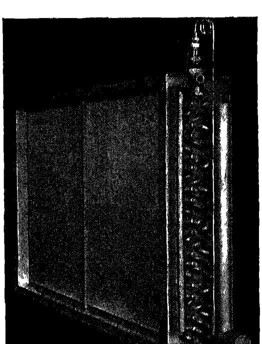
*Enter. Air 0°F, 500 FPM, 5 lbs steam

TYPE M MODULATING COILS

Non-freeze design with general construction similar to Type K Coils. Wide range of standard sizes.

TYPE B BOOSTER HEATING COILS

Featuring same advanced engineering as Type K Coils. Complete range of sizes.

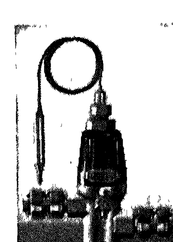


COOLING COILS

Type R for Refrigerant.
Type W for Water.
Equipped with flat fins for rapid air-side condensate removal.
Built in wide range of standard sizes.

THERMOSTATIC EXPANSION VALVES

High capacity Thermostatic Expansion Valve for use with Freon or Methyl. Capacities up to 20 tons Freon.



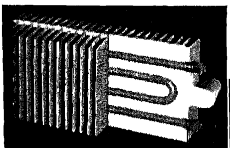
ELECTRIC AIR HEATER CO.

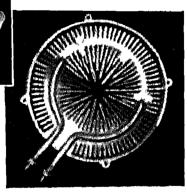
DIVISION OF AMERICAN FOUNDRY EQUIPMENT CO.

MISHAWAKA, INDIANA

Manufacturers of

ELECTROMODE



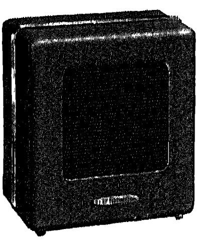


Cross-section Views of Electromode Heater Grids—Circular and Rectangular Types

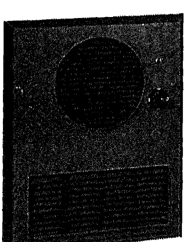
ALL-ELECTRIC AIR HEATERS

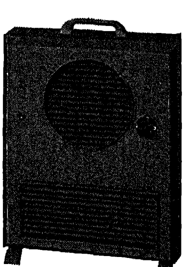
Cast aluminum grids in electric heaters are a radically different innovation from the popular conception of electric heating units. Aluminum, a metal of highest thermal conductivity, is cast on a tubular element. This process seals the heating element, preventing all oxidation.

HOME & OFFICE HEATERS



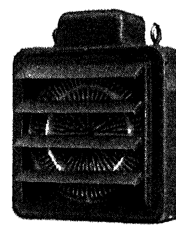
Model PJ-15—A package of comfort for chilly bathrooms, nursery, den, office, workshop, etc. Can be used in average baseboard receptacle on 115 volts.



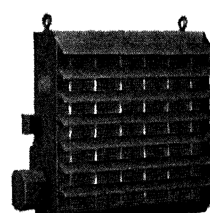


The Electromode BILT-IN-WALL and PORTABLE models, illustrated at left, in addition to the exclusive cast aluminum element, feature down-draft warm air delivery. This scientific principle shows remarkable efficiency in warm air distribution. Both models made in capacities from 1 KW to 5 KW (3415 Btu to 17,075 Btu). The modern cabinet is the same for all capacities and designed for 2 in. x 4 in. wall construction. Approximate wall opening required is 14⅜ in. wide by 18½ in. high.

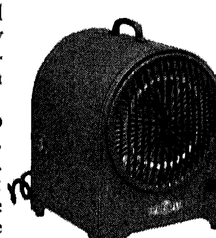
INDUSTRIAL HEATERS



Circular Grid - Made in capacities from 2 KW (6830 Btu) to 9 KW (30,735 Btu). Each heater is provided with eye bolts and adjustable louvers. The cabinets are made of heavy furniture steel.



Industrial Unit - Made in capacities from 10 KW (34,150 Btu) to 90 KW (307,350 Btu). The cabinet is made of heavy steel and furnished with eye bolts and adjustable louvers.



Industrial Portable A heavy duty portable Electromode made in capacities from 1 KW (3415 Btu) to 9 KW (30,735 Btu). Each heater is a self-contained unit complete with cable and with a suitable plug cap.

Send to the factory for literature and complete details on any or all Electromode models. Our Engineers will be glad to estimate heater sizes for your application and submit full recommendations. Qualified engineering representatives are located in the principal cities for prompt service. A wide variety of models and capacities are available for any kind of a space heating job.

Hastings Air Conditioning Co., Inc.

Hastings, Nebr.

Manufacturers of

Air Conditioners.
Unit Heaters.
Utility and Package Blowers.

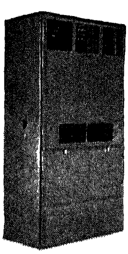
Dealers and Representatives in Principal Cities

A Complete Line of Highly Successful COLD WATER Air Conditioners. Capacities listed depend on entering air and water temperatures.

All equipment available for combination heating and cooling.

FLOOR MODELS

Floormasters Unusual design and special features permit maximum installation possibilities with minimum floor space and installation costs.

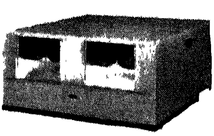


Air Delivery 2240 cfm. Cooling Capacity 3 to 6 tons. Dimensions - Height 93 in., Width 48 in. Depth 25 in. Motor - ½ hp. Filters 3 16 in. x 25 in.

Royal For offices, homes, hospitals, etc.

Air Delivery 500 cfm. Cooling Capacity 1 to 2 tons. Motor 1/6 hp. Filter 1 16 in. x 25 in. Dimensions Height 40 in., Width 28 in., Depth 20½ in.

CENTRAL PLANTS



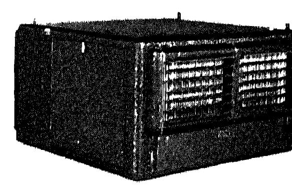
Sectional construction for ease of handling. Motors inside mounted to provide very neat appearing compact units.

SPECIFICATIONS

Size	CFM	Motor Hp	Filters	Capacity Tons
CP 30	3,000	1	5	4-9
CP 40	4,000	1	8	6-12
CP 60	6,000	2	10	9-18
CP 80	8,000	3	12	12-24
CP120	12,000	5	20	18-36

GENERAL UTILITY MODELS

Master—Singly or in multiple are suitable for any business or space size. Large jobs handled without duct work by proper location of units.



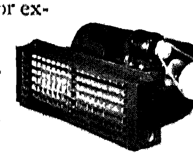
Air Delivery 2,240 cfm. Cooling Capacity - 3 to 6 tons. Dimensions - Height 29 in., Width 49 in., Depth 50 in. Motor - ½ hp. Filters 4 16 in. x 23 in.

Majestic—Similar to Master except size. Air Delivery 2,240 cfm. Cooling Capacity 1½ to 3 tons. Motor ¼ hp. Filters 2 16 in. x 25 in. Dimensions - Height 26 in., Width 28 in., Depth 40 in.

Zephyr—Same capacity, motor and filter as the Royal. For use where suspended or concealed units are desired. Dimensions Height 26 in., Width 24 in., Depth 28 in.

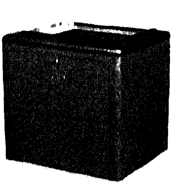
UNIT HEATERS

Centrifugal Type for extreme quietness and efficiency.

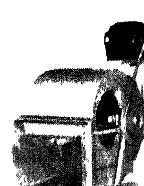


Steam pressure - to 150 in. per sq in. Finish Brown wrinkle enamel and stainless steel louvers.

PACKAGE AND OPEN TYPE BLOWERS



May be knocked down for narrow doorways. Finished in attractive green wrinkle.



Utility type blowers are available with or without motors and in any discharge desired.

All sizes from 9 in. to twin 21 in.
Air deliveries from 1000 cfm to 16,000 cfm.

Write for Catalogues, Literature, or Information

F. Jaden Manufacturing Co., Inc.

ESTABLISHED 1914

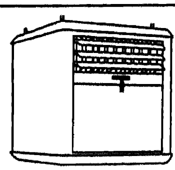
Hastings, Nebraska

Golden Rod
Fresh AIR CONDITIONING

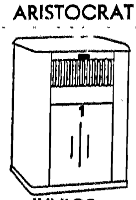
SPECIALISTS IN WATER COOLED AIR CONDITIONING

Unit Conditioners (*Shown at Left*)

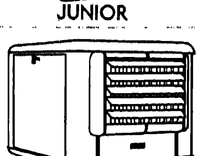
These models applicable to any cooling installation large or small. Unsurpassed efficiency—sturdy construction. 50-100 per cent fresh air recommended, assuring complete ventilation and odor elimination. For heating, steam coils can be furnished on Universal and Imperial units, or circulating hot water may be used with all models.



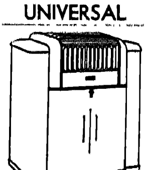
ARISTOCRAT



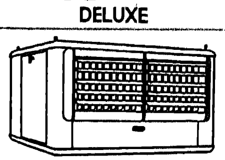
JUNIOR



UNIVERSAL



DELUXE



IMPERIAL

FURNACE TYPE CENTRAL PLANTS

Three models designed for use with gravity hot air furnaces. Provides complete summer and winter conditioning, including winter humidification.

CENTRAL PLANTS

Designed for larger installations employing extensive duct systems. Two standard models—specials quoted on request.

CENTRIFUGAL UNIT HEATERS

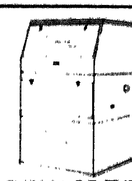
Designed for installations requiring silent operation. Ideal for use with duct systems.

ATTIC VENTILATION

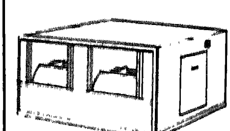
Designed with centrifugal blowers to insure positive displacement of air. Easily installed—sturdy—silent.

BLOWERS

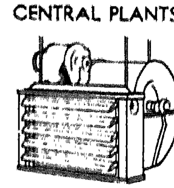
Complete line of high quality cabinet, base type, utility and twin blowers. 16 gage welded construction—bronze bearings—ground steel shafting.



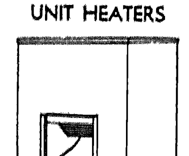
FURNACE TYPE



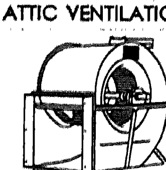
CENTRAL PLANTS



UNIT HEATERS



ATTIC VENTILATION



BLOWERS

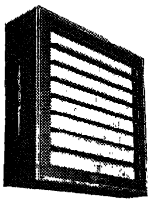
Wire or Write for Complete Details

Kramer Trenton Co.

Manufacturers of

HEATING, COOLING AND REFRIGERATION DEVICES

Trenton, New Jersey

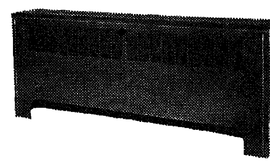


KRAMER UNIT HEATERS

All-copper heating element. Oval-section tubes with hair-pin bends. High-discharge air velocity insures proper heat distribution. For pressures up to 150 lb.

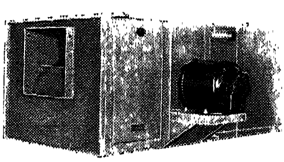
Send for Bulletin H-141

KRAMER COPPER CONVECTORS



All-copper heating element. Oval tubes with fins metallically fused to tubes. Noiseless operation. Guaranteed for operating steam pressures up to 50 lb.

Send for Bulletin H-240

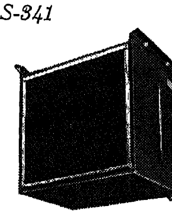


HEATING and AIR CONDITIONING UNITS for Residential Use

Designed for split-system installations. A range of sizes adaptable to residential requirements. Rubber mountings and flexible connections minimize noise.

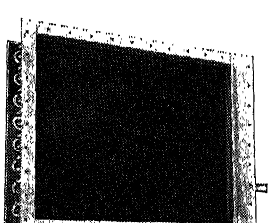
Send for Bulletin SS-341

KRAMER COMFORT COOLERS



Suspended type for small tonnages—1 to 3 tons—and for remote compressor operation. All-copper coils. Specially designed grille for proper diffusion.

Send for Bulletin R-141

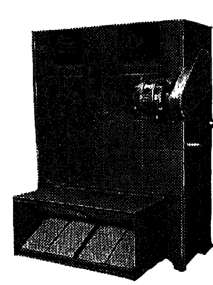


KRAMER TURBO-FIN

For blast heating and cooling. All-copper blast surfaces; fins metallically fused to tubes. Air side flow-disturbers. Coil finished in electro tin plate for permanence.

Send for Bulletin AC-540

KRAMER AIR CONDITIONING UNITS
Ceiling and Floor Type



Wide variety of sizes and capacities—2 to 30 tons in cooling; 65,000 to 1,280,000 Btu per hour in heating. Accurately rated. All-copper Turbo-fin coils; fiberglas air filters. Complete cabinet types for either floor or ceiling mounting.

Send for Bulletin AC-440

McQuay, Inc.

1602 Broadway, N.E., **Minneapolis, Minn.**

MANUFACTURERS OF AIR CONDITIONING EQUIPMENT

Sales Offices in all Principal Cities

- **Air Conditioners**
- **Air Conditioning Coils**
- **Blast Heating Coils**
- **Refrigeration Coils**
- **Convection Radiation**
- **Unit Heaters**
- **Unit Coolers**

PROVEN McQuay PRODUCTS

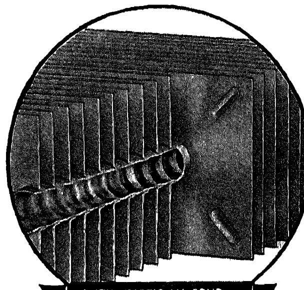
- **Comfort Coolers**
- **Blower Coolers (Suspended & Floor Type)**
- **Room Coolers (Cabinet Type)**
- **Ice Cube Makers**
- **Icy-Flo Accumulators**
- **Zeropak Low Temp. Units**

THE EXCLUSIVE McQUAY FRICTIONAL BOND FIN-AND-TUBE COIL ASSEMBLY

The McQuay Fin and Tube assembly in all McQuay coils and cores is one of the reasons McQuay products are considered "Tops in Over-All Efficiency" by many heating and refrigeration authorities.

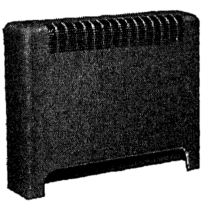
Heat transfer efficiency primarily depends on three elements in coil construction. First, "Area of Contact," Second, "Contact Pressure" and finally "Quality of Contact" between collar and tube.

In McQuay coils *all three* necessary elements are found developed to their highest degree. The famous McQuay "Wide Fin Collar" plus Exclusive Hydraulic Expansion together with the polished surface, secured by "spinning" the fin collar, truly provides the last word in Heat Transfer.



WITH FRICTIONAL BOND COIL CONSTRUCTION

McQUAY CONVECTORS



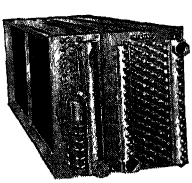
CONVECTOR

The new modern McQuay convector has been designed to meet all qualifications for reliability, efficiency, beauty, and cleanliness. The enclosures, exposed, recessed and concealed, with their pleasing lines and rounded corners, offer unlimited possibilities for harmonious decorative treatment.

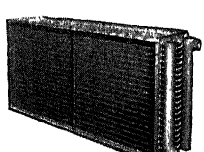
The heating element is constructed of a series of round seamless copper tubes to which are attached straight fins.

All McQuay convector enclosures are constructed from high-grade furniture steel, properly reinforced to make a sturdy cabinet. The enclosures are available in fourteen individual types with three styles of grilles.

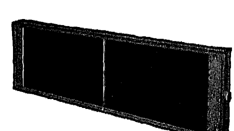
We offer the services of our engineering and design departments to help solve your special heating problems.



COMBINATION COOLING COIL



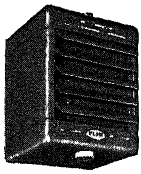
WATER COIL



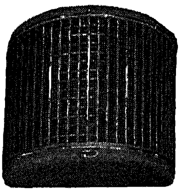
STEAM BLAST COIL

MORE THAN 1,000,000 STANDARD COIL SIZES

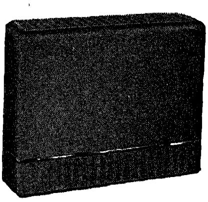
McQuay manufactures the most complete line of Standard Coils in the Industry.
Coils for Heating—up to 10 rows deep using low or high pressure steam or hot water.
Non-Freeze—(steam distributing tube) type coils 1 and 2 rows deep.
Removable Plugs—(cleanable tube) type coils one to 12 rows deep.
Water Coils for Cooling—one to 12 rows deep.
Direct Expansion Coils—for cooling 1 to 10 rows deep.
Refrigeration Coils—all types and sizes.
Special Coils—of various materials furnished on order for special applications.



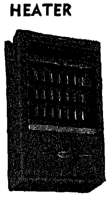
STANDARD UNIT HEATER



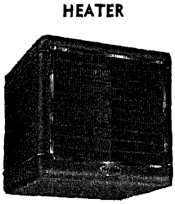
RADIAL UNIT HEATER



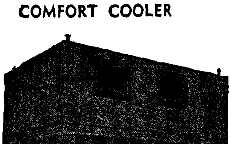
CABINET TYPE UNIT HEATER



GAS FIRED UNIT HEATER



COMFORT COOLER



AIR CONDITIONER (YEAR-ROUND)

UNIT HEATERS

The new and exclusive McQuay Radial Heater joins a distinguished old family of proven heating units—the Down Flow—the Gas Fired—the Cabinet Unit—the Large Blower Type, and the veteran Standard Unit Heater, making the McQuay line the most complete in the industry.

This newest McQuay development provides wide uniform heat distribution, lower installed cost and fine appearance, combined in one unit. One Radial Heater can now be used in place of two or more Standard Units—effecting important savings in piping expense as well. Actually it is a deluxe unit heater at lower cost than standard units, in most cases.

All McQuay unit heaters feature the exclusive Frictional Bond coil construction. All types of Heaters are furnished in a wide range of sizes with motors to meet all electrical current characteristics, making it convenient to select the proper size heater for every installation.

McQUAY COMFORT COOLERS

Made in two types—one for use with water or brine; another for freon or methyl chloride. Eight sizes in each type—all with 4-speed motors.

AIR CONDITIONERS—COLD WATER AND FREON TYPES

Choice of recirculation of indoor air, entire intake of outside air, or a combination of both. Cold water or brine used in one type; freon or methyl chloride in another. Modern "sound isolated" construction assures quiet operation. Capacities to 6 tons.

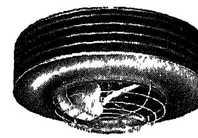
LARGE TYPE AIR CONDITIONING UNITS

Suspended and floor types, cools, dehumidifies, filters, and circulates air in summer; heats, humidifies, filters and circulates air in winter. Extreme flexibility and accessibility "built-in". Cooling capacities from 5 to 50 tons in both Suspended and Floor Type.

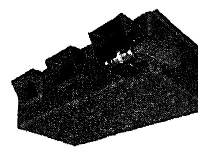
McQUAY ICY-FLO ACCUMULATORS

The new practical "Storage-Battery" for refrigeration effect is now available for handling heavy loads of short duration.

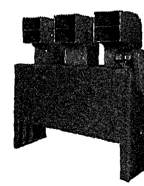
New Descriptive Bulletins are ready on all McQuay Products. Write McQuay, Inc., Minneapolis, Minnesota.



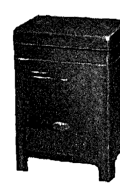
DOWN FLOW UNIT HEATER



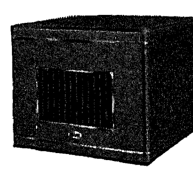
BLOWER TYPE UNIT HEATER



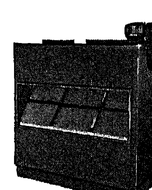
BLOWER TYPE UNIT HEATER



COMFORT COOLER



AIR CONDITIONER



AIR CONDITIONER (YEAR-ROUND)

Modine Manufacturing Company

Heating and Air Conditioning Division

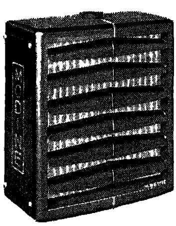
General Offices: 17th and Holburn Sts., **Racine, Wis.**

Factories at Racine, Wis. and La Porte, Ind.

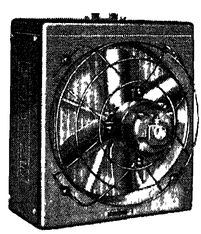
Branches in all Principal Cities

Complete information on the following products including engineering data and prices, can be secured by writing to the Modine General Offices at Racine, Wisconsin—or by communicating with nearest Modine representative.

MODINE UNIT HEATERS



Front View



Back View

The New Modine Unit Heater incorporates in its design, many features which contribute to more satisfactory and economical industrial and commercial heating.

Sound Silenced—Interior surface of casing is coated with acoustical-mastic, deadening noise from within. Venturi fan shroud, integral with casing, quiets air in-rush sound. Velocity generator eliminates air-rush noise peaks. Concentric rings of fan guard act as vibration dissipators.

Safety Fan Guard—Provides staunch, steel safeguard again hazard of unshielded fans. Safety is built in as standard equipment.

Protection Against Rust—Available by Bonderizing—When applied, Bonderizing of casing and sheet metal parts makes them resistant to formation and progress of rust. It holds finish to metal, making it more durable and permanently fine in appearance.

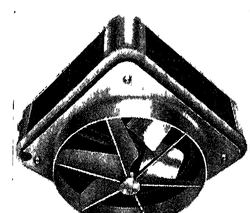
And These Additional Features—(1) *Velocity Generator* gives greater heat throw without increasing power requirements. (2) *Patented Expansion Bend* permits tubes to stand extreme expansion. (3) *Direct Pipe Suspension* permits installation without hangers—saves cost.

Unit Heaters—Capacities and Dimensions
(In Inches)

Model No.	Over-all Height	Width	Depth Less Motor	E.D.R.	C.F.M.	Motor R.P.M.
74	11¾″	10″	6″	74	296	1580
104	16¾″	14¼″	9″	104	350	1580
140	16¾″	14¼″	9″	140	540	1580
172	19½″	14¼″	9″	172	661	1580
206	22½″	19″	11″	206	843	1120
252	22½″	19″	11″	252	980	1120
304	22½″	19″	11″	304	1290	1120
362	24″	19″	11″	362	1450	1120
414	22½″	19″	11″	414	1370	1120
514	27½″	23″	11½″	514	2250	1120
606	27½″	23″	11½″	606	2440	1120
711	27½″	23″	11½″	711	2430	1120
808	29½″	26½″	13″	808	2760	1120
904	29½″	26½″	13″	904	3370	1120
1050	33¾″	26½″	13″	1050	3770	1120
1200	33¾″	26½″	13″	1200	4120	1120
1380	33″	34″	11″	1380	4960	1120
1610	30″	50¾″	11″	1610	5920	1120
2030	30″	54¾″	11″	2030	7720	1120

All above models are available with variable speed motors. Units for hot water application also available.

MODINE VERTICAL DELIVERY UNIT HEATERS

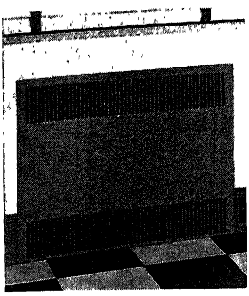


Modine Vertical Delivery Unit Heaters are indicated wherever conditions call for more directly downward delivery of air than is provided by the conventional design of unit heaters. In factories where high clearance is essential, as for craneways, Modine Verticals serve excellently. Over store and office doorways, they fit perfectly into the need for a blast of warmed air to offset the in-pressing winter gales. In a room where only one unit heater is needed, the Modine Vertical gives delivery of heat to the entire perimeter of the room. Or by adjustment of the Modine Cone-Jet Deflectors, this delivered air may be concentrated in limited directions, even checked almost entirely to a single side of the unit.

Similarly, multiple installation of Modine Verticals may be controlled as to heated air deflection so as to give more delivery of heated air to outside walls, thus conforming to normal heating requirements. Modine Verticals are made in thirteen models. Write for catalog.

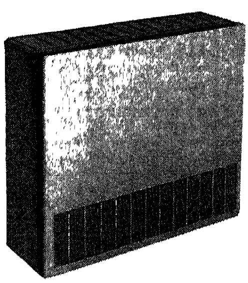
MODINE COPPER CONVECTORS

The popular copper radiators for commercial and public buildings, low cost houses, etc.—wherever the benefits of copper convector heating are desired. Attractive enclosures with removable fronts. Wide selection of grilles. Rust protection of enclosures available by Bonderizing. High capacity copper heating units. Enclosures are made in three types of Recessed and Floor and Wall Cabinets and Concealed (plaster-front) types. **Catalog 240-E.**

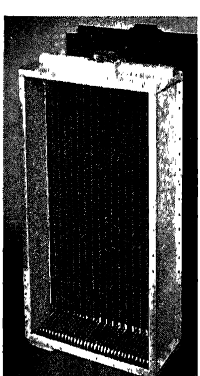


CABINET UNIT HEATERS

Modine Cabinet Unit Heaters are designed for the heating of offices, lobbies, corridors, auditoriums, etc. Used in conjunction with a steam or hot water system, they elimate the need for unsightly obsolete radiators. Models include the Floor Cabinet, Wall Cabinet, Ceiling and Recessed types—each available in three capacities: 105, 310 and 450 E.D.R. **Bulletin 840.**

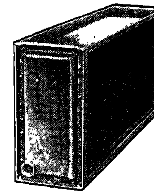


MODINE BLAST HEATERS



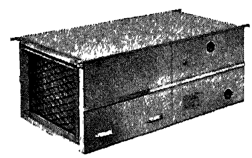
Made in over 250 sizes, types and capacities to meet the specific demands for heat transfer service. Outstanding features are: (1) Expansion Bend. (2) All steam carrying passages are cylindrical for greatest possible strength. (3) From inlet to outlet condenser is of copper or copper alloy. (4) Copper fins are bonded metallically to tubes. **Catalog 340.**

MODINE COOLING COILS



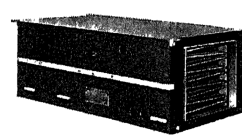
For use in central system cooling and air conditioning plants, Modine Cooling Coils, Cold Water Type, are installed with a blower fan and duct work. Adaptable where cold water or non-corrosive brine is used as the cooling medium. Coils are available in *cleanable* and *continuous Tube* types. **Catalog 540.**

UNIT COOLERS (Blower Type)



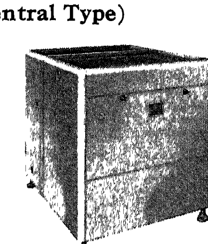
For stores and offices. This unit cools, cleans, dehumidifies and circulates the air. Equipped with powerful, yet quiet blower, extra deep cooling coils and large-area air filters. May be installed with or without duct work. Choice of cold water or Freon cooling coils. **Bulletin 440.**

AIR CONDITIONER (Apartment House Type)



A compact unit performing every function of complete winter and summer air conditioning—for apartments, hotel suites, residences, offices, and shops. Its compactness allows installation in a closet above shelving or in a hall above a false ceiling. Uses steam or hot water for heating; cold water or Freon for cooling. Two sizes. **Bulletin 638-B.**

AIR CONDITIONER (Large Central Type)



For residential and commercial year-'round air conditioning—may be used in straight air conditioning or split systems. Uses steam or hot water for heating and cold water or Freon for cooling. **Catalog 639.**

The Herman Nelson Corporation

General Offices and Factories at Moline, Illinois

Sales and Service Offices in the Following Cities:

Portland, Maine	Saginaw, Mich.	Nashville, Tenn.	Omaha, Nebr.
Boston, Mass.	Grand Rapids, Mich.	Memphis, Tenn.	Oklahoma City, Okla.
Westfield, Mass.	Cleveland, Ohio	Indianapolis, Ind.	Tulsa, Okla.
New York, N. Y.	Cincinnati, Ohio	Chicago, Ill.	Dallas, Texas
Watervliet, N. Y.	Toledo, Ohio	Milwaukee, Wis.	Missoula, Mont.
Syracuse, N. Y.	Washington, D. C.	Moline, Ill.	Denver, Colo.
Buffalo, N. Y.	Baltimore, Md.	Peoria, Ill.	Salt Lake City, Utah
Philadelphia, Pa.	Richmond, Va.	Des Moines, Iowa	Spokane, Wash.
Harrisburg, Pa.	Roanoke, Va.	St. Louis, Mo.	Seattle, Wash.
Pittsburgh, Pa.	Charlotte, N. C.	Kansas City, Mo.	Portland, Ore.
Johnstown, Pa.	Miami, Fla.	Emporia, Kans.	Los Angeles, Calif.
Detroit, Mich.	Birmingham, Ala.	Minneapolis, Minn.	San Francisco, Calif.

The complete line of quality heating, ventilating and air conditioning products manufactured by The Herman Nelson Corporation, makes possible the specification of **HERMAN NELSON** for industrial, commercial and public buildings of all types.

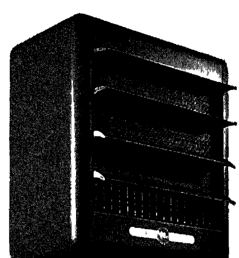
Industrial areas may be efficiently heated with Herman Nelson hiJet Heaters. The complete line of Horizontal Shaft or Vertical Shaft Propeller-Fan hiJets or the Blower-Fan Type hiJets provides a wide range of application and heating capacities.

Commercial buildings, too, are well adapted to the application of hiJet Heaters. The DeLuxe hiJet, as well as the Propeller-Fan Types, provide a handsome exterior, heating efficiency and quiet operation. For winter and/or summer air conditioning, the Year Around Air Conditioning Unit may be specified.

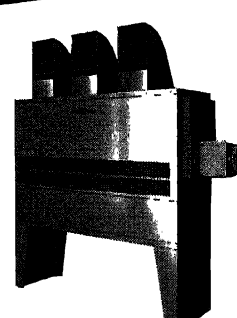
Public Buildings may be most satisfactorily provided with proper heating and ventilating with the Herman Nelson Air Conditioner For Schools. This equipment maintains desired room air conditions at all times, prevents overheating, eliminates drafts, and is particularly applicable to areas where large groups of people gather. Ventilation of toilets and laboratories may be accomplished with the Herman Nelson Exhauster.

Engineering Data applying to Herman Nelson Products may be obtained from our sales representatives or The Herman Nelson Corporation, Moline, Illinois.

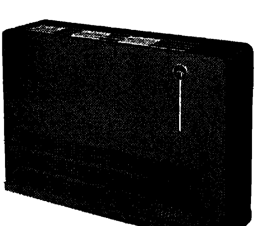
Vertical Shaft Propeller-Fan Type hiJet Heater discharges air vertically downward, or available to discharge at an angle to vertical in various directions.



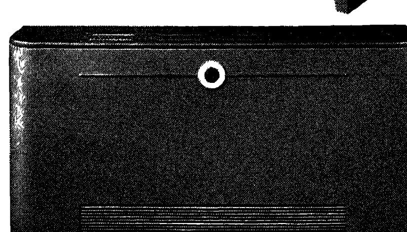
Horizontal Shaft Propeller-Fan Type hiJet Heater (with adjustable louvers) projects air downward in desired direction. Also available with attractive grille.



Blower-Fan Type hiJet Heater provides efficient heating of large areas. With by-pass damper if desired. For floor, wall, ceiling or inverted wall mounting.



DeLuxe hiJet Heater (at left) provides unusually large heating coverage. For floor, wall or ceiling mounting.



Air Conditioner for Schools (right) available for damper or radiator control in classroom or auditorium types.

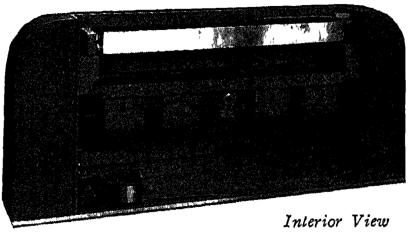
John J. Nesbitt, Inc.

Holmesburg, Philadelphia, Pa.

11 Park Place, New York City 205 W. Wacker Drive, Chicago, Ill.

Manufacturers of

THE NESBITT SYNCRETIZER Heating and Ventilating Unit,
sold by John J. Nesbitt, Inc., and American Blower Corporation;
NESBITT HEATING SURFACE with Dual Steam-distributing Tubes,
NESBITT SERIES H HEATING SURFACE, and
NESBITT SERIES W COOLING SURFACE,
sold by leading manufacturers of fan-system apparatus;
WEBSTER-NESBITT UNIT HEATERS and AIR CONDITIONERS
See page 987), distributed in U. S. A. by Warren Webster & Company.



Interior View

The Nesbitt Syncretizer—Series 400

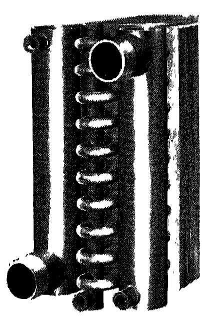
The last word in heating and ventilating units for schoolrooms, offices, etc., where the continuous introduction of outdoor air is desired. For engineering data, get Publication No. 225-1; for "The Story of Syncretized Air," Publication No. 231.

Nesbitt Series B Thermovent

For heating and ventilating auditoriums, gymnasiums, assembly halls, and similar gathering places. Publication No. 227-1.

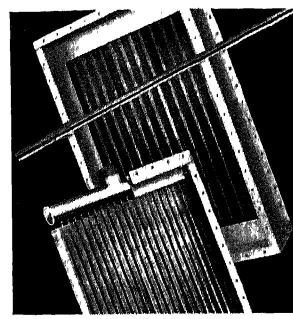
NESBITT COOLING SURFACE

Series W (Water) Surface with exclusive drain feature



Uncased Surface Showing Drain Header

For air cooling and cooling and dehumidifying (with cold water) or air heating (with hot water). Constructed of copper tubes and plate-type aluminum fins. Available in either continuous or cleanable tube type, in single sections having one to eight rows of tubes deep, in three fin spacings, in eleven fin widths, and up to sixteen finned tube lengths. Sturdy galvanized casings. For particulars and engineering data send for Publication No. 233.



NESBITT HEATING SURFACES

Nesbitt Heating Surface with Dual Steam-distributing Tubes

Copper tube-and-fin surface for low-pressure applications. Perfectly adapted to close, continuous automatic control with modulating steam valves. Steam-distributing tubes within the condensing tubes carry the steam equally to the full section assuring UNIFORM discharge temperatures even under a throttled steam supply; eliminating temperature stratification; preventing tube freezing without preheaters; giving ideal system results.

Cased or uncased units of many sizes and capacities. Sold (like all other Nesbitt Surface) by leading manufacturers of fan-system apparatus (list upon request). For full particulars and engineering data, send for Publication No. 229-1.

Nesbitt Series H Heating Surface

A lightweight, enduring, highly efficient blast-coil heating surface designed for use with steam pressures up to 200 lb gauge. Well suited to high-pressure as well as low-pressure applications. Seven types, each in eight fin widths and up to sixteen finned lengths—a total of 784 sizes from which to select. Sold by leading manufacturers of fan-system apparatus (list upon request). Send for Publication No. 232 for complete engineering data.

Factory:
NEWARK, N. J.

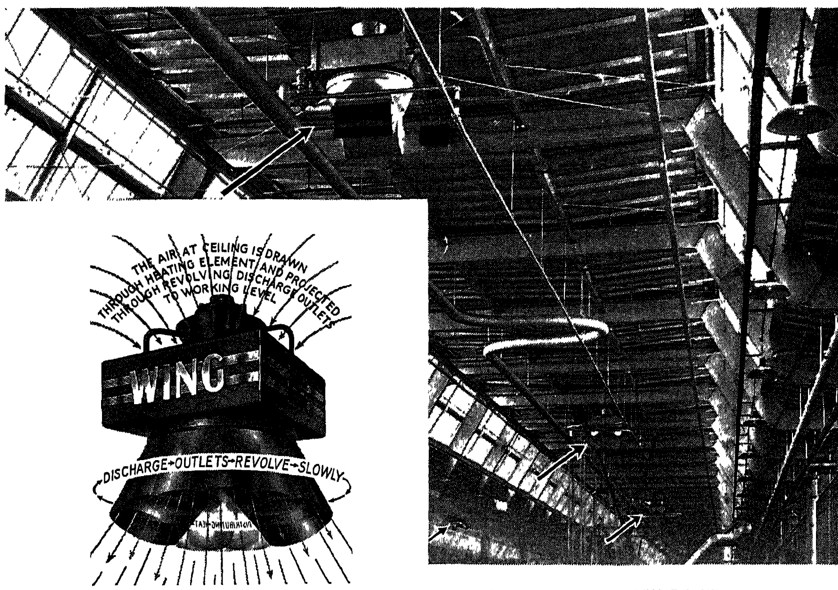
L. J. Wing Mfg. Co.

59 Seventh Avenue, **New York, N. Y.**

Canadian Factory:
MONTREAL

Branch Offices in **Principal Cities**

Wing Featherweight Unit Heaters—Revolving Discharge Type



This innovation in the method of distributing heat produces a sensation in heating comfort never before attained—a sensation of fresh, live, invigorating air.

The fact that the outlets revolve assures uniform and thorough distribution of comfortably warmed air throughout the entire working area, without drafts, hot spots or cold spots.

Such an unprecedented high efficiency in distributing heat is the result of nearly 20 years of constant study by Wing engineers to improve on the Floodlight System of heating originated by WING in 1920. This method projects the heated air vertically downward by means of light-weight, ceiling-suspended unit heaters.

It has needed only this latest refinement of slowly revolving discharge outlets to bring that method to perfection.

The WING Revolving Discharge type supplements the WING line of standard fixed discharge outlets, illustrated and described on the following page.

The latest type of WING Unit Heater—with Revolving Discharge Outlets—is just as great a contribution to the art of industrial heating as was the Ceiling-Suspended Unit Heater, originated by WING in 1921.

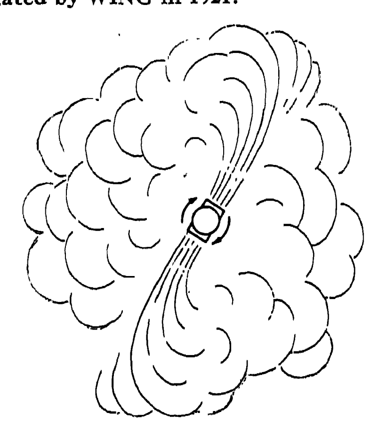
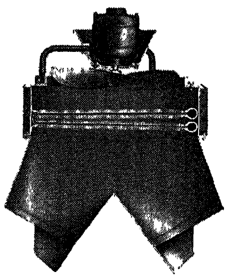


Diagram showing thorough coverage of area to be heated by means of revolving discharge outlets



"*Stationary Discharge*" *Unit Heater*

WING FEATHERWEIGHT UNIT HEATERS

Floodlights of Heat for Economical Heating of Industrial and Commercial Buildings

Low heating cost and uniform diffusion of comfortable warmth are two features in which WING Floodlight Heating excels all other systems.

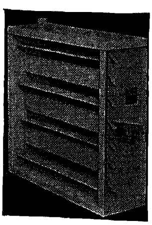
Located near the ceiling or roof, WING heaters prevent accumulation of hot air in the upper spaces with the accompanying costly waste of heat.

They project the air, comfortably warmed, to the working area where the heat spreads to every point. Vertical downward discharge from multiple outlets at proper velocity assures even distribution.

The lightness of WING units permits their suspension directly overhead in any location. Furthermore, one WING unit heater with multiple discharge takes the place of several one-direction heaters at less cost for units, piping, wiring and installation.

There is a type for every condition from low ceiling to installations as high as 55 ft. from the floor.

Type HC heaters should be used wherever roof or ceiling height permits. Type LC has the same capacity as Type HC and is designed for low buildings. *Bulletin II-8.*



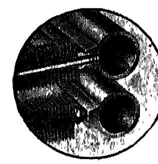
Variable Temperature Heating Section.

Wing Featherfin Heater Sections

For heating or cooling air for any purpose by steam, hot water, cold water or refrigerant. Offer slight resistance to air flow.

Variable Temperature Sections

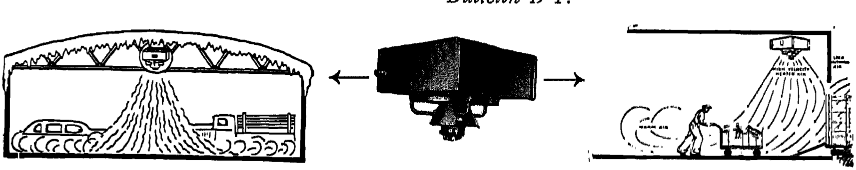
Allow close control of the delivered air temperature without danger of freezing which is likely when control is accomplished by throttling of steam. Invaluable in supplying fresh air for space heating or in process work. *Bulletin IIS-1.*



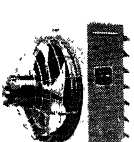
Detail of Wing Featherfin Heating Element showing Compression Union Tube Connection

Wing Garage Heaters—For effective and economical heating of garages. Sometimes cut heating costs in half. *Bulletin G-1.*

Wing Door Heaters—For instantaneously heating inrush of cold air at large doorways of industrial buildings. *Bulletin D-1.*



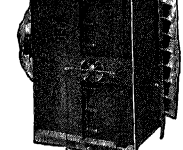
Wing Utility Unit Heaters



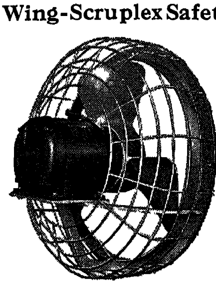
A lightweight suspended unit heater for delivering heated air in one general direction. Has the same powerful fan and rugged heating element as WING Featherweight Unit Heaters. This is the latest refinement of the original horizontal lightweight heater which was developed by WING. *Bulletin U-4C.*

Wing Industrial Fog Eliminators

Eliminate fog, odor and fumes in dyeing, bleaching and finishing plants, creameries, pasteurizing, bottling, canning and packing plants, chemical works, paper mills, steel pickling plants, etc. No ducts are required. *Bulletin FE-12.*



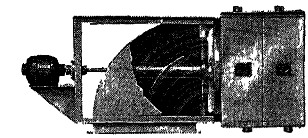
Wing-Scruplex Safety Ventilating Fans



A propeller type fan that will deliver air against static pressure, quietly and efficiently.

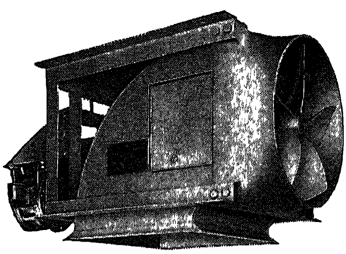
Moves the air forward in straight lines with minimum eddy. Capacities to 100,000 cfm. *Bulletin F-8.*

Wing Featherfin Process Heating Units



For manufacturing processes such as drying, aging, etc., requiring the recirculation of the heated air. Motor or turbine located outside air current. *Bulletin P-2.*

Wing-Scruplex Exhausters



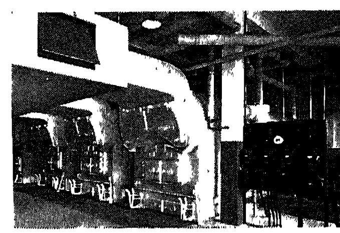
For economically moving air wherever ducts are used. It combines the efficient WING-Scruplex Propeller Fan with a housing which places the motor entirely outside the air duct. Motor and drive remain cool and clean and are easily accessible.

The powerful WING-Scruplex Fan delivers high air volume with low power consumption against any pressures for which duct systems should be designed. V-belt or direct drive.

Light, compact and easy to install. *Bulletin 78-A.*

WING System of Controlled Combustion

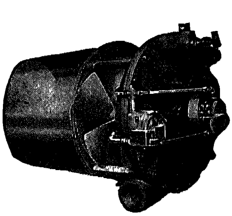
For low pressure heating boilers and small power boilers. Increases capacity and permits use of lowest cost fuel. Includes Type EM Blower equipped with fully enclosed dustproof motor with speed regulating rheostat and automatic control. Eliminates necessity of frequent firing, allowing intervals as great as 24 hours even in zero weather. *Bulletin M-86.*



Installation of Wing System of Controlled Combustion in a large school

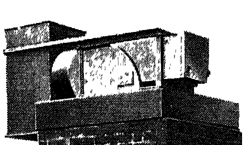
Wing Turbine-Driven Blowers

Applied to hand, stoker, oil or pulverized fuel fired boilers, increase boiler capacity, maintain constant steam pressure and permit complete combustion of low-cost fuels. The exhaust steam, free from oil, can be used for heating or processes. *Bulletin T-97A.*



WING Draft Inducers

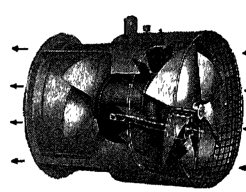
Installed in breeching or flue, or on chimney top; provide positive, exact draft regardless of weather conditions or inadequate chimney or breeching construction. Suitable for coal, oil, or gas-fired boilers; industrial furnaces and kilns. *Bulletin I-10.*



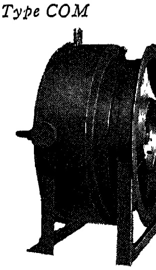
Chimney-Top Installation

WING Motor-Driven Blowers

Type COM for static pressures up to 10 in. W. G. and volumes up to 50,000 cfm. Type EMD for moderate static pressures up to 2½ in. Both blowers have fully-enclosed dustproof constant speed motor and built-in adjustable control vanes. Type COM has double-staged axial flow fan; Type EMD, single stage fan. Extremely compact; discharge can be vertical, horizontal or inclined. *Bulletin CO-3.*



Type COM



Type EMD

The Trane Company

2021 Cameron Avenue, **La Crosse, Wisconsin**

MANUFACTURERS OF HEATING, COOLING AND AIR CONDITIONING EQUIPMENT

Over 80 U.S. Branch Offices

Allentown, Pa., Altoona, Pa., Amarillo, Tex., Appleton, Wis., Altanta, Ga., Aurora, Ill., Baltimore, Md., Birmingham, Ala., Boston, Mass., Brooklyn, N. Y., Buffalo, N. Y., Canton, Ohio, Charleston, W. Va., Chattanooga, Tenn., Chicago, Ill., Cincinnati, Ohio, Clarksburg, W. Va., Clarksville, Tenn., Cleveland, Ohio, Columbus, Ohio, Dallas, Tex., Davenport, Ia., Dayton, Ohio, Denver, Col., Des Moines, Ia., Detroit, Mich., Flint, Mich., Gainesville, Fla., Grand Rapids, Mich., Greensboro, N. C., Greenville, S. C., Harrisburg, Pa., Houston, Tex., Indianapolis, Ind., Jackson, Miss., Kalamazoo, Mich., Kansas City, Mo., Knoxville, Tenn., LaCrosse, Wis., Lake Charles, La., Little Rock, Ark., Livingston, Mont , Los Angeles, Calif., Louisville, Ky., Memphis, Tenn., Miami, Fla., Milwaukee, Wis., Missoula, Mont., New Orleans, La., Newark, N. J., New York, N. Y., Oklahoma City, Okla, Omaha. Neb., Peoria, Ill., Philadelphia, Pa., Phoenix, Ariz., Pittsburgh, Pa., Portland, Me., Portland, Ore., Portsmouth, Ohio, Providence, R. I., Richmond, Va., Roanoke, Va., Rochester, N. Y., Salt Lake City, Utah, San Antonio, Tex., San Francisco, Calif., Scotia, N. Y., Seattle, Wash., Sioux City, Ia., South Bend, Ind., Spokane, Wash., St. Louis, Mo., St. Paul, Minn., Syracuse, N. Y., Toledo, Ohio, Washington, D. C., West Hartford, Conn., West Haven, Conn., White Plains, N. Y., Wilkes-Barre, Pa., Zanesville, Ohio.

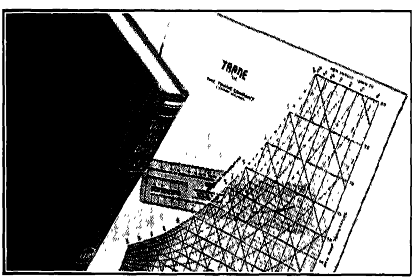
Sales Connections All Over The World

Export Dept.: 75 West St., New York, N. Y.

In Canada: TRANE COMPANY OF CANADA, LTD., Mowat and King Sts., W., Toronto, Ont. (11 Branches)

TRANE EDUCATIONAL MATERIALS

Trane Air Conditioning Manual



Trane offers the engineering profession a comprehensive, straightforward and unbiased textbook covering the fundamentals of air conditioning. Trane engineers have gathered all available material, sifted and analyzed it carefully to produce in one volume the essence of air conditioning practice. The Trane Air Conditioning Manual not only shows how to design every type of air conditioning system, but also clarifies underlying principles enabling both the student and the engineer to reason out their own problems rather than to blindly follow complicated formulas. Price —$5.00.

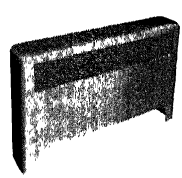
Trane Air Conditioning Ruler and Psychrometric Chart

To solve air conditioning problems with speed and accuracy, The Trane Company has developed the Air Conditioning Ruler and Psychrometric Chart. It eliminates the laborious calculation entailed by outmoded methods—saves two-thirds of your time in figuring air conditioning problems.

TRANE PRODUCTS

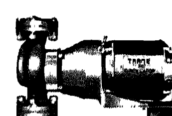
Trane Convectors

In using the Trane Convector it costs no more for the smooth, steady flow of clean, even heat obtained. It costs no more for a lighter, yet sturdier, unit which has superior heat transfer ability. You pay no bonus for the harmonious design of this clean, space saving method of heat diffusion for all steam and hot water heating systems. Trane offers a complete line for both visible and concealed installation.



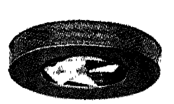
Trane Warm Water Heating

Trane Warm Water Heating Systems bring low cost luxury and comfort to the most modest residence. The Trane Circulator, Trane Flo Valves and Fittings make the hot water heating system function properly and economically. Unique in design and application to achieve lower initial cost of complete heating systems.



Trane Unit Heaters

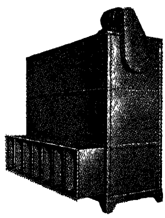
Trane pioneered and introduced a unique heating unit—the Trane Projection Unit Heater—which accomplishes a superior diffusion of heat from advantageous high or low ceiling installation. The broad range of Trane Projection, Propeller and Blower Type Unit Heaters enables Trane to make unbiased recommendation of the unit best suited to do your job.



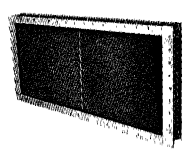
Trane also manufactures a huge line of standard projection, propeller, and blower unit heaters for every conceivable unit heating need in buildings of all types.

Trane Climate Changers

The Trane Climate Changer line affords a unit selection for practically every known air conditioning purpose. Five major types possess a diversity of application possibilities as well as a capacity of range from 250 to 20,000 cfm. Units may be arranged for complete summer and/or winter air conditioning and may be equipped for either steam or hot water heating, and for cooling with water, brine or direct expansion refrigerant. Literally scores of sizes and styles are available for floor, suspended or concealed installation—comfort or process—in all types of buildings everywhere.

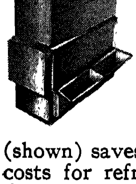


Trane Coils



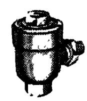
The integral fin-and-tube construction of the Trane Coil provides complete heat transfer for all heating, cooling, drying, and air conditioning services. Trane has several thousand different styles and types of coils to meet all requirements. Types include: high and low pressure steam coils; hot and cold water coils; blast coils; drying coils; direct expansion coils; coils of special materials for special gases or liquids; easy-to-clean coils for water containing foreign matter; coils for installation in units, in ductwork, or for separate service.

Trane Cooling Equipment



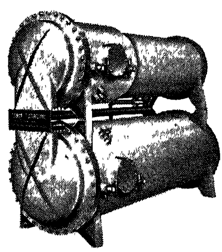
Trane manufactures a complete line of Evaporative Condensers, Evaporative Coolers, Product Coolers, Brine Spray Units, Comfort Coolers, Railroad and Bus Air Conditioners, and Radio Tube Coolers. The Trane Evaporative Condenser (shown) saves up to 90 per cent in water costs for refrigerant condensing. Trane Cooling Equipment is available in the correct size or type to meet any requirement.

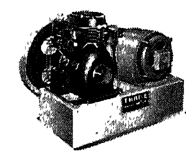
Trane Heating Specialties



There are over fifty Valves, Traps, Vents, Strainers and allied specialties in the Trane Heating Specialty line. Designed to afford protection and accurate control to the steam, vapor, or vacuum heating system, Trane Specialties are expertly fabricated of select materials to insure top performance and create definite fuel savings. Trane Heating Specialties meet the rigid specifications of the United States Navy.

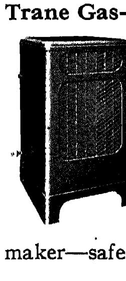
Trane Refrigeration Units





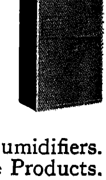
The Trane Turbo-Vacuum Compressor is a hermetically sealed, centrifugal type water chiller which operates under low pressure. It is a complete "equipment room" in one compact package. Only two moving parts and 25 per cent lighter than comparable types. Trane also manufactures a complete line of Reciprocating Compressors available in sizes from 3 to 50 tons capacity. Built for long term performance.

Trane Gas-Fired Equipment



Trane manufacturers a complete line of gas-fired equipment, including Gas Space Heaters, Unit Heaters, and Horizontal and Vertical Gas-Winter Air Conditioners. These units embody a light-weight heat generator and heat exchanger—speedy heat maker—safe and automatic.

Other Trane Equipment



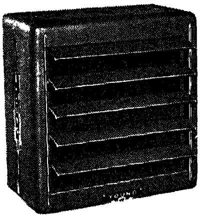
Trane's complete line also includes handsome, sturdy Self-Contained Air Conditioners in 3, 5, 7½, 10 and 15-ton capacities. Also, Condensation, Circulating and Booster Pumps, Temperature Control Valves, Air Washers, Spray Nozzles, Fans, Unit Ventilators, Humidifying Conditioners and Dehumidifiers. Bulletins available on all Trane Products.

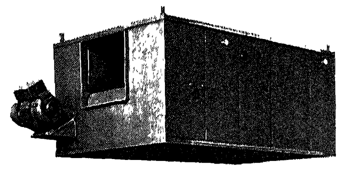
YOUNG RADIATOR Company

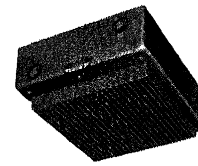
Offices in all Principal Cities

Racine, Wis.

Reg. U.S. Pat. Off.


Model "SH" Unit Heaters


Air Conditioning Units


VERTIFLOW Unit Heaters

Young heating and cooling products are available in many sizes and capacities. Complete information on any product—capacities, dimensions and installation drawings—may be secured by requesting catalogs.

Air Conditioning Units—For heating, cooling, humidifying, dehumidifying, cleaning and circulating air are made in eight sizes for home or industrial installations.

VERTIFLOW Unit Heaters Type "VH"—An industrial type unit heater designed for ceiling suspension. Twelve models of capacities from 30,000 to 480,000 Btu per hour.

Unit Heaters Models "SH" and "TH"—A complete line of single and twin fan suspended units, 32 sizes with capacities from 18,000 to 658,000 Btu per hour.

"FH" Unit Heaters—Ten models with capacities from 120,000 to 800,000 Btu per hour.

STREAMAIRE Convectors—Designed for one pipe, two pipe steam or hot water. Six types of cabinets are available.

Commercial Units—A compact heating or cooling surface.

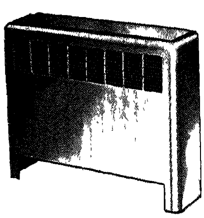
Cooling Coils—Type "W" with continuous tubes, Type "K" with removable headers, for use with water or brine.

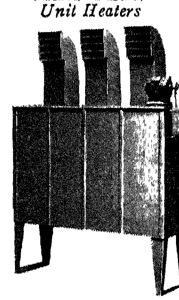
Blast Units—An encased heating surface used in forced air heating or cooling systems.

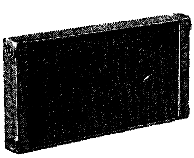
Ductless Air Conditioning Units—A floor unit of high capacity operating with steam or hot water systems.

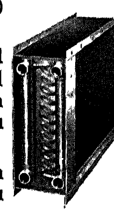
"FC" Heating Units—Floor units provided with two fans—designed to operate on steam or hot water heating systems.

Evaporators—Designed for mechanical refrigeration systems using Freon or methyl cloride.

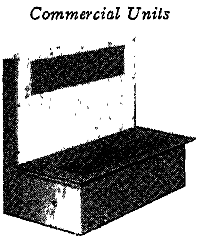

STREAMAIRE Convectors

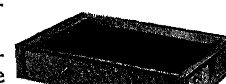

Model "FH" Unit Heaters

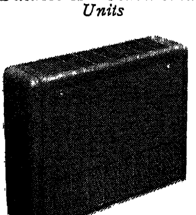

Commercial Units

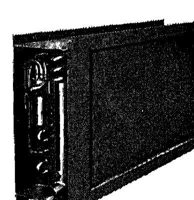

Type "W" Cooling Coils


Type "K" Cooling Coils


Ductless Air Conditioning Units


Blast Units


Type "FC" Heating Units


Evaporators

AIR SYSTEM EQUIPMENT

•

Air systems for heating, cooling and ventilating services are produced by grouping various machines and accessories, each performing a function in the complete cycle of the desired operation. The essential parts and accessories described by the manufacturers are contained in the following groups:

AIR FILTERS AND CLEANERS (p. 862-877)

Mechanical and electrical methods of filtering, also air washing and purifying apparatus and their applications.

Technical data on this subject will be found in Chapter 28.

HUMIDIFYING UNITS (p. 878-882)

For supplying moisture to air and controlling its volume as desired for industrial and commercial uses, or for comfort requirements.

Technical data is contained in Chapter 23.

COOLING TOWERS AND SPRAY EQUIPMENT (p. 879-882)

For cooling and reclaiming water used in industrial processes and air conditioning.

Technical data will be found in Chapter 26.

HEAT TRANSFER SURFACES (p. 883-886)

As parts of heating and cooling units, and for separate use in industrial and commercial heating and cooling systems.

Technical data is contained in Chapter 25.

CONDENSING UNITS AND REFRIGERATING MACHINERY (p. 887-898)

For refrigerating processes and for cooling purposes in industrial, commercial and comfort air conditioning service.

Technical data will be found in Chapter 23.

FANS AND BLOWERS (p. 899-914)

For use as separate air circulating equipment, or as parts of heating and air conditioning units.

Technical data is contained in Chapters 22 and 29.

MOTORS (p. 915-917)

Used in conjunction with blowers, fans, stokers, oil burners and other heating, cooling and air conditioning apparatus.

Technical data on motors will be found in Chapter 35.

REGISTERS AND GRILLES (p. 918-932)

Air diffusion equipment for use with heating, ventilating and air conditioning systems.

Technical data relating to this equipment is contained in Chapters 30 and 31.

SHEET METAL AND TUBULAR PRODUCTS (p. 933-935)

Sheets for air ducts and enclosures; pipes for gas, refrigerants, steam, water, etc.

Technical data on pipe and piping is contained in Chapters 14 and 17.

Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

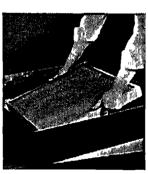
The Air-Maze Corporation

5202 Harvard Avenue, **Cleveland, Ohio**

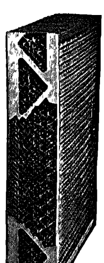
Resistance—For 2 in. thick panels the resistance varies from 0.089 in. to 0.10 in. H_2O when handling 2 cfm per square inch of filter area (288 fpm velocity); and for 4 in. thick filters the resistance varies from 0.121 in. to 0.140 in. H_2O at 2 cfm per square inch (288 fpm velocity); the variation being in accordance with the different types of filter media construction available. To obtain specific restriction data write for graph RE-2A.

Construction—AIR-MAZE filters are of patented construction consisting of a maze of alternately placed and exactly spaced crimped galvanized wire screens of selected meshes; these are arranged with precision so as to create graduated and progressive density, and to positively embody the baffle impingement principle. The filter element is enclosed in a heavy gauge metalescent enameled steel frame having an open end to simplify servicing.

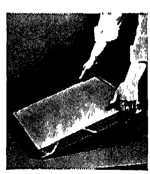
EASY TO CLEAN AND CHARGE



Wash out filtered matter in a pan of hot water or under a stream of hot water.



Cut-away View



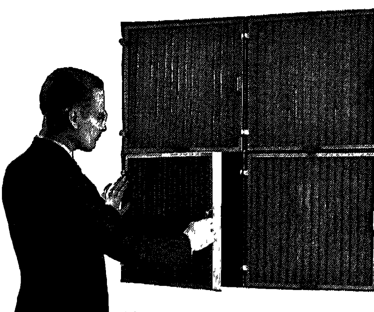
From a flat surface raise one end and let it drop sharply several times. This facilitates drainage.

After cleaning and also after charging, set panel on edge, with open end down, to drain.

Cleaning—Simply tap panel a few times on a hard surface to remove heavy accumulations and then wash under a stream of hot water or in a pan of hot water. Steam also cleans the panels quickly and effectively. Be sure filter is dry before charging.

Charging—(For general applications) Spray both front and back of panel with just enough oil to coat the wires. Any inexpensive oil of S.A.E. 40 or 50 viscosity is suitable. An ordinary insect spray gun will do the work splendidly. Or, if desired, panel may be immersed in oil and then thoroughly drained.

AIR-MAZE INSTALLATION FRAMES



Air-Maze Panel holding frames assure efficient, attractive installations.

AIR-MAZE panel holding frames are constructed of metalescent enameled heavy gage steel having ¾ inch flanged back edge. A thick felt lining on inside of flange insures against air leakage when panels are in place. One frame may be used alone in single panel installations, or a group of frames may be supplied, fixed together; thus a large bank of filter panels may be provided. Every frame section is fitted with snap catches as standard equipment; a lift handle is installed on each panel.

In determining frame sizes, ⅝ inch is allowed over the EXACT width, and ⅝ inch over the EXACT height dimensions of the panels. These dimensions include frame edge, clearance and felt edge seals.

Specify AIR-MAZE—for all air filter installations and you will be assured of *efficient, economical* **performance.** Write for specification bulletin CCC-69.

Engineering Service Available—The Air-Maze Engineering Department will gladly offer installation suggestions for special air filter applications.

Other AIR-MAZE Products—In addition to the panel types, Air-Maze Corporation also manufactures a complete line of circular shaped air filters for use in various Railroad, Industrial and Automotive applications.

Literature Available—Catalog GPC-740 describing industrial types "A," "B," Greastop, and Kleenflo panel filters. Catalog describing Air-Maze Oil Bath type, Multimaze and Unimaze filters for internal combustion engine, air compressor and blower applications.

AMERICAN AIR FILTER COMPANY INC.

673 Central Avenue, **Louisville, Ky.**

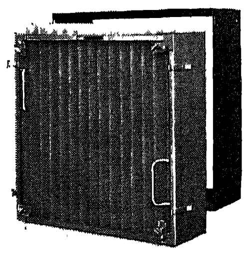
Representatives in Principal Cities

Dust Engineering—Dust Engineering is that branch of applied science which deals with the origin, nature and characteristics of the small solid air-borne particles called "dust," and the development of methods, processes and apparatus for its control or elimination.

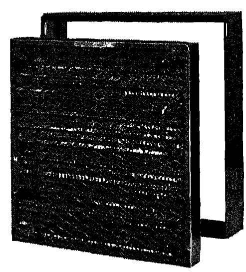
The American Air Filter Company, Inc., has had an important part in advancing the science of Dust Engineering. The efforts of its Research and Engineering Staff for the past twelve years have been devoted exclusively to the study of dust problems and the development of a complete line of air cleaning equipment for modern air conditioning, building ventilation and the control of process dust in industry.

American Air Filter products, therefore, not only embody the knowledge accumulated from years of constant research and the experience gained from designing, building and applying thousands of air filters, but are backed by ample technical and financial resources to insure their outstanding position in the Dust Engineering field.

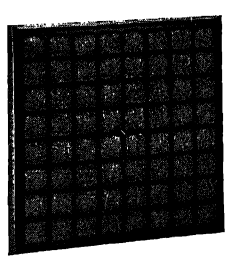
Products—American Air Filters are available for every condition, with operating characteristics and efficiencies to suit specific problems. In general, there are two distinct types based upon the "viscous film" and "dry mat" principles. Each type is made in several styles which differ in method of operation, servicing, space required and initial cost to meet the various conditions encountered in air cleaning problems. A discussion of various filter types will be found in the Technical Data Section under "Air Cleaners."



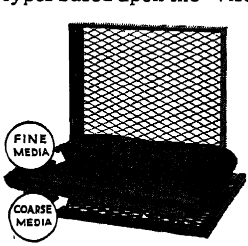
Airmat Type PL-24 Filter



M/W 2 Filter



Throway Air Filter



Renu-Vent Filter

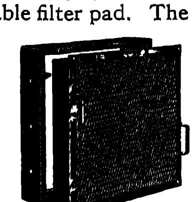
Air filters are generally used for the removal of dust, dirt, bacteria and other foreign matter from the air and are applied to general ventilation, modern air conditioning, process dust control; for air compressors and Diesel Engines; mill motors, turbo-generators and other electrical applications; and for air or gas under pressure to remove entrained oil, moisture and dirt.

Air Filters In Air Conditioning—Filtered air is today recognized as essential in modern air conditioning. There are other important factors which contribute to our comfort such as temperature, air movement and humidity, but science today emphasizes the prime necessity of pure air for health and efficiency.

Air cleaners have, of course, always been considered an integral part of large central systems. These are usually of the fully automatic type such as the Multi-Panel filter, illustrated in the accompanying photograph.

There are now available to manufacturers of unit air conditioners moderate priced unit filters such as the Renu filter, the Throway filter, and other types of filters illustrated on this page.

The Renu filter is an entirely new departure in air filter construction. It consists of a permanent metal frame provided with a removable cover and renewable filter pad. The cover



Standard Viscous Unit Filter

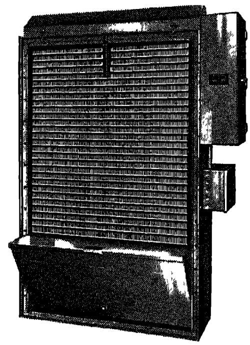
is easily removed without the use of tools, and filter pad can be lifted out and replaced with a new one at very small expense.

The Throway filter, as the name implies, is designed to be discarded after it has served its maximum period of usefulness and replaced with a new filter unit. The Filter pad is enclosed in a perforated cardboard container which makes it possible to readily dispose of the dirty filter by burning it.

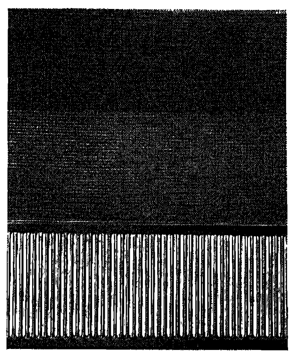
There is probably no single item which costs as little and may mean as much in the design of an air conditioner as air filtration. These units are furnished in any dimensions or shapes desired—usually in units handling 400 cfm and from 2 in. to 4 in. thick. They are usually made in the following sizes—20 x 20 in., 16 x 25 in. and 16 x 20 in. High cleaning efficiencies can be secured, with a resistance to air flow ranging from 1/16 in. to 3/8 in. water gauge.

Automatic Self-Cleaning Air Filters—The American line of automatic air filters is among the most complete ever offered. Proved in principle and performance by years of actual service.

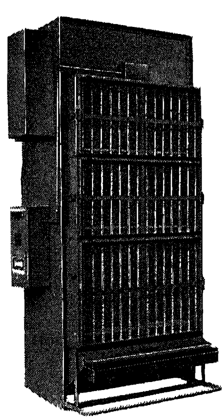
Electro-Matic Air Filter—Incorporates electrical precipitation as an integral function of an automatic self-cleaning viscous filter to obtain a higher over-all efficiency in dust removal. Its higher efficiency as an air cleaning unit, is due principally to the collection of the finer dust particles and smoke, by electrical precipitation. In combination, these two methods of cleaning air not only give the highest efficiency in dust removal but offer operating advantages found only in the automatic self-cleaning filter.



Armored Multi-Panel Automatic



Section of Multi-Panel filter curtain showing unique construction of new Armored Panel. Dark portion of screen is bakelite-fibre coated. The bright uncoated screen lies immediately behind the Armored section of the preceeding panel and provides the second or intermediate stage of air cleaning. The Armored section is at the bottom of the panel.



Electro-Matic Air Filter

Armored Multi-Panel Filter—Introduces an entirely new and unique panel construction to further improve the already outstanding performance of the famous Multi-panel air filter. The new "armored" panel with its three stages of air cleaning maintains the present high efficiency and normal operating and resistance of the Multi-panel filter and offers the added advantage of handling excessive lint concentrations or heavy dust loads without clogging.

Standard Viscous Unit—The American Unit Air Filter incorporates the time tested unit principle of construction. Each unit consists of a standard steel frame and interchangeable cell equipped with automatic latches to facilitate removal for cleaning and recharging.

Airmat Filter Dry Type—The filtering media in this type is the Airmat sheet, a dry filter mat composed of thin sheets of gauzy, cellulose tissue. The Airmat sheets are supported in screen pockets mounted in a unit frame of box-like construction. These unit frames can be set up to meet any capacity requirement or space condition. Airmat sheets are renewable—their life depending on dust conditions and hours of service.

Airmat filters are used both for comfort and industrial air conditioning. In the latter field they are particularly well adapted for the recovery of valuable dusts and for abating the dust nuisance prevalent in so many industrial plants. They are available in two types, the PL-24 as illustrated and the Well Pocket type unit.

Our standard data books and catalogues are in most engineering files or libraries. We will be glad to furnish complete data to engineers or manufacturers.

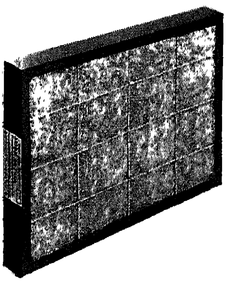
Blocksom and Company

Michigan City, Ind.

AXIOM **PARATEX**

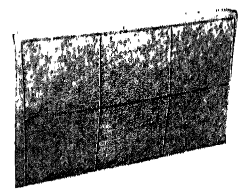
KEEPING PACE WITH MODERN AIR CONDITIONING

The Axiom and the Paratex Air Filters Offer All Necessary Advantages



The Axiom Air Filter
A throw-away type

The Following Facts Prove This Statement



The Paratex Air Filter
A permanent type

CHECK THESE FEATURES

Light Weight (Av. 2 lbs)
Easy to Change
Flameproofed
Treated with Deodorizer
Repels Bacteria
Large Dust Capacity
Low Initial Cost
Low Upkeep
95.6% Efficient

Composed of three types of tightly curled cactus fibre on intake side, hair on exit side, thus impinging all kinds of dirt, dust, pollen and bacteria.

The Axiom Air Filter is really flameproofed, receiving two flame proofing treatments. In a test made by national authorities of the six leading air filters, the Axiom Air Filter led by a wide margin, having an overall efficiency of 95.6 per cent. This is very important as it insures the top performance of the heating and air conditioning equipment.

With all these extra features it is still lower in cost.

Light Weight (Av. 1½ lbs)
Easy to Install
Treated with Deodorizer
Repels Bacteria
Large Dust Capacity
Easy to Clean
High Efficiency
Permanent
Low Cost

Composed of tightly curled hair. Dipped in Latex rubber to serve as a binder.

Cannot dent, chip, rust; available in any size, shape, thickness or density.

Clean by simply rinsing in water or spraying with a hose.

It is practically impossible to damage the Paratex Air Filter even with careless handling.

Does not need the expense or mess of a special spraying of oil at any time. The fine tightly curled hair does an efficient, thorough job of cleaning the air.

Due to resilient construction it offers perfect seal at all times.

Very reasonable in price.

You owe it to the performance of your equipment to investigate these two air filters.

A great many of the nation's leading manufacturers are users of
AXIOM AND PARATEX AIR FILTERS.

Over fifteen year's experience in the air conditioning field.
Let our Research and Engineering Department serve you.

Coppus Engineering Corporation

339 Park Avenue, **Worcester Mass.**

MANUFACTURERS OF AIR FILTERS, STEAM TURBINES, GAS BURNERS, FORCED DRAFT BLOWERS, COOLING FANS

"COPPUS AIR FILTERS PASS CLEAN AIR"

The Coppus Unit Air Filter (patent No. 2050508 and other patents pending) is of the dry type using as filter material all-wool felt. It consists of a distender frame (*C*, Fig. 2), a filter "glove" (*E*, Figs. 1 and 2) and a retainer grid (*B*, Fig. 1). The edges of the retainer grid form a reenforced sheet metal box (*A*, Fig. 1) for protection of the filter element.

The edges of the filter glove are reenforced on all four sides assuring an air tight seal against by-passing of dirty air. By tightening the wing studs which hold the distender frame and the retainer grid together, the filter glove is stretched and held tautly inside of the filter box, giving the pockets a tapered shape so essential for an even air flow.

This design has the advantage of providing an effective filter area entirely unobstructed by wire or screen supports. Cut, Fig. 3 shows the tapered filter pockets on the clean air side. The filter glove can be readily replaced without removing the unit filter from the installation. No auxiliary frames for insertion of the filter cells are required as the completely assembled unit filters can be bolted together to a filter bank of any desired size.

All metallic parts are rust-proofed and Duco Painted.

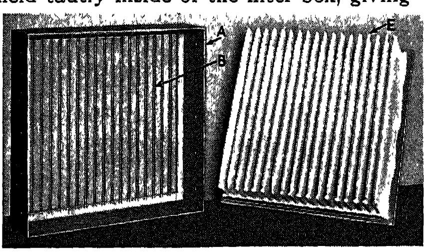


Fig. 1

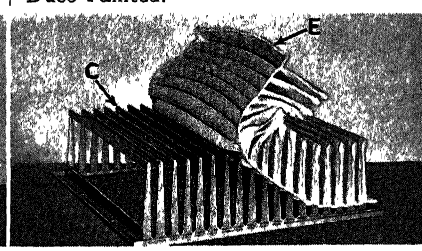


Fig. 2

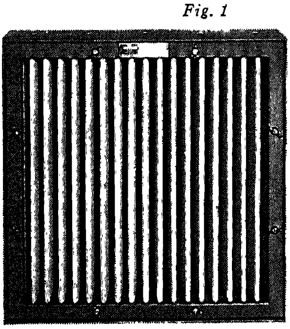


Fig. 3

Specifications

Normal Rating: 800 cfm.
Resistance when clean: 0.2 in. W.G.
Dust Arrestance (cleaning efficiency): 99.61 per cent (Tested in accordance with A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work).
Dimensions: 20 by 20 in. by 5¾ in.
Weight per unit: 25 lb.

ANOTHER COPPUS BLUE RIBBON PRODUCT

Outstanding Advantages

1. It has an exceptionally high dust arrestance.
2. It maintains a high dust arrestance even under diverse conditions of neglect.
3. Its operation is not impaired by atmospheric conditions.
4. It is a Medium Air Resistance Type (Class C) according to the A.S.H.V.E. Code for Air Cleaning Devices.
5. It is easily and quickly cleaned without removing the filter element.
6. Its cost of upkeep is very low because the *permanent* filter element is reconditioned periodically with a vacuum cleaner.
7. It combines scientific knowledge and practical engineering methods with highest quality of material and workmanship.

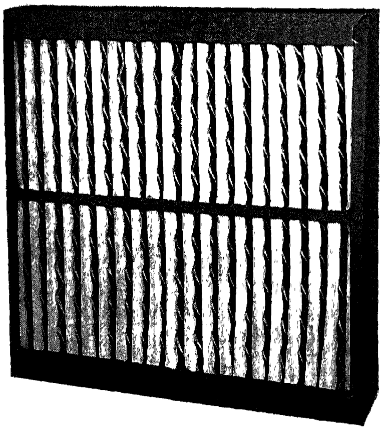
Write for Complete Bulletins


Cleaning Filter Elements with Portable Vacuum Cleaner

Davies Air Filter Corp.

396 Fourth Avenue, **New York, N. Y.**

Air Conditioning, Process, Building, Industrial Filters



AIRPLEX RENEWABLE FILTER

Airplex filter medium is cotton fibre, specially processed and lightly glazed. Each filter contains 30 sq ft of filter medium and gives 500 to 1000 hours active service. Functions efficiently in temperatures below freezing and up to 200 F. Not affected by temperature or humidity—will not disintegrate. Filters can be cleaned several times before they are discarded.

Each filter is a complete cartridge—replacement can be made quickly, hence is not neglected.

Standard Sizes Airplex Filter

Process, Industrial, Building	20 x 20 x 4 in. 25 x 20 x 4 in. 24 x 24 x 4 in.
Home Air Conditioning Units	20 x 20 x 2 in. 25 x 16 x 2 in.

PERMAT WASHABLE FILTER

Filter medium of fine spun hair glass closely packed and secured between two sheets of galvanized wire cloth;these long flexible glass fibres do not break and cannot be drawn into the air stream.

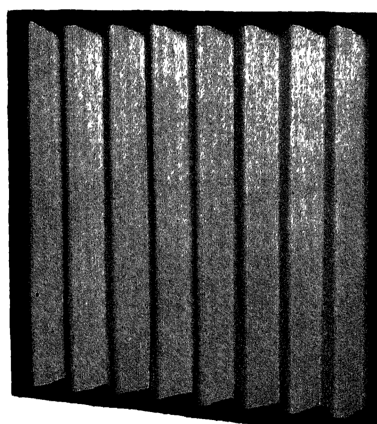
Filter element supported in a steel frame, rust proofed or galvanized as required.

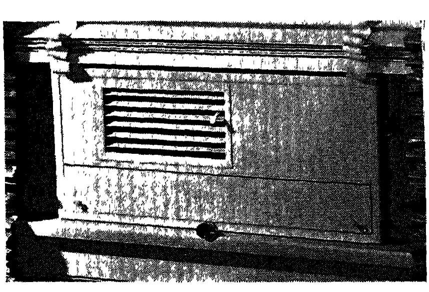
Glass wool, being chemically inert, is not attacked by gases or liquids, will not rust or disintegrate, will last many years. Water, hot or cold, with or without grease solvents, used for cleaning, depending on type of air pollution.

Standard Sizes—Permat Filter

Frame Size	Filtering Surface	Capacity cfm	Resistance
20 x 20 x 2 in.	800	800	.1″ W.G.
25 x 16 x 2 in.	800	800	.1″ W.G.

Other sizes are available on request.





"FILTERAIRE"

For installation in windows—any type of construction. Provides controlled ventilation, pure clean air, and eliminates disturbing noises. Attractive appearance—harmonizes with building finish and furniture.

Model B-5 capacity—500 cu ft of fresh filtered air per minute.

Model B-3 capacity 300 cfm.

Standard colors ivory and brown. Other colors are available.

W. B. CONNOR ENGINEERING CORP.

Dorex Division

114 East 32nd Street, **New York, N. Y.**

Manufacturers of a Complete Line of Odor Removal Equipment

Representatives in All Principal Cities **Canada: Arthur S. Leitch Co., Ltd., Toronto, Ont.**

Dorex Odor Adsorbers are the application of the gas mask principle to commercial and industrial air purification. They employ the same powerful adsorption medium—specially processed, highly activated, granular coconut shell carbon, in multiple, removable, perforated canisters or tubes. Exposed to the air stream they remove from the air and retain all odorous gaseous impurities. Upon saturation Dorex carbon may be economically reactivated.

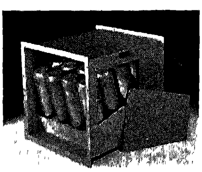
"WORKS LIKE A GAS MASK"

DOREX

ODOR ADSORBERS

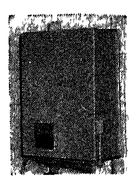
Dorex Odor Adsorbers make possible substantial reductions in both initial and operating costs in ventilating, heating and cooling systems because more air can be recirculated and less fresh air brought in without sacrificing air purity.

Dorex Odor Adsorbers are simple, compact, and easily installed. Dorex engineers in principle cities will gladly study any problem and make definite recommendations without charge. Complete engineering catalogs available on request.



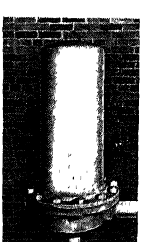
Type H

Type H—Consists of multiple, removable, carbon-filled, perforated, metal canisters so arranged that all air will pass uniformly through carbon beds. Suitable for any air volume. Flexible design permits adaptation to varied space requirements. For air intakes and exhausts and recirculation systems. Obtainable with corrosion resistant canisters.



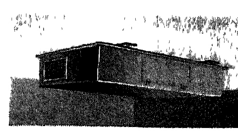
Type W

Type W—Wall cabinet type for individual confined spaces. Completely self-contained, sturdy, galvanized iron casing. Ideal for doctors' offices, lavatories, hospitals, restaurants, cafes, etc. In capacities of 150 and 300 CFM. Complete with motor, switch, fan and dust filter.



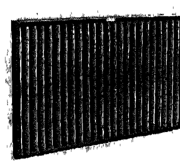
Type PL

Type PL—For extracting oil vapors, fermentation odors and other gaseous impurities from compressed air. Prevents contamination in processes using compressed air for pneumatic agitation of foods, medicines and cosmetics. Simply and easily installed. Available for compressed air capacities from 25 CFM to 350 CFM.



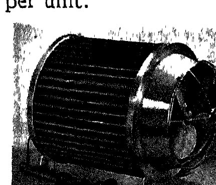
Type CL

Type CL—Identical to the Type W but designed for larger areas and ceiling mounting. Available in capacities from 500 to 1500 CFM. Finished in cream enamel.



Type G

Type G—For duct or casing installation or simple attachment to standard dust filters or inlet or outlet grilles. Especially for recirculation systems to continuously remove odors from reused air. Consists of one, two or three staggered rows of carbon-filled tubes designated as G-1, G-2, and G-3, respectively. Five (5) standard stock sizes: 20 in. x $20\frac{1}{8}$ in., 20 in. x $16\frac{1}{2}$ in., 24 in. x $23\frac{3}{4}$ in., 25 in. x $16\frac{1}{2}$ in., 25 in. x 20 in., for capacities of 1000 to 1500 CFM per unit.



Type SQ

Type SQ—The **Odorsorber.** For ordinary living odors in smaller areas, offices, homes, etc.—has a multitude of practical uses. Black enamel tubes in polished aluminum casing. Available in four sizes. Complete with motor and fan.

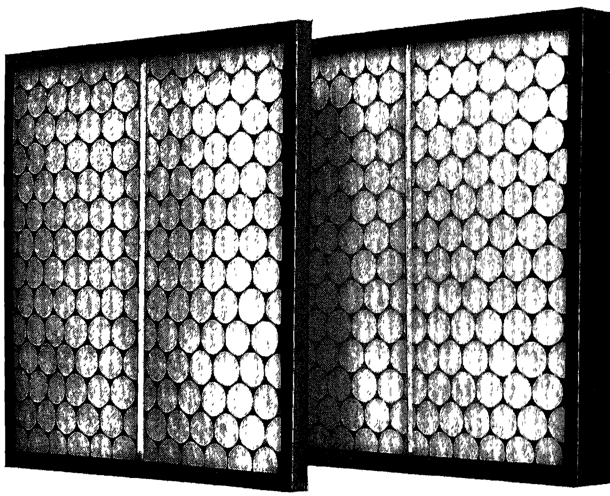
Owens-Corning Fiberglas Corporation

Toledo, Ohio

AIR FILTERS FOR USE IN RESIDENTIAL, COMMERCIAL *and* INDUSTRIAL HEATING, VENTILATING *and* AIR-CONDITIONING SYSTEMS

FIBERGLAS* DUSTOP* AIR FILTERS

*Trademark Reg. U. S. Pat. Off.



Dust-Stop No. 1 Filter, 1 in. thick, at left. No. 2 Filter, 2 in. thick, shown at right. No. 2 Filters are designed for application where their greater dust-holding capacity permits longer intervals between replacements.

The Fiberglas Dust-Stop Air Filter consists of a series of non-combustible Fiberglas Mats, progressively packed—coarse glass fibers of lesser density at the intake and fine glass fibers of greater density at the discharge face—between stamped metal grilles bound with a fiberboard frame.

Mats are coated with non-evaporating, incombustible adhesive having extraordinary wetting power, will retain viscosity under operating temperatures ranging from 15 F below to 300 F above zero, will not flow or charge the air with adhesive.

Engineered to Provide High Efficiency, Fiberglas Dust-Stop Air Filters also provide low cost of installation and maintenance.

Efficiency—97 per cent (Tested according to A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work).

Available in Two Standard Types—Fiberglas Dust-Stop No. 1 (1 in. thick) is designed for greatest operating economy in commercial and industrial applications. No. 2 (2 in. thick) is recommended for use in unsupervised installations. It permits longer intervals between replacements. Both may be used in domestic applications.

Manufacturers' Acceptance

Dust-Stop Air Filters are widely accepted by manufacturers of air conditioning equipment.

Engineering Service—Owens-Corning Fiberglas Corporation maintains offices in several metropolitan centers where representatives, qualified to assist in the planning of filter installations, are available for consultation.

Literature—Data sheets on all standard Fiberglas products and applications will be furnished to engineers and manufacturers on request.

Standard Sizes for Equipment*

Standard Sizes (Nominal)	Ratings		Average Resistance Inches Water Gauge Clean
	Cfm	Fpm	
20″ x 25″ x 1″	1000	300	.065
20″ x 20″ x 1″	800	300	.065
16″ x 25″ x 1″	800	300	.065
16″ x 20″ x 1″	640	300	.065
20″ x 25″ x 2″	1000	300	.125
20″ x 20″ x 2″	800	300	.125
16″ x 25″ x 2″	800	300	.125
16″ x 20″ x 2″	640	300	.125

*Other standard and any special sizes available.

Owens-Corning Fiberglas Corporation

Toledo, Ohio

AIR FILTER FRAMES FOR HEATING, VENTILATING *and* AIR CONDITIONING SYSTEMS

FIBERGLAS* DUSTOP* FILTER FRAMES

*Trademark Reg. U. S. Pat. Off.

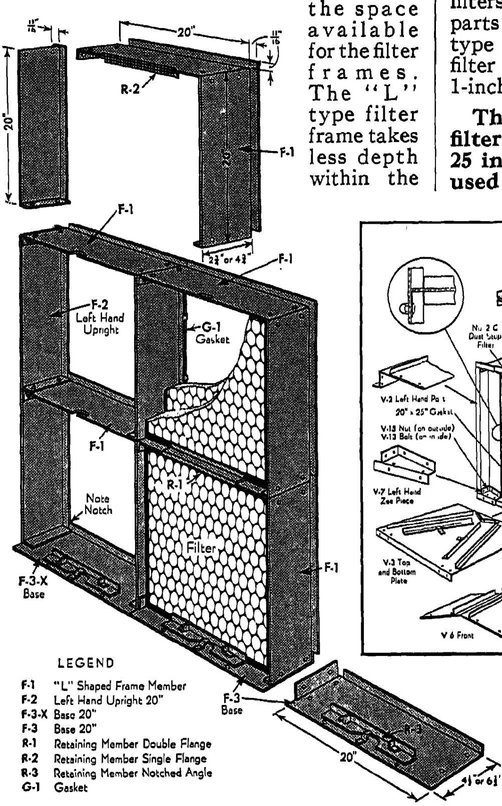
Fiberglas Dust-Stop "L" and "V" Filter Frame Assemblies are installed by engineers of commercial and industrial heating, ventilating and air conditioning. Frame members of heavy steel are assembled vertically in combinations to satisfy any CFM and space requirement.

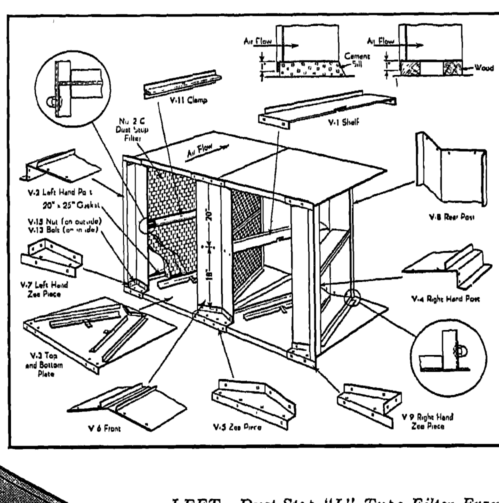
Both types of frames are designed for the convenient and correct handling of Dust-Stop filters. They meet all Fire Underwriters' and local Fire Ordinance requirements, as well as the requirements of Federal Specifications for filter frames.

The choice between the "L" type and "V" type frames is determined wholly by the space available for the filter frames. The "L" type filter frame takes less depth within the duct or plenum chamber but requires a larger face area for the same CFM capacity. The "V" type frame requires a face area approximately the same as the cross-sectional area of a duct which will handle the volume of air for which the filters are rated.

Two Depths of "L" Frames—The "L" frame, two filters deep, is designed to hold two Dust-Stop No. 1 filters in each cell. The "L" frame, four filters deep, holds four Dust-Stop No. 1 filters in each cell. The frame that is four filters deep is identical in every way to the frame two filters deep except that the depth of all parts is 2 in. more. When specifying "L" type frames indicate two-filter or four-filter depth. "V" frame is available, four 1-inch filters deep per cell, only.

The "L" frame uses 20 x 20 in. filters only. The "V" frame uses 20 x 25 in. filters only. Filters are always used two or more in series in each cell.





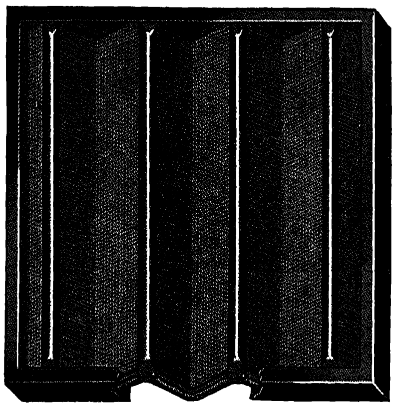
LEFT—Dust-Stop "L" Type Filter Frame.

ABOVE—Dust-Stop "V" Type Filter Frame

H. J. Somers, Incorporated

Factory and General Office

6063 Wabash Avenue **Detroit, Mich.**



All Welded Vee Type

Somers Washable Air Filter

Somers Hair Glass Filters provide everything required in an efficient air-cleaning system. Consider these features: High rating for dust, soot and bacteria separation. Require no adhesive, coating or impregnation. Indestructible in normal service. Minimum Low Pressure Drop. Odorless and non-absorptive. Fireproof; Washable; Do not rot nor disintegrate; Permanent.

Somers Hair Glass Filters consist of a hot galvanized frame holding galvanized wire cloth packed with hair-spun glass strands. The glass strands are flexible, do not break up and cannot be drawn into air stream.

Hair-Glass, being chemically inert, has no facility of absorption; it cannot rust and lasts indefinitely in service. Water either hot or cold may be used to clean it, without impairing its efficiency. These filters eliminate the necessity, the expense and the inconvenience of periodic replacement.

Somers Washable Air Filter—All Welded Vee Type—Stock Sizes (Partial List)

Frame Size Height and Length In.	Frame Depth In.	Filter Surface Sq In.	For Average Dry Filter Installations CFM	Wet Application where water sprays are applied against filter for humidifying CFM
15 1/2 x 24 1/2	3 5/16	1023	1023	511
15 5/8 x 24 5/8	3 1/4	1110	1110	555
16 x 21 1/2	3 1/16	816	816	408
16 x 25	3 1/8	1056	1056	528
16 x 25	3 1/4	1632	1632	816
16 x 25	3 1/4	1344	1344	672
16 x 25	3 1/4	1440	1440	720
16 x 25	3 5/16	864	864	432
16 1/2 x 24 1/2	3 1/8	800	800	400
18 x 18	3 5/16	864	864	432
19 x 20	3 5/16	1482	1482	741
19 1/4 x 19 5/8	3 1/4	1039	1039	519
19 1/4 x 20	3	1039	1039	519
19 3/8 x 19 1/2	3	936	936	468
19 1/2 x 19 1/2	3 1/4	1053	1053	526
19 1/2 x 19 1/2	2	480	480	240
19 1/2 x 19 1/2	3	936	936	468
19 1/2 x 19 1/2	3 1/4	1170	1170	585
20 x 25	3 5/16	1800	1800	900
20 x 30	3 5/16	1800	1800	900
20 x 20	3 1/16	1040	1040	520
20 x 30	3	1560	1560	780
20 x 20	3 1/4	1200	1200	600
20 x 20	2	480	480	240
20 x 20	3	840	840	420
20 x 20	3	960	960	480
20 x 20	3 1/4	1320	1320	660
20 x 25	3 1/4	1560	1560	780
20 3/8 x 20 1/4	3	550	550	275

Other sizes from 9 1/2 x 30 to and inclusive of 31 in. x 23 1/4 also available. Send for complete stock size list.
Frames zinc plated for 100 hour salt water spray test. Refill may be inserted if necessary.
Quotations and further engineering data, including master holding frame drawings will be sent on request.

Staynew Filter Corporation

Air Filters for Every Purpose

6 Leighton Ave. Rochester, N. Y.

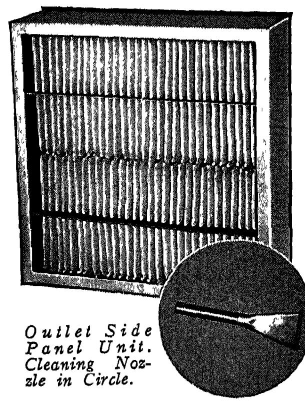
EASY TO CLEAN—LONG LASTING

Cleaning is easily effected by use of any vacuum cleaner with special nozzle. See illustration below.

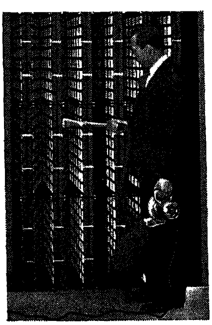
Protectomotors operate from 3 months to a year without cleaning. Panel units average slightly longer wear than Multi-V-Type units—several years at least without replacement.

Panel Units: Consist of Panel Insert and Frame. The Insert is composed of two rows of 60 hollow loops or fins 6 in. deep, formed of rust-resisting embossed wire mesh, supported by a retaining grate of steel or aluminum and similar spacing grate. Each row of fins is covered with a single piece of Feltex Filtering Medium, a felt-like material specially made for the application. Specifications below:

Overall Dimensions (Depth less locking keys)........20 x 20 x 6⅝ in.
Size of Insert........19½ x 19½ x 6 in.
Capacity (average conditions)........800 cfm
Area of Filtering Medium........42 sq ft
Linear velocity of air........19 fpm
Resistance of clean filter to air flow 0.185 in. water gauge.
Total Weight........28 lb

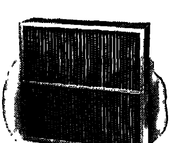


Outlet Side Panel Unit. Cleaning Nozzle in Circle.



Multi-V-Type Filter Showing Ease of Cleaning

Multi-V-Type Units: Filtering medium (closely pressed cotton fibres between two sheets of cotton gauze) is arranged in patented V-shaped pockets in a fibre-board and pressed metal frame. These patented cells can be quickly and inexpensively replaced when worn out. Their arrangement makes possible an active filtering surface of 27 times face area. In certain installations the Multi-V-Type is more desirable than the Panel Unit because its construction fits the space better, or because it is lighter in weight per square foot of filtering area, or for reasons of economy. (Protectovent Window Ventilator, which supplies clean, fresh air to home or office, employs Multi-V-Type inserts). *Complete specifications mailed promptly on request.*



Wire-Klad Filter

Wire-Klad Units: Unique method of construction permits a high efficiency filter at low cost. Fins are reinforced on both sides with screen cloth, producing a rigid, long-wearing, *flame-resisting* filter that may be repeatedly cleaned with vacuum or compressed air, or flushed with water or liquid solvents. Made in 2 in. and 4 in. deep units.

Specifications 2 in. and 4 in. Units

Sizes	Capacity—Wool Felt	Capacity—No. 6460 Cotton	Filtering Area sq ft
20 in. x 20 in.	800 cfm. @ 0.13 in. wg	800 cfm. @ 0.08 in. wg	2 in. 18.5 4 in. 38.5
16 in. x 25 in.	800 cfm. @ 0.12 in. wg	800 cfm. @ 0.075 in. wg	2 in. 18.5 4 in. 38.5
16 in. x 20 in.	600 cfm. @ 0.11 in. wg	600 cfm. @ 0.07 in. wg	2 in. 14.8 4 in. 30.7
20 in. x 25 in.	1000 cfm. @ 0.12 in. wg	1000 cfm. @ 0.08 in. wg	2 in. 23.0 4 in. 48.0

Write for Catalog Mentioning Special Interests

PROTECTOMOTORS ALSO MADE FOR INTERNAL COMBUSTION ENGINES, COMPRESSORS, TURBO-GENERATORS, AIR TRANSMISSION LINES, ETC.

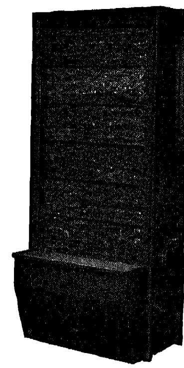
PROTECTOMOTOR AIR FILTERS

Staynew Filter Corporation

Air Filters for Every Purpose

6 Leighton Ave. Rochester, N. Y.

Automatic Filter



PROTECTOMOTOR AUTOMATIC FILTER

(For efficiently and economically filtering large volumes of air for all ventilating purposes)

This latest model Staynew Automatic Filter operates on the principle of dust impingement. Dust is caught by moving filter panels moistened with oil from a reservoir. The panels are automatically timed to operate at pre-determined intervals (approximately 20 seconds each half hour; panel moves 4¾″ each time), depending on the amount of dust to be removed and the air velocity. The filter possesses a number of unusual features which increase efficiency in dust removal and reduce operating costs. Several of these features are fundamental in design and found in no other filter. The result is a rugged, high-efficiency unit with which extremely large volumes of air can be filtered at low cost.

Two Series of Panels

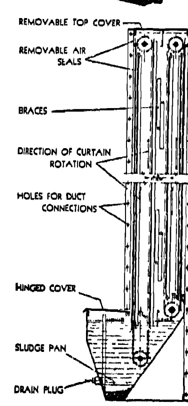
There are *two* series of endless moving filter panels. Each series provides two stages of filtration — four stages in all. The double series of moving filter panels is exclusive with Staynew. Efficiency is twofold.

Counter-clockwise Panel Travel

The panels travel in a counter-clockwise direction. This is an important, select feature. It means that the outlet side of each series of moving panels is the clean side always. The mechanism is actuated by a ⅙ hp motor with a reliable timing device (Telechron).

Sizes and Capacities

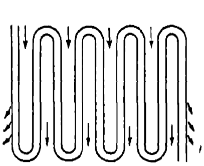
Two standard widths are made, 2 ft 9 in. and 4 ft 3 in., ranging in height from 4 ft to 13 ft by 3 in. steps. Capacities are from 2,025 cfm to 20,200 cfm for single units. Almost unlimited capacities may be secured by bolting together units.



Sectional View

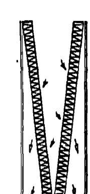
PROTECTOMOTOR DRY TYPE FILTERS

(For removing foreign matter from the air at atmospheric or other pressures, with various types for building ventilation, dust recovery, oxygen chamber and all air cleaning purposes.)



Cross Section Showing Panel Unit Construction

The fin or V-type construction is used in all Protectomotor dry filters. This basic principle permits (1) a large area of filtering medium to occupy the smallest possible space, and (2) the intake currents to move parallel to the filtering surface at low velocity. Protectomotor Dry Filters require no adhesive material to catch dust—*odorless* air is assured. Authorities agree that the positive dry filter is most efficient in stopping the smaller air-borne particles. Protectomotor dry filters actually prevent the passage of bacteria.



Cross Section of 2 Multi-V-Type Cells in V formation

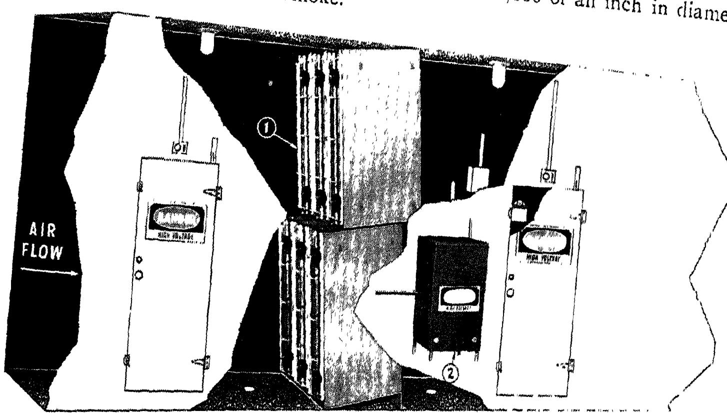
Westinghouse Electric & Manufacturing Co.

Edgewater Park Precipitron Department Cleveland, Ohio

THE PRECIPITRON*

First Commercially Practical Electrostatic Air Cleaner

The Westinghouse PRECIPITRON is the first commercially practical electrostatic method of removing dirt, dust and other air-borne impurities in ventilating and air conditioning systems. The PRECIPITRON—more efficient than mechanical filters—removes microscopic foreign matter as small as 1/250,000 of an inch in diameter—even freeing the air of tobacco smoke.



Typical six-cell PRECIPITRON installation capable of handling up to 4500 cfm (1) cells; (2) Power Pack.

FOR MASS AIR CLEANING JOBS

The PRECIPITRON provides a complete answer to mass air cleaning jobs in all Commercial, Industrial, and Public Buildings using forced ventilation or air conditioning duct systems.

Applications—The PRECIPITRON protects documents, decorations, merchandise and reduces cleaning costs in commercial and public buildings. It protects machines, stocks and production in industrial plants. In all applications it reduces maintenance costs and improves working conditions.

It is used in office buildings, theatres, banks, telephone exchanges, libraries, hospitals, laboratories of all kinds, hotels, department stores, art galleries—in fact, wherever dirt removal is important.

Advantages—More efficient than mechanical filters. Safe. Easily installed. Non-clogging, and non-varying resistance. Easily cleaned. Listed by Underwriters' Laboratories and passed by them on standard flame tests for fire hazard on duct installation.

Sizes—The PRECIPITRON is available, complete for installation, to accommodate from 300 cfm (for a single 18 in. cell) to any desired volume through multiple cell arrangements. Cells come in two sizes—18 in. x 8⅛ in. x 23⅜ in. and 36 in. x 8⅛ in. x 23⅜ in. For a 90 per cent efficiency, the 18 in. and 36 in. cells are rated at 300 and 600 cfm respectively. For 85 per cent efficiency, ratings are 375 and 750 cfm. Two sizes of Power Packs are available. The Type S for installations up to 12 36-in. cells and Type L for 12 to 50 36-in. cells.

Non-Varying Resistance—The PRECIPITRON has no screens to clog—and its resistance is always constant. Once each month accumulated dirt is washed down the sewer with a hose.

Information—Westinghouse will gladly provide complete information about the PRECIPITRON. Address your requests to Section G, Precipitron Department, Westinghouse Electric & Manufacturing Company, Edgewater Park, Cleveland, Ohio.

*Trade-mark Registered in U.S.A.

(*See also Pages 826-827*)

Oakite Products, Inc.

General Offices: 36 E. Thames St., **New York, N. Y.**

OAKITE

MATERIALS . . . METHODS . . . SERVICE

CLEANING

Established 1909

Representatives in all Principal Cities of the United States and Canada

Specialized OAKITE Materials for:

Slime Control in Re-circulating Systems. Cleaning Air Filters, Lube Oil Coolers, Heat Exchange Equipment.	De-Scaling Condensers, Compressors, Jacket Water Coolers, Diesel and Gas Engine Cooling Systems.

Controlling Slime Growths

Control of bacteria and slime growths in re-circulating water supplies is inexpensively established with Oakite Airefiner. A dry, non-volatile, white powder, completely soluble, it prevents formation of slime accumulations and their unpleasant odors. Economical to use . . . one lb to each 300 gallons of water usually recommended.

Prevents Equipment Corrosion

Oakite Airefiner prevents corrosion of eliminators, air washing chambers, spray heads, etc., because it maintains water at a point sufficiently alkaline to counteract the tendency of the water to become acidified. In addition, it gives wash waters greater wetting-out action, thus making dirt removal more complete. Water lines are also kept free of scale.

Free Booklet Gives Details

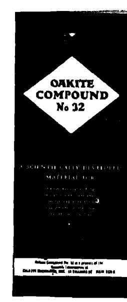
Cleaning Air Filters

For cleaning viscous type filters, hot or cold solutions of recommended Oakite material may be used. Short immersion of filter, followed by rinse, thoroughly removes dust, dirt, soot, lint and pollen without injuring filtering medium or frame metal. Steam cleaning methods also available. Full filtering capacity is economically restored.

FREE 16-page booklet gives details.

De-Scaling Equipment Safely

When scale and rust deposits form in jacket water coolers, ammonia condensers, compressors or other water-cooled mechanical refrigerating or similar equipment . . . heat transfer is reduced, operating efficiency impaired. These insulating deposits may be effectively, safely removed with Oakite Compound No. 32 simply by soaking with or circulating recommended strength of solution. Does not harm base metal. Method is easy, economical. New 20-page booklet gives successful, widely used formulas and directions for this and such other work as de-scaling and cleaning Diesel, gas and gasoline engine cooling systems, lube oil coolers, other heat exchange equipment. A copy is yours FREE.



Free Booklet

Nation-Wide Service

Because Oakite cleaning materials are backed by a binding GUARANTEE and supplied through a nation-wide organization of Service Representatives throughly experienced in their application, users are assured of obtaining the economies and advantages they provide to effectively promote maximum performance of air-conditioning and mechanical refrigerating equipment.

American Moistening Company

ESTABLISHED 1888

ATLANTA, GA.
BOSTON, MASS.

Providence, R. I.

CHARLOTTE, N. C.
GREENVILLE, S. C.

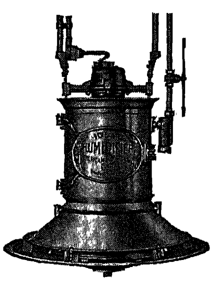
AMCO

UNIT HUMIDIFYING AND AIR CONDITIONING EQUIPMENT

A few of many AMCO products with a Long Record of Dependable Performance

Sectional Humidifiers.
Amtex Humidifiers.
Hand Sprayers.
Mine Sprays.
Fabric and Paper Dampeners.
Mechanical Psychrometers.
Electro Psychrometers.
Sling Psychrometers.
Hygrometers.

The Amco line of devices for the supply, maintenance and control of humidity is complete in its ability to meet any presented problem of applied humidification. Used independently or as an adjunct to Central Station equipment, these devices automatically maintain any required humidity condition in a capable uniform performance.

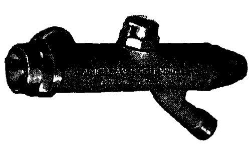


IDEAL HUMIDIFIERS—Senior Type

A high capacity unit for use where conditions require a great amount and good distribution of moisture. Motor driven fan gives wide distribution of atomized spray. Amco heads serve the triple purpose of humidifying, air washing and cooling.

IDEAL HUMIDIFIERS—Junior Type

Similar in construction to Senior Type. Used where medium capacities are required.



AMCO ATOMIZER—No. 4

Quality and quantity of spray are maintained even under adverse conditions because this atomizer is automatically self-cleaning. When the compressed air supply is shut off, either manually or in response to a humidity control, *both* air and water nozzles are thoroughly cleaned.

AMCO HUMIDITY CONTROLS

Compressed Air Operated

An extremely accurate and active device operated by compressed air which assures a regulation of humidity within exceedingly close ranges.

AMCO HUMIDITY CONTROL

Electrically Operated

Similar in principle to the Compressed Air Type except that the hydroscopic element operates electrical contacts which control the units.

April Showers Company

4126 Eighth Street, N. W. **Washington, D. C.**

(Trade Mark Reg. U. S. Pat. Office)

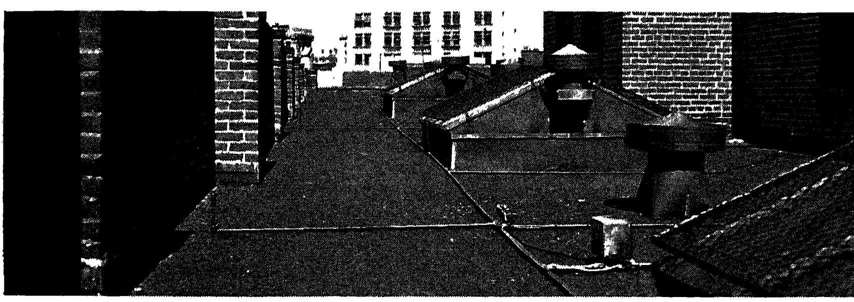
AUTOMATIC EVAPORATIVE ROOF COOLING

AN EFFECTIVE WATER INSULATOR for all KINDS of ROOFS

Distributors and Dealers in Principal Cities

• **Solar radiation** is converted to cooling effect, reducing normal heat transmission 50 per cent upward. Entire roof surface temperature is normally held at wet bulb temperature when evaporative factors are favorable. Skylights, shown below, are cooled with the same degree of effectiveness as roof.

• **APRIL SHOWERS** is operated by an automatic electric thermal control or self-contained expansion type control placed upon the roof in the SUN, and is turned off by the cooling effect of evaporation. Water used may be from city mains, wells, or waste water from condenser units.



Installation on U. S. Treasury Annex Building, Washington, D. C.

• **APRIL SHOWERS** roof cooler maintains a moderate temperature of roof and attic space. Roofs of built-up composition, waterproofed with pitch, will remain firm and intact. Disintegration of roof is minimized when held to normal temperature and protected with APRIL SHOWERS.

• Water consumption is approximately twenty gallons per day for 1,000 sq ft.

APRIL SHOWERS spray heads are scientifically constructed of fine brass and bronze, accurately machined for maximum operating efficiency, of low water consumption, operates on water pressures from 10 lb to 60 lb, and are designed for full and for fractional circle coverage to insure complete sprinkling of all roof surface. Installations are made with copper and bronze.

• Spray Head flow tables and performance data supplied; spray heads for humidification and other special uses made to order. Write for information on any problem involving use of water for cooling or humidification. Hundreds of installations have already proven the high insulating value of water cooling of roof areas.

• May be used on slate, tile, built-up, metal, slag, gravel, composition shingles, and cement roofs.

• **Inquiries will be promptly answered by our Engineers. Estimates free.**

Binks Manufacturing Co.

Plant and Executive Offices

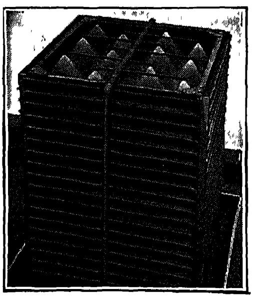
3114-3140 Carroll Avenue, **Chicago, Ill.**

Branch Offices

CLEVELAND, OHIO........1153 Leader News Bldg.	NEW YORK, N. Y........51 E. 42nd St.
DETROIT, MICH........2832 E. Grand Blvd.	PHILADELPHIA, PA........2045 N. Broad St.
LOS ANGELES, CALIF........239 W. 15th St.	PITTSBURGH, PA........200 Point Bldg.
MILWAUKEE, WIS........743 N. Fourth St.	SAN FRANCISCO, CALIF........778 Brannan St.
NEW ORLEANS, LA........400 Balter Bldg.	WINDSOR, ONTARIO, CANADA

Representatives in Principal Cities

COOLING TOWERS—SPRAY NOZZLES—COOLING PONDS—SPRAY PAINTING EQUIPMENT



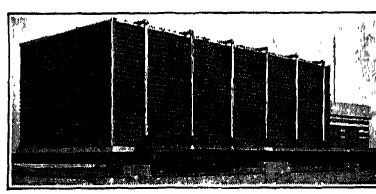
"Binks" Atmospheric Spray Cooling Towers

"Binks" Atmospheric Spray Cooling Towers are made in a wide range of sizes to handle capacities from 5 to 1200 gpm. They are used and recommended for all types of industrial water cooling work and are suitable in the standard sizes for refrigeration plants ranging from 2 to 240 tons and for Diesel engine plants ranging from 30 to 3600 hp.

The towers consist of a heavy shop welded copper-bearing steel frame, genuine wrought iron manifold with welded feeder arms and bronze nozzles. All metal parts are hot dip galvanized after fabrication.

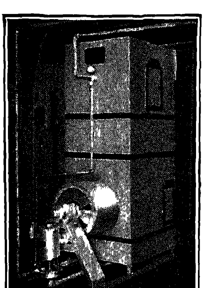
Louvres consist of clear all heart redwood or galvanized steel, to suit requirements.

Due to standardized construction features "Binks" Atmospheric Spray Towers may be quickly erected on the job, for either ground or roof installation from a simple elevation print furnished with each unit, without skilled factory supervision.



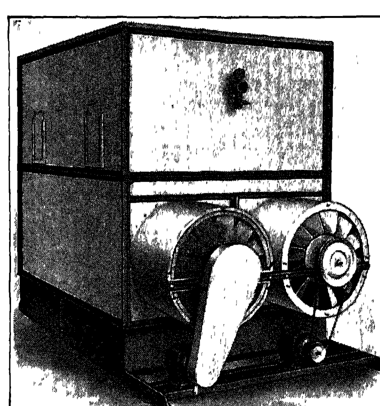
All bolts and nuts for assembling are cadmium plated, louvres being inserted to the tower frame in slip fit louvre retainer channels requiring the use of no bolts, nuts or any other fastening.

There are more than 3000 "Binks" Atmospheric Towers now in operation.



"Binks" Indoor Forced Draft Cooling Towers

"Binks" Indoor Forced Draft Cooling Towers are made in standard sizes from 5 gpm up to and including 300 gpm. They are extensively used for the cooling of jacket water for Diesel engines and large capacity air compressors, for condensing water for refrigeration machines and various industrial processes.



This type of equipment is exceptionally well adapted in locations where outdoor mounting of towers on building roofs would be extremely costly due to piping the installation to and from the tower. These units may be conveniently placed adjacent to the process with provision being made for fresh air inlet to the tower, such as open doors or windows, whereas the saturated air is conveyed to the outdoors through a duct.

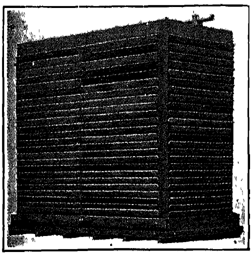
Write for Complete Data on New Small Towers for Self-Contained Air Conditioners and Similar Equipment!

The Marley Company

(Fairfax and Marley Roads,) Kansas City, Kansas

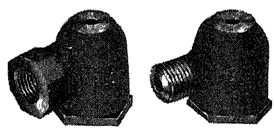
Branches or Agents in Principal Cities

Spray Nozzles and a Complete Line of Water Cooling Equipment

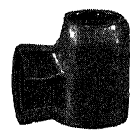


MARLEY NATURAL DRAFT TOWERS

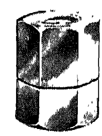
Practically unlimited range of closely graduated sizes, entirely shop fabricated. Minimum initial, maintenance and operating costs. Many exclusive MARLEY advantages. **Bulletins 201 and 202.**



MARLEY Small 2-Piece Nozzles for Brine Spraying, Air Washing and Similar uses.



MARLEY Ice-Melting Nozzle for cooling systems using ice.

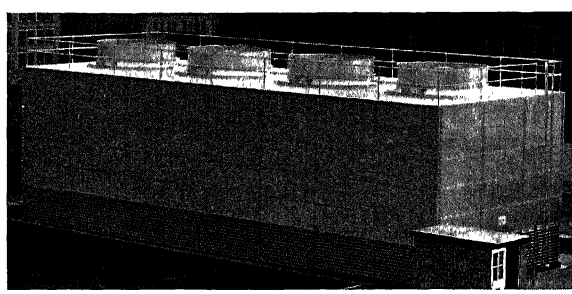


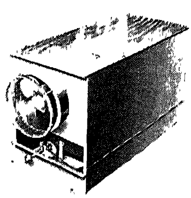
MARLEY Humidifying Nozzle adds moisture to air in open rooms or duct system.

Also Water Cooling Nozzles for Cooling Towers, Spray Ponds, etc.

MARLEY PATENTED NON-CLOG SPRAY NOZZLES

Made in scores of types and sizes. Practically any metal or alloy the purpose may demand. **Bulletins 101 and 102.**





SMALL INDUCED DRAFT TOWERS

Small, self-contained, steel units for 2 to 170 ton service, to go indoors or out. Smaller sizes (horizontal air flow) shipped all assembled, larger ones (vertical air flow) all shop fabricated for fast, easy assembly at location. **Bulletins 503 and 505.**

LARGE MARLEY MECHANICAL DRAFT TOWERS

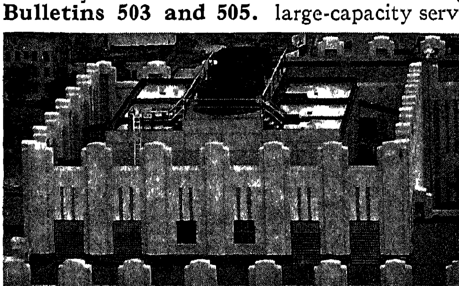
Both Forced and Induced Draft Towers, for heavy duty water cooling services of all kinds. Any capacity, with one fan or many, individually engineered to the exact requirements of each installation. MARLEY patents cover a variety of important features for extreme operating flexibility, high efficiency and economy.

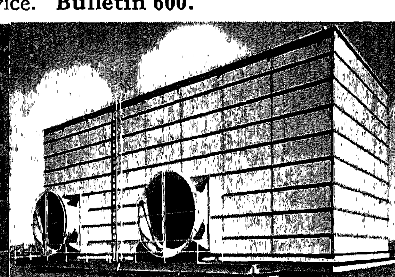
Redwood or Steel are standard materials, Transite and other materials on special order.

"Double-Flow" Induced Draft (below left) for largest capacities. **Bulletin 602.**

"Standard" Induced Draft (above) for usual large-capacity service. **Bulletin 601.**

Forced Draft (below right) for suitable applications in large-capacity service. **Bulletin 600.**





Also Many Other Types of Towers, Spray Ponds and Related Equipment

AEROFIN CORPORATION

410 So. Geddes Street

Syracuse, N. Y.

AEROFIN

Standardized Light-weight Heat Exchange Surface

Branch Offices

CLEVELAND, CHICAGO, NEW YORK, PHILADELPHIA, DETROIT, DALLAS, TORONTO

Aerofin is the modern Standardized Light-Weight Encased Fan System Heating and Cooling Surface originated by *Fan Engineers* to meet the present and future requirements of this highly specialized field. All Standard AEROFIN Units are furnished as completely encased Units, ready for pipe and duct connections. The patented casings are built of pressed steel and are exceptionally strong and rigid, protecting the Unit from all the strains of pipe connections and expansion or contraction in service. The casings are flanged on both faces, top and bottom, and template punched for bolting together adjacent Units, or for duct connection.

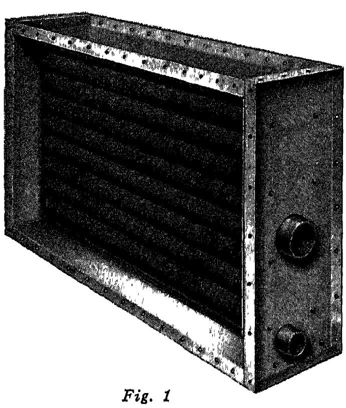


Fig. 1

Aerofin Non-freeze heater (Fig. 1) is non-freeze, non-stratifying spiral fin coil built into casing for air conditioning units or for installing in ducts. May be installed horizontally or vertically. Used on any two-pipe steam system for preheating or reheating. Modulating control on preheaters.

Available in 13 lengths and 3 widths, from net face area of 2.76 sq ft to 26.28 sq ft.

Tubing 1 in. O.D. Innertube ⅝ in. O.D.

Headers—Cast Brass.

Fins—spiral, turned copper.

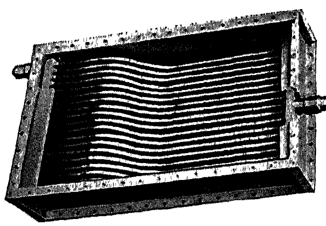


Fig. 2

Flexitube Aerofin (Fig. 2) is distinguished from all other developments by its off-set tubes, so arranged as to absorb all expansion and contraction strains.

Headers—Cast bronze or aluminum.

Tubing—⅝ in. O.D. copper, admiralty or aluminum.

Joints—Where admiralty or copper tubes are used together with bronze headers tubes are brazed to headers using Mueller patented joint. Where both aluminum tubes and headers are used tubing is welded to headers.

Casings—Copper, aluminum or galvanized iron.

Design—Constructed with headers on opposite ends making possible installation of units with tubes horizontal or vertical.

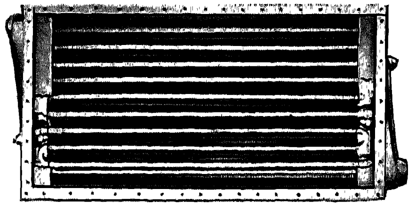


Fig. 3

Universal Aerofin (Fig. 3) is distinguished by its "S" bend construction of

tubing, units designed with steel headers on opposite ends, the ends of the "S" bends being connected thereto by compression nuts, the bends taking care of the expansion and contraction of the tubing.

Recommended where close control is desired.

Headers—Pressed steel.

Tubing—1 in. O.D. Copper, admiralty or aluminum.

Casings—Copper, aluminum or galvanized iron.

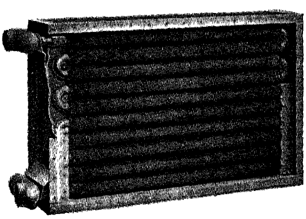


Fig. 4

High Pressure Aerofin (Fig. 4) is of continuous tube design, being recommended where extremely high pressures of steam are used.

Headers—Pressed steel.

Tubing—1 in. O.D. Copper, aluminum or admiralty.

Casings—Copper, aluminum or galvanized iron.

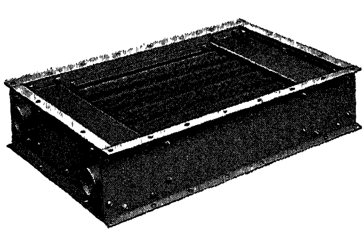


Fig. 5

Booster Aerofin (Fig. 5) is of the continuous tube design, recommended where small volumes of air are used, or to raise the air temperatures in branch ducts, etc.

Headers—Cast iron.

Tubing—5/8 in. O.D. Copper or aluminum.

Casings—Copper, aluminum or galvanized iron.

Aerofin Encased Booster Units: (Fig. 5). For horizontal or vertical air flow. Six sizes, 150 to 1624 cfm.

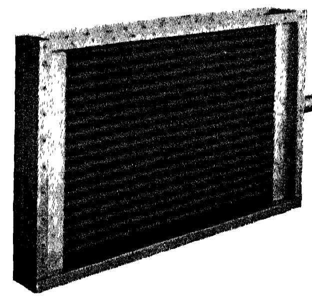


Fig. 6

Narrow Width Aerofin: (Fig. 6) recommended for water cooling or for flooded Freon systems. Made in straight tubes only with headers on opposite ends, joints between headers and tubing being brazed. Construction similar to Flexitube AEROFIN.

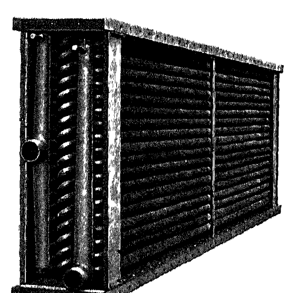


Fig. 7

Aerofin Continuous Tube Water Coils (Fig. 7) are designed for air cooling by circulating cold water through the AEROFIN and air over extended fin surface. Made for either horizontal or vertical air flow.

Tubes and fins are copper, completely tinned with permanent metallic bond

between fin and tubes. Headers are made of one-piece cast bronze and casings of heavy galvanized iron or copper.

Units tested to 1000 lb hydrostatic pressure.

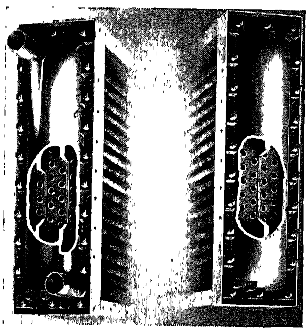


Fig. 8

Aerofin Cleanable Tube Units (Fig. 8) for cooling only and all made with headers removable to permit cleaning out tubes. Recommended for use where sediment or scale forming chemicals are present in the cooling water.

Headers—Cast iron.

Tubing—Copper or admiralty.

Casings—Copper or galvanized iron.

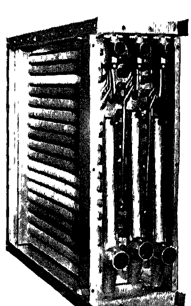


Fig. 9
End plate removed showing distributing and suction headers.

Aerofin Direct Expansion Units: (Fig. 9) Row Control Type—Recommended for use where cutting on or off rows of tubes in direction of air flow is desired. Suitable for use with Freon or Methyl-Chloride.

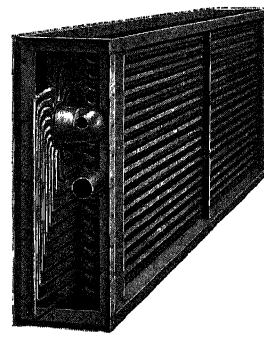


Fig. 10

Aerofin Direct Expansion Units: (Fig. 10) Centrifugal Header Type—Recommended where control of rows in direction of air flow is not required.

Advantages: Weighs but 9 to 16 per cent of same equivalent cast iron surface and occupies one-third of the space. Eliminates expensive foundations and building re-inforcement. Can be suspended from roof beams or trusses if necessary.

AEROFIN Sizes

Flexitube: 13 standard lengths, three widths, one and two rows deep.

Narrow: same as Flexitube.

Universal: 17 standard lengths, two widths, one and two rows deep.

Continuous Tube: 13 standard lengths, three widths, 2-3-4-5 and 6 rows deep.

Cleanable Tube: 17 standard lengths, one width, 2 and 4 rows deep.

Direct Expansion: Row Control—11 standard lengths, 3 widths, 1-2-3 rows deep. Face Control—11 standard lengths, 3 widths, 2-3-4-5-6 rows deep. Centrifugal Header—11 standard lengths, three widths, 2-3-4-5-6 rows deep.

Steel Supporting Legs: 18 in. and 24 in. high. Punched same bolt hole centers as standard casings. Quickly attached. No other foundation required.

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus. List upon request.

Write Syracuse for Heating Bulletin G-32; Direct Expansion Bulletin DE-34 on refrigeration type units; Continuous Tube Bulletin C. T. 34 for Water Cooling Coils; or phamplet on Cleanable Type AEROFIN for cooling.

The G & O Manufacturing Company

138 Winchester Avenue **New Haven, Connecticut**

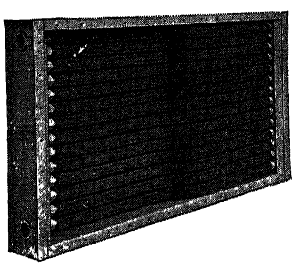
G&O

SQUARE FIN TUBING

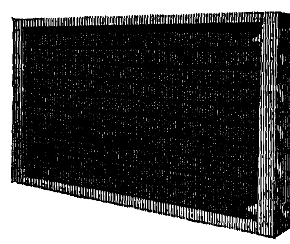
STRAIGHT LENGTHS—U-BENDS—CONTINUOUS COILS

RADIATING ELEMENTS FOR ALL HEAT TRANSFER PURPOSES

G&O Finned Radiation Coils for industrial applications are available in a wide range of sizes.



Universal U-102



Standard No. 10

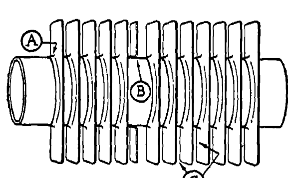
Send for Catalog and Price List

THE use of INDIVIDUAL fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface.

Fins of any size or shape may be obtained giving any desired proportion of primary and secondary surface.

A square fin has about 30 per cent greater surface than a round fin of a diameter equal to one side of the square.

Individual fins permit of any fin spacing; also, of using fins in groups at intervals along tubes.



A—Generous Fin Collar provides large contact area between Tube and Fin.

B—Tube expanded against Fin Collar; insures mechanically tight joint, made permanent by bond of high temperature alloy—complete thermal contact.

C—Free air-flow passages; non-clogging.

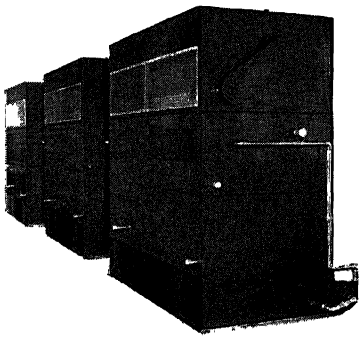
STANDARD SIZES

O.D. of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8"	7/8" sq.	6	0.80 sq. ft.
3/8"	7/8" r'd.	6	0.60 sq. ft.
3/4"	1 1/2" r'd.	6	1.55 sq. ft.
3/4"	1 5/8" sq.	6	2.40 sq. ft.
1"	2 1/8" sq.	6	4.00 sq. ft.
1 3/8"	2 3/8" r'd.	4	2.33 sq. ft.

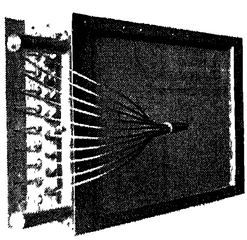
Acme Industries

Jackson **Michigan**

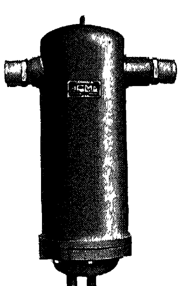
Offices in 30 Principal Cities



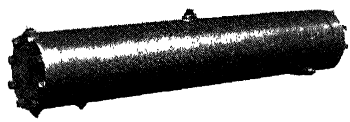
EVAPORATIVE CONDENSERS
Ask for Catalog No. 27



AIR CONDITIONING COILS
Direct Expansion
and
Water Cooled
Ask for Catalog No. 34



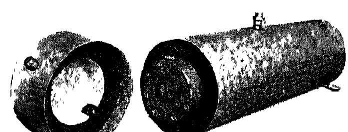
OIL SEPARATORS
5 Standard Sizes
Up to 50 Tons Capacity
Ask for Catalog No. 36



AMMONIA CONDENSERS
Ask for Catalog No. 21

FREON CONDENSERS
Ask for Catalogs Nos. 23 and 24

DRY-EX WATER CHILLERS
Shell and Tube Type
Through Tubes—No Bends
Refrigerant in Tubes
Controlled Water Velocity
Small Refrigerant Charge
Ask for Catalog No. 30



FLOODED SHELL AND TUBE COOLER
Plain or Insulated
Cleanable Tubes and Heads
Designed for

DRINKING WATER SYSTEM
BOTTLING PLANTS
Processing and ingredient Water
Ask for Catalog No. 29

ACME HEAT INTERCHANGERS
Prevents Liquid Return to Compressor, Chattering Valves, Sweating Suction Line—Acts as a Booster.

You don't have to choose between quality and price—BUY ACME

ACME ALSO MANUFACTURES

Pipe Coils **Liquid Receivers**

Accumulators **Specialties**

Baker Ice Machine Co., Inc.

Omaha, Nebr.

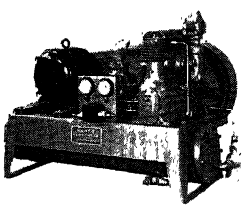
MANUFACTURERS OF INDUSTRIAL AND COMMERCIAL REFRIGERATION AND AIR CONDITIONING

Sales and Service in Principal Cities — **Cable Address: BAKERICE**

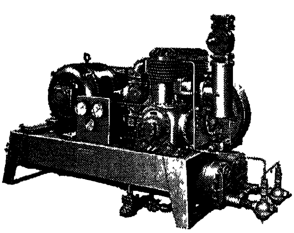
AUTHORITY ON MECHANICAL COOLING FOR 35 YEARS

Precision manufactured and designed for maximum service and economy, Baker equipment is world-famous for its high quality and dependable performance. Important compressor features include: full force feed lubrication, honed cylinders, double trunk type pistons, Timken roller bearings. Write for specifications and descriptive literature.

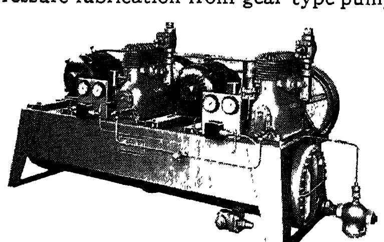
BAKER FREON AIR CONDITIONING UNITS



Small Condensing Units Complete line of self-contained, automatic units. From ¼ hp to 15 hp capacity. 2-and 4-cylinder types. Air- or water-cooled.



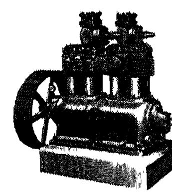
Large Condensing Units Available in 6 models, 20 to 60 hp inclusive, 4-cylinder, self-contained, automatic units. Shell and tube condenser-receiver. Pressure lubrication from gear type pump.



Dual Condensing Units

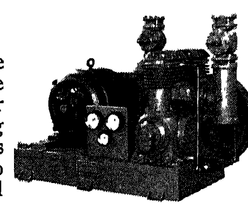
Designed especially for variable load requirements. Dual 4-cylinder type water-cooled unit. Automatic capacity control. Shell and tube type condenser.

Compressors

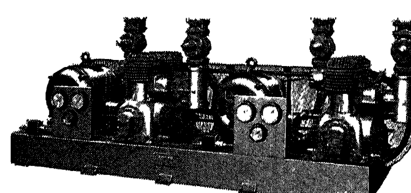


4-cylinder type, available in sizes from 10 hp to 60 hp. Semi-steel cylinders and pistons. Counter-balanced crankshaft, precision ground. Nickelite connecting rod bearings.

Compressor Units



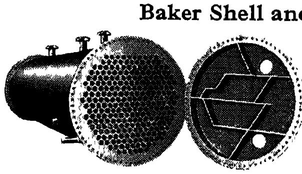
Arranged for use with evaporative type condenser or water cooling tower. Sizes range from 2 to 60 hp. 2- and 4-cylinder types. Automatic controls.



Dual Compressor Units for Capacity Control

19 different models, 10 hp to 120 hp inclusive. Two separate 4-cylinder compressors, automatic controls, independent motors, pressure lubrication, panel type gauge board. For use with separately mounted shell and tube or evaporative condenser.

Baker Shell and Tube Condensers and Liquid Coolers



(1 to 250 tons capacity)

Horizontal multipass or vertical shell and tube construction. Complete range of sizes up to 2500 sq ft of cooling surface in single shells. Code welded, seamless steel or hard copper tubes. Easily cleaned.

Brunner Manufacturing Company

Utica, New York, U. S. A.

For Years the Symbol of Quality

The Brunner line of refrigeration equipment includes Air Conditioning models up to and including 25 hp for all types of high temperature applications within their capacity, using either "Freon-12" or Methyl Chloride as refrigerant.

The Brunner field sales organization is available in all parts of the country, backed by outstanding achievements in engineering, and adoption of modern methods and design of air conditioning equipment.

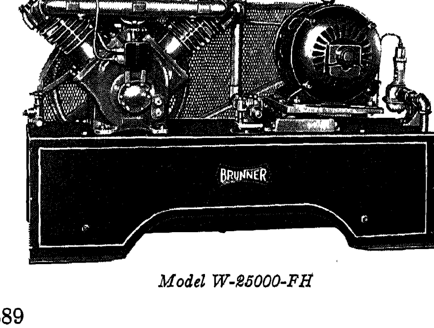
Installation of Brunner refrigerating units is insurance of the finest quality of materials and workmanship—plus the highest efficiency possible in modern design and manufacture.

SPECIFICATIONS					CAPACITIES Air Conditioning Units Based on 75 F Water Temperature "Freon-12" Refrigerant	DIMENSIONS
Model No.	Hp	Cyls.	Bore & Stroke	Rpm	Btu per Hr 40° Evap. Temp.	L.W.H.
W 300-FH	3	4	3¼ x 2¼	260	38547	47″ 23½″ 27″
W 500-FH	5	4	" " "	420	62270	" " "
W 750-FH	7½	4	4¼ x 3	260	91526	60″ 30″ 39″
W 1000-FH	10	4	" " "	350	123211	" " "
W 1500-FH	15	4	" " "	525	184815	" " "
W20000-FH	20	4	4¼ x 5	435	255046	72¼″ 31½″ 47¾″
W25000-FH	25	4	" " "	540	316652	" " "

Additional air and water cooled models from ¼ hp for commercial and industrial applications.

BRUNNER DEPENDABILITY . . .

is based on time-proven features of design and manufacture . . . all parts are precision machined within extremely close tolerances . . . bronze bearings throughout . . . extra large fin surface on cylinders and heads . . . bellows seal . . . silent eccentric drive (except on 20 hp and 25 hp models, which employ crankshaft) . . . suction and discharge valves in "all-in-one" plate assembly . . . heavy-duty motor with high starting torque . . . adjustable motor base . . . multiple V-belt drive. Throughout, Brunner Refrigeration Units are geared to the demands of *heavy-duty* service. Completely illustrated catalog on request.



Model W-25000-FH

Curtis Refrigerating Machine Company

Division of Curtis Manufacturing Company

1959 Kienlen Ave., **St. Louis, Mo., U. S. A.**

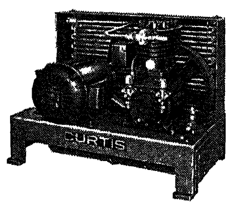
ESTABLISHED 1854

93 Condensing Units from 1/6 to 30 hp

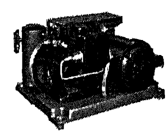
Unit Coolers and Evaporator Coils

PRODUCTS: Refrigerating Machinery; Forced Draft Cooling Units; Cooling Coils, Condensers, Shell and Tube Coolers, Valves, Fittings and Accessories, Complete Refrigerating Equipment for Dairies, Creameries, Ice Cream Cabinets, Ice Cream Making Plants, Cold Storage Locker Systems, Walk-in Coolers, Drinking Water Systems, Commercial and Low Temperature Cooling, Processing and Air Conditioning Installation, Packaged and Remote Types.

Commercial Refrigeration



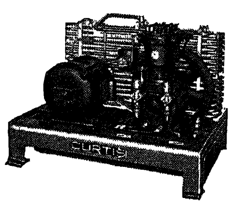
1½ hp Air Cooled Condensing Unit. Other sizes from ¼ to 5 hp.



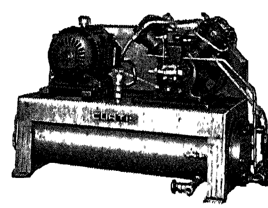
1/6 to ½ hp Self-Contained Condensing Unit.

45 air cooled condensing units from ⅙ to 5 hp, inclusive, and 47 water cooled units from ⅓ to 30 hp, inclusive. All models available for either Freon (F-12) or Methyl Chloride. Mechanical advantages include Timken Bearings, Centro-Ring Positive Pressure lubrication.

Special models are available for ice cream, frozen food cabinets and for the dairy industry.



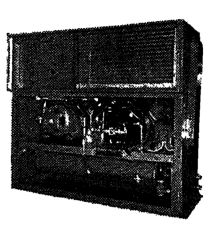
5 hp Water Cooled (Counterflow) Condensing Unit. Other sizes from ⅓ to 5 hp.



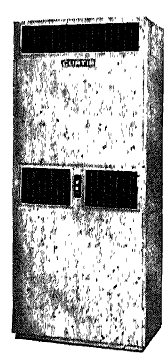
15 hp Cleanable Shell and Tube Condensing Unit. Other sizes from 3 to 30 hp.

Air Conditioning

For stores, offices and all types of commercial establishments Curtis offers complete packaged, refrigerated air conditioning units, requiring only water and electrical connections to install. Cools, dehumidifies, circulates and filters the air. Eliminates costly installation expense. Adaptable for heating.



7½-10-15 ton Remote or Central Type Air Conditioner.



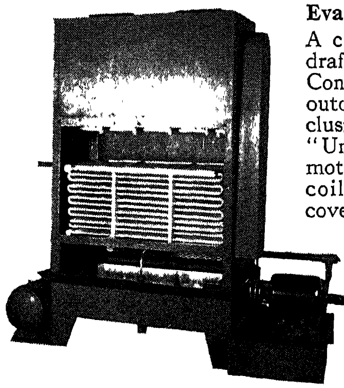
3 and 5 ton Packaged Type Air Conditioner.

Marlo Coil Co.

6135 Manchester Ave., **St. Louis, Mo.**

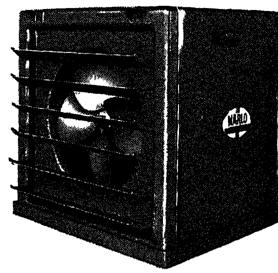
Refrigeration Equipment Manufacturers

Brine Spray Units—Unit Coolers—Evaporative Condensers—Low Temperature Units—Air Conditioning Units—Heating and Cooling Coils.



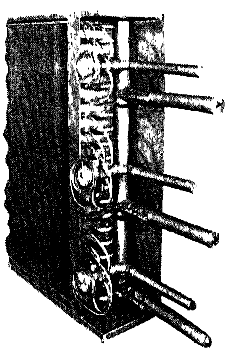
Evaporative Condenser

A combination forced-draft Cooling Tower and Condenser for indoor or outdoor installations. Exclusive Marlo features are "Unidrive" pump-blower motor; all prime surface coils; internal surface covered with corrosion resistant mastic; frame electric welded and galvanized after fabrication. See Bulletin No. 404.



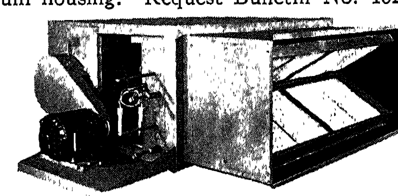
Unit Coolers

In this new model, air is pulled instead of forced through coils, thus utilizing complete coil surface and obtaining greater efficiency. Available in eight sizes, for all common refrigerants. All-aluminum housing. Request Bulletin No. 402.



Air Conditioning Coils—Blast Coils

Durably built; conservatively rated; available in materials suitable for any cooling or heating medium. All coils thoroughly dehydrated and tested at 1,000-pound pressure under water. Ask for Bulletin No. 396.

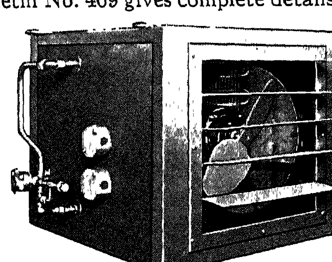


Air Conditioning Units

Air Conditioning Units in either ceiling suspended or floor type. Capacities from 900 cu ft to 12,000 cu ft. Sturdily built on angle welded iron frames of sectional design for easy installation. Bulletin No. 409 gives complete details.

Brine Spray Units

Specially designed to maintain temperature below freezing, and yet eliminate all defrosting problems. Write for Bulletin No. 403.



Low Temperature Unit

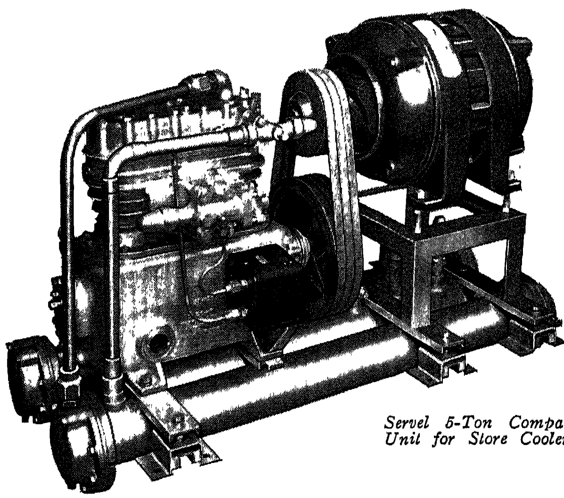
Designed for sub-zero temperature application. Equipped with the original Marlo electric-heating element for manual or automatic defrosting. Available for any refrigerant. Full details in Bulletin No. 407.

Servel, Inc.

Electric Refrigeration and Air Conditioning Division

Evansville, Indiana

AIR CONDITIONING



Servel 5-Ton Compact Unit for Store Coolers

Servel concentrates its engineering and manufacturing facilities on refrigerating machines for use in the air-conditioning industry.

Backed by 18 years of experience in building heavy-duty low-pressure refrigerating machines, Servel units offer every modern feature, plus a record of proven dependability.

This year, for the first time, Servel offers a series of compact, highly specialized Freon units for self-contained store coolers. These all carry 4- or 8-cylinder compressors, dynamically balanced, with provision for full floating suspension, and are extremely compact.

For remote applications, Servel offers a full line of units—both air-cooled and water-cooled—ranging in capacity from ½ ton to 20 tons. As a result of recent advancements in material and design, these models set a new standard for compactness, quiet operation, and freedom from vibration.

REMOTE MODELS | **COMPACT MODELS—For Store Coolers**

Model	Compressor	Tons	Model	Dimensions W	Dimensions D	Dimensions H
WJ75AF*	4 cyl. 1⅜ x 1⅜	¾				
WZ100AF*	4 cyl. 1⅝ x 1⅜	1				
WZ150AF*	4 cyl. 1⅝ x 1⅜	1½				
WQ200AF*	4 cyl. 1¾ x 1¾	2	WXQ200AF	31	19	22¾
WQ300AF*	4 cyl. 1¾ x 1¾	3	WXQ300AF	31	19	22¾
WT500AF	4 cyl. 2⅛ x 1¾	5	WXT500AF†	37½	19	25¼
WK750AF	8 cyl. 2⅛ x 1¾	7½	WXK750AF	40¾	23½	31½
WK1000AF	8 cyl. 2⅛ x 1¾	10	WXK1000AF	41¾	23½	32½
WV1500AF	4 cyl. 3¾ x 3¾	15				
WV2000AF	4 cyl. 3¾ x 3¾	20				

Smaller self-contained models built to order on quantity contract.

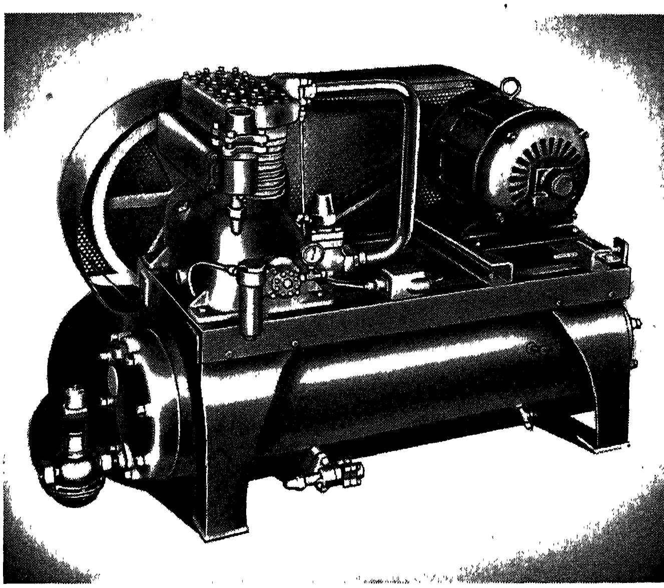
*Also available in air-cooled models.

†Illustrated above.

UNIVERSAL COOLER CORPORATION

Marion, Ohio

Automatic Refrigeration Exclusively Since 1922



Model W-1500, 15 hp Condensing Unit.

A *complete* LINE of CONDENSING UNITS AND COMPRESSORS

MANUFACTURERS: We offer you a complete line of commercial condensing units and compressors. Our unique policy of selling to manufacturers only is ideally suited to your business. Our products are accepted by nationally and internationally known makers of refrigerating and air conditioning equipment. Data and quotations sent promptly at your request.

ARCHITECTS AND ENGINEERS: We will gladly send you descriptive matter and technical capacity and performance data on our condensing units. You can use this information with confidence in preparing your own specifications for refrigeration and air conditioning work.

CONTRACTORS: These splendid refrigerating Machines are available to you through many of the leading manufacturers of refrigerating, air conditioning and fan equipment. We maintain no local dealers or branches to compete with you in contract work.

We will gladly send complete data.
Universal Cooler Corporation, Marion, Ohio.
Universal Cooler Company of Canada, Ltd., Brantford, Ontario.

Listed under Reexamination Service of Underwriters Laboratories, Inc.

Worthington Pump and Machinery Corporation

WORTHINGTON Carbondale Division **CARBONDALE**

General Offices: HARRISON, NEW JERSEY

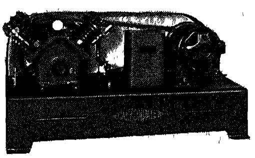
ALBANY	CHICAGO	EL PASO	NEW HAVEN	PROVIDENCE	SEATTLE
ATLANTA	CINCINNATI	FORT WORTH	NEW ORLEANS	ROCHESTER, N.Y.	SPRINGFIELD, MASS.
BALTIMORE	CLEVELAND	GALVESTON	NEW YORK	ST. LOUIS	SYRACUSE
BIRMINGHAM	DALLAS	HOUSTON	PHILADELPHIA	ST. PAUL	TULSA
BOSTON	DENVER	KANSAS CITY	PITTSBURGH	SAN FRANCISCO	WASHINGTON, D.C.
BUFFALO	DETROIT	LOS ANGELES	PORTLAND, ORE.	SALT LAKE CITY	WILMINGTON, DEL.

Representatives in Principal Cities of Foreign Countries CA1-1

REFRIGERATION SYSTEMS FOR AIR CONDITIONING IN COMFORT COOLING OR INDUSTRIAL PROCESS

Complete refrigerating systems for use with Freon-11, Freon-12, Methyl Chloride, Ammonia, or Carbon Dioxide, either direct-expansion or water cooling applications. A complete line of refrigeration compressors, permitting impartial recommendations. A nation-wide organization of Distributors in major cities to provide sales and engineering service and plan complete air conditioning systems of the central or unit type. Architects, Engineers, and Contractors are invited to consult with us. Write to Harrison, N. J., or any branch office, for bulletins on these products.

Small Self-contained Units

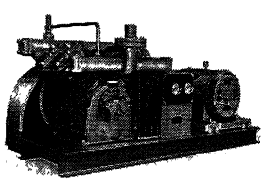


Freon - 12 or methyl chloride condensing units; motors ¼ to 25 hp; ratings up to 25 tons.

Overall Dimensions

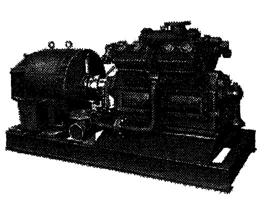
Smallest: 20 in. long, 16 in. wide, 16 in. high
Largest: 108 in. long, 45 in. wide, 52 in. high

Medium Self-contained Units



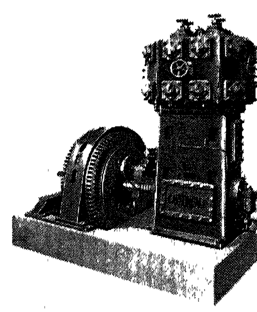
Freon - 12 or methyl chloride units, equipped with capacity control; patented Feather Valves; motors of 30 to 60 hp.

Large Self-contained Units



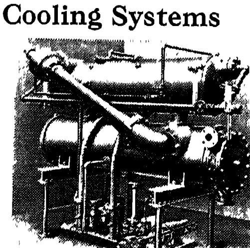
Freon - 12 refrigeration units utilizing the new, modern eight-cylinder V-type compressor built into a compact unit. V-belt or direct motor drive. Variable capacity control. Motors of 75, 100, and 125 hp.

Vertical Duplex Double-acting Compressors



Freon - 12 or ammonia; large tonnage compressors; force-feed lubrication; roller main bearings. Crankcase sealed from cylinders, preventing contamination of oil by refrigerant. Equipped with patented Feather Valves; automatic capacity control features. Crosshead incorporated in enclosed crankcase. Capacities up to 250 tons.

Steam Jet Water Cooling Systems



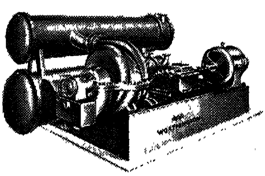
15 to 600 tons. Either surface or barometric condenser may be furnished.

Miscellaneous

High and low side equipment for every purpose.

Worthington Pump and Machinery Corporation, Carbondale Division

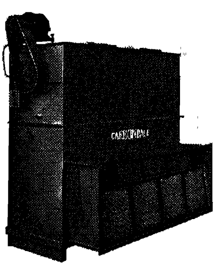
Centrifugal Refrigeration Water Cooling Systems



Freon-11 centrifugal compressor, water cooler and water-cooled condenser in compact unit assembly. Electric motor or steam turbine drive. 42 unit sizes . . . 150 to 650 tons.

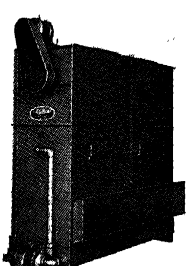
Air Conditioning Units

For Direct Expansion Freon-12 or Chilled Water Circulation



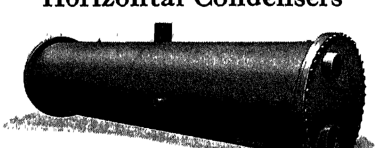
Vertical and horizontal; 500 to 12,000 cfm; large air passages; slow speed, quiet rugged fans; separable sections; readily accessible. The design permits flexibility in installation arrangements.

Shower Condensers



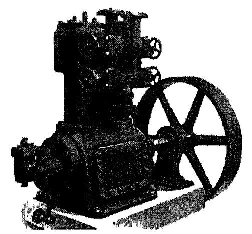
A combined condenser, receiver, and modified cooling tower, in one assembly, for Freon-12 or methyl chloride systems; 2 to 130 tons refrigeration; built in separable sections; all parts easily accessible. Saves 90 to 95 per cent in cost of water.

Horizontal Condensers



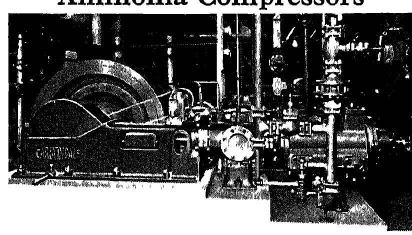
Atmospheric drip type, for warm corrosive waters. Double-pipe for closed systems, can be retubed without shutting down. Multi-pass for closed systems and space saving.

Vertical Ammonia Compressors



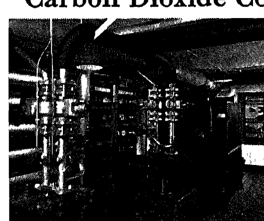
Pressure-lubricated; roller main bearings; safety heads; patented Feather Valves; belt drive, or direct-connected to electric motor, Diesel or gas engine; ratings from 2 to 160 tons in one unit.

Horizontal Ammonia Compressors



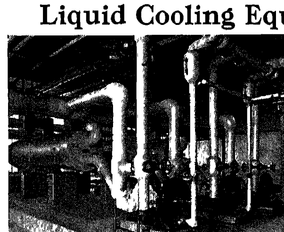
Single and duplex; single-stage and two-stage; belt drive, or direct-connected to electric motor, Diesel, gas or steam engine; patented Feather Valves; ratings from 60 to 750 tons. Automatic capacity control features are easily applied. Space requirements vary depending upon type and drive.

Carbon Dioxide Compressors



A series of convenient types and sizes for every requirement is available.

Liquid Cooling Equipment



Various designs of horizontal single and multi-pass types, for a wide range of services; also vertical types. Chillers for oil dewaxing. Single and double-pipe for milk, wort, chemicals, etc. Cold liquid circulating systems.

The Vilter Manufacturing Company

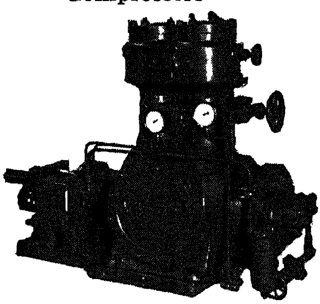
Since 1867

Milwaukee, Wisconsin

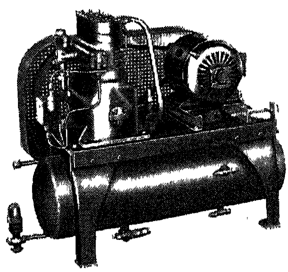
AIR CONDITIONING EQUIPMENT FOR INDUSTRIAL OR COMFORT COOLING

COMPRESSORS OF MODERN DESIGN

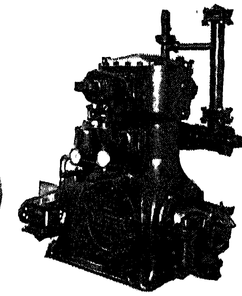
Ammonia Compressors



Freon Condensing Unit



Freon Compressors for Large Installations



Ammonia Compressors—The result of over seventy years of research development and experience gained through thousands of installations of all types, in all industries. Famous for high tonnage capacity at low hp and low operating costs. Built in a wide range of capacities from 2 to 100 tons standard A.S.R.E. rating in Vertical Types; up to 750 tons in Horizontal Type.

Freon Compressors—Embody many outstanding new features that prevent leakage and minimize friction—resulting in extremely low relative hp per ton. Made in capacities up to 150 tons. Capacitrols available at slight additional cost provide flexibility of operation.

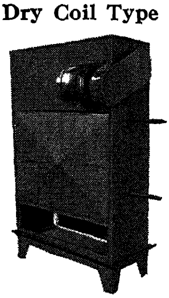
Freon Condensing Units—Self-contained units made in sizes from ¼ hp to 30 tons capacity. Embody latest engineering features.

Unit Air Coolers—Available in a wide range of sizes and types for any air conditioning requirement—product coolers, dry coil coolers, spray type coolers, low temperatures electric defrosting coolers, and floor or ceiling central system air conditioners.

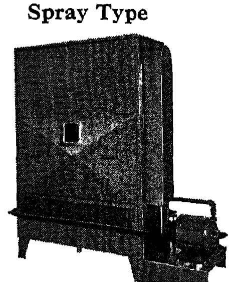
Self-Contained Air Conditioners—3 and 5 hp. May be placed directly in space to be cooled, or connected with duct work. Attractively finished to harmonize with any decorative scheme.

UNIT AIR CONDITIONERS

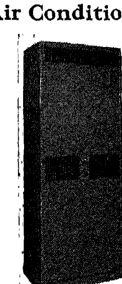
Dry Coil Type



Spray Type



Self-Contained Air Conditioner



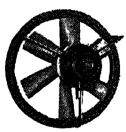
Vilter also builds a complete line of shell and tube water coolers, brine coolers, condensers; air conditioning coils, evaporative condensers and air washers—and special units for central station comfort cooling systems.

Autovent Fan & Blower Company

1809-23 N. Kostner Ave., **Chicago, Illinois**

FANS — BLOWERS — UNIT HEATERS

Member National Association of Fan Manufacturers and Industrial Unit Heater Association



AUTOVENT "31 Series" PROPELLER FANS

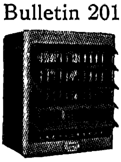
Exclusive Autovent design—will not churn air or overload motor. Ruggedly constructed for indefinite economical operation under severe conditions. Available for quiet operation. Capacities from 450 to 38,000 cfm. Bulletin No. 200-A.

ACID-MOISTURE PROOF PROPELLER FANS

For use where corrosive acid fumes or excess moisture exists. All exclusive Autovent features. Available with Bakelite lacquer protective coating for average condition or a new, Heresite Chemical (375°) baked finish for severe cases. Bulletin No. 201-A.

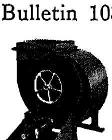
VAPOR-EXPLOSION PROOF PROPELLER FANS

Safe ventilation regardless of the dust or fume hazard. Non-ferrous fan wheels and chemical coatings furnished to requirement. Underwriters Label Class 1, Group D. motors. 12 in. to 36 in. fan sizes. Bulletin 201-A.



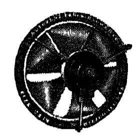
New AUTOVENT Super-Type UNIT HEATERS

This suspended type heater forces air circulation and directs warm air to lower part of room. Heating element of flat copper fins, attached to seamless, drawn copper tubes. No welded or brazed joints in coil or header. Fans have non-overloading power feature. Motor furnished to requirement. Capacities from 20,000 Btu to 398,500 Btu. Bulletin 103.



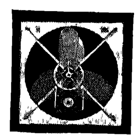
AUTOVENT "V" BELT DRIVEN UNIT BLOWERS

Forwardly curved blades. Quiet bearings. Motor mounted on steel pedestal, integral with blower housing. Air delivery can be decreased or increased. Interchangeable motors. Sturdily constructed of sheet steel. Bulletin No. 300-A.



AUTOVENT "BW" PROPELLER FANS

A slow speed operating bucket wheel type fan. Provides efficient ventilation for unlimited uses from 750 to 40,000 cfm. Sturdy construction. Minimum power. Quiet operation. Bulletin No. 202-B.

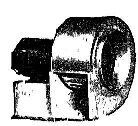


COOLVENT ATTIC FANS

Solve the low cost summer comfort-cooling and ventilating problem for homes with a quiet attic fan! Bearings and motor are rubber mounted to insure quiet operation. Built in sizes 24 in. to 54 in. Bulletin No. 40.

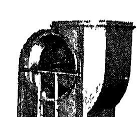
ALLVENT ALL-PURPOSE FAN

Special, QUIET operating fans designed for stores, offices, factories. Bulletin No. 206.



AUTOVENT UNIBLADE VOLUME BLOWERS

Motor driven—universal discharge for fume hoods, chemical labs, processing, drying, forced draft, etc. Handle low volumes of air at medium pressures. Wheels range from 6 in. to 11 in. diameters—same design as heavy duty blowers. Can be mounted on floor, wall or ceiling. Direct Connected Blowers for general ventilating applications available in wheel diameters up to 25 in. Bulletin No. 300-A.



AUTOVENT UNIBLADE TYPE "H" or "HB" BLOWERS

Ideal for ventilating and air conditioning installations featuring backwardly inclined blades and non-overloading power characteristics. Also comes with forwardly curved blades. Belt driven, centrifugal type in single or double widths with 12½ in. to 75 in. wheel diameters. Can be furnished to any speed or discharge requirement. Bulletin No. 301. Backward curve type HB Belt Driven Blowers, Bulletin No. 302.

The Complete Line of **Autovent** Propeller Fans and Blowers is tested and rated in accordance with the Standard Test Code adopted jointly by the *National Association of Fan Manufacturers* and the American Society of Heating and Ventilating Engineers.

American Coolair Corporation

Jacksonville, Florida

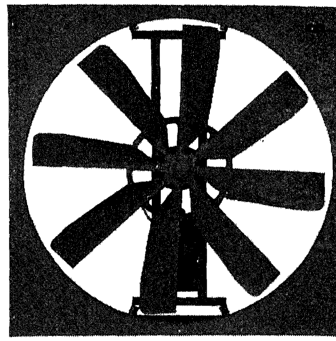
Cooling and Ventilating Fan Systems
For Homes, Offices, Stores, Factories, Etc.
A Pioneer Manufacturer of Attic Fans for Home Cooling
Charter Member—Propeller Fan Manufacturers Association

COOLING BY AIR MOVEMENT

Engineers and architects know well that people working or living within a building can be kept cool, healthfully and economically, by sufficient movement of fresh air. It is not so well known, perhaps, that such cooling requires four to eight times the volume of moving air needed for simple ventilation. Satisfactory cooling requires a complete air change in working or living quarters at least once a minute.

The American Coolair Corporation pioneered in the manufacture of attic fans for home cooling and during the past 12 years, Coolair engineers have been directly responsible for many of the developments in this field.

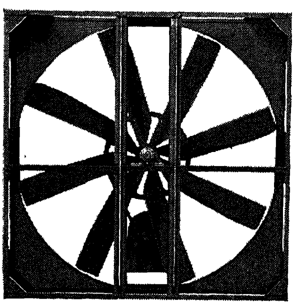
In planning a Coolair installation, determine cubic content of space to be cooled or ventilated and select fan of ample capacity based on the table of Recommended Air Changes shown. Write for Coolair's FREE catalog containing detailed installation suggestions for home and commercial jobs.



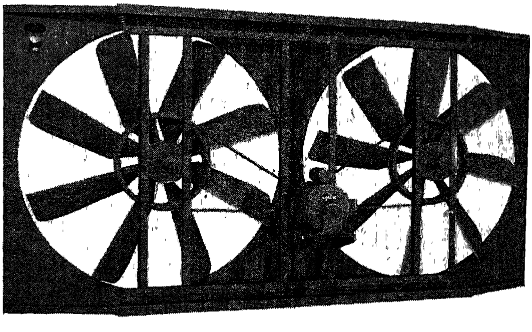
Coolair Type O, showing new built-in springs. Ideal for attic, window and wall installations in homes, offices, restaurants, stores, etc.

FEATURES OF COOLAIR FANS

1. **New Built-In Springs**—completely insulate moving parts from frame, eliminating vibration noise. Simplify window, wall and attic installations. 2. **Light, Compact Fabricated Steel Frame**—fits into many places where fans with bulky metal housings cannot be used. Easy to handle and install. 3. **Reversible**—when equipped with reversible motor, fan will blow in or exhaust at will. 4. **Ball Bearings in Fan Hub**—Eliminate sleeve bearing chatter and end thrust knock. Permit operation in any position (specify ball bearing motor for vertical or angle discharge). Uses grease instead of oil, requiring attention only once a year (once each three years for residential service). 5. **Eight Large, Slowly-Moving Steel Blades**—instead of four or less usually found on cheaper fans. Up to 12 blades on Type S models. Low tip speeds for quiet performance, steady flow of air. 6. **Efficient V-Belt Drive**—and small motor for efficient operating speed. 7. **Efficient Long Hour Service Motors**—rubber mounted on adjustable support. Nationally known make approved for this application. 8. **Certified Air Ratings**—in accordance with Standard Test Code of *NAFM* and ASHVE. 9. **Five-Year Guarantee on Residence Fans** (belts and motors 1 year).



Coolair Type S, diameters 6 to 9 ft, capacities up to 150,000 cfm. Used in hotels, factories, auditoriums, etc.



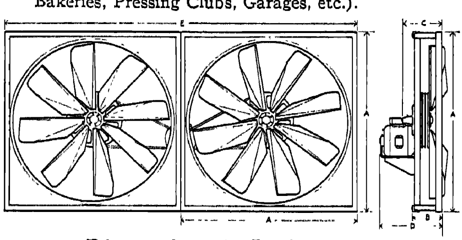
Coolair Type T, space-saving, efficient twin unit widely used where headroom is limited. Shown with new built-in springs. U. S. Patent No. 2,108,738

Performance Data—Coolair Belt Drive Fans

Fan Size		Hp	Fan rpm	Cu Ft Air Per Min
2-O 26″	B	1/4 L	448	4600
	C	1/4 H	535	5500
	D	1/3	595	6100
2 1/2-O 32″	B	1/4	395	7500
	C	1/3	432	8200
	D	1/2	495	9400
	D	3/4	567	10700
3-O 38″	B	1/4	308	10000
	C	1/3	335	10800
	C	1/2	394	12800
	D	3/4	440	14200
	D	1	485	15700
3 1/2-O 44″	B	1/3	244	12500
	C	1/2	294	15000
	C	3/4	325	16600
	D	1	358	18300
	D	1 1/2	410	21000
4-O 50″	B	1/2	248	18200
	C	3/4	290	21300
	D	1	320	23500
	D	1 1/2	366	26800
4 1/2-O 56″	B	1/2	210	21800
	C	3/4	236	24500
	C	1	260	27000
	D	1 1/2	298	30900
	D	2	328	34000
5-O 62″	B	1/2	175	24600
	C	3/4	201	28200
	C	1	222	31200
	C	1 1/2	254	35700
	D	2	280	39300
	D	3	321	45100

Fan Size		Hp	Approx. Fan rpm	Cu Ft Air Per Min
6-S 72″	B	1	171	32000
	C	1 1/2	190	35500
	C	2	206	38500
	D	3	263	49000
	D	5	306	57500
7-S 84″	B	2	163	49500
	C	3	176	53500
	D	5	227	72000
	D	7 1/2	262	80000
8-S 96″	B	3	147	66500
	C	5	176	80000
	D	7 1/2	204	93000
	D	10	259	117000
9-S 108″	B	5	150	98000
	C	7 1/2	165	108000
	D	10	193	128000
	D	15	230	150000
2-OT 26″ Twin	B	1/3	450	9400
	C	1/2	514	10700
2 1/2-OT 32″ Twin	B	1/2	393	15300
	C	3/4	450	17500
3-OT 38″ Twin	B	1/2	295	19200
	C	3/4	337	21800
3 1/2-OT 44″ Twin	B	3/4	257	26200
	C	1	283	28800
4-OT 50″ Twin	B	1	234	34600
	C	1 1/2	270	40000

B—Very Quiet (Homes, Theaters, Hospitals, etc).
C—Quiet (Stores, Offices, Restaurants, Barber Shops, etc.).
D—Industrial (Laundries, Factories, Canneries, Bakeries, Pressing Clubs, Garages, etc.).



Dimensions in Inches

Fan Size	A	B	C	D Approx.
2-O	30 5/8	6 3/8	8 3/4	13
2 1/2-O	36 5/8	7 3/16	9 11/16	13
3-O	42 5/8	7 3/4	9 5/16	14
3 1/2-O	49	7 3/4	10 3/8	17
4-O	55 1/8	9	11 11/16	19
4 1/2-O	61 1/8	9 1/2	12 1/4	20
5-O	67 5/8	9 9/16	12 1/4	20
6-S	75 1/4	15 1/2	17 1/2	28
7-S	86 7/8	21 11/16	22 9/16	34
8-S	98 15/16	20	34	38
9-S	111 1/8	20	34	38

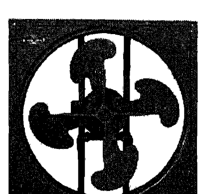
Fan Size	A	B	C	D Approx.	E
2-OT	30 5/8	6 9/16	9 1/8	13	61 1/4
2 1/2-OT	36 5/8	7 3/8	9 7/8	13	73 1/4
3-OT	42 5/8	7 7/8	9 3/8	14	85 1/4
3 1/2-OT	49	7 7/8	10 1/4	17	98
4-OT	55 1/8	9 5/16	12 1/16	19	110 1/4

Recommended Air Changes

Type Building	Air Change in Minutes for: Cooling	Ventilating
Bakeries, Boiler Rooms, Canneries, Pressing Clubs, Laundries, Restaurants, Kitchens, Etc.	1/2-1	2-5
Barber Shops, Beauty Parlors, Billiard Rooms, Bowling Alleys, Garages, Factories, Restaurants, Stores, Etc.	1/2-1 1/2	3-6
Auditoriums, Churches, Club Rooms, Schools, Theaters, Etc.	3/4-1 1/2*	4-8**
Homes, Apartments, Hotels, Hospitals, Offices, Banks, Undertaking Establishments, Etc.	3/4-2	4-8

*When based on seating capacity, allow 150-200 cfm per person.
**Minimum on basis of seating capacity is 35 cfm per person.

COOLAIR DIRECT DRIVE FANS



Coolair Direct Drive Fan 16-24 in.

Manufactured in four sizes from 16 to 24 in. in diameter. Positively quiet at 1200 rpm, not excessively noisy at 1800 rpm. Unique noise-reducing design of blade is covered by U. S. Patent No. 1855660.

WRITE FOR FREE ILLUSTRATED COOLAIR CATALOG

Buffalo Forge Company

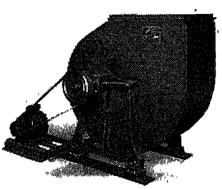
450 Broadway, **Buffalo, N. Y.**

Branch Offices

ALBANY, N. Y. .... 1305 Standard Bldg.
ATLANTA, GA. .... 305 Techwood Drive
BALTIMORE, MD. .... 508 St. Paul St.
BOSTON, MASS., Melrose Sta. .... 486 Main St.
CHICAGO, ILL. .... 20 North Wacker Drive
CINCINNATI, OHIO .... 626 Broadway
CLEVELAND, OHIO .... 418 Rockefeller Bldg.
DALLAS, TEXAS .... 1801 Tower Petroleum Bldg.
DAVENPORT, IOWA—D. C. Murphy Co., 305 Security Bldg.
DENVER, COLO.—Hendrie & Bolthoff Mfg. & Supply Co., 1635 Seventeenth St.
DES MOINES, IOWA—D. C. Murphy Co., 214 Old Colony Bldg.
DETROIT, MICH.—Coon DeVisser Co., 2051 W. Lafayette Blvd.
GREENVILLE, S. C. .... 21 Blue Bldg.
HOUSTON, TEXAS .... 505 Rusk Bldg.
KANSAS CITY, MO. .... 424 Dwight Bldg.
KITCHENER, ONT., CANADA—Canadian Blower & Forge Co., Ltd.
KNOXVILLE, TENN.—C. F. Secton .... 702 Empire Bldg.
LOS ANGELES, CALIF. .... 708 Pershing Square Bldg.
MIAMI, FLA.—Southern Air Conditioning Corp., 149 N. E. 20th Terrace
MINNEAPOLIS, MINN. .... 2102 Foshay Tower
NASHVILLE, TENN.—Southern Sales Co., 117 Fifth Ave., North
NEWARK .... 27 Washington St., Room 203
NEW ORLEANS, LA.—Devlin Bros. .... 1003 Maritime Bldg.
NEW YORK, N. Y. .... 39 Cortlandt Bldg., Room 1110
PHILADELPHIA, PA. .... 703 Cunard Bldg.
PITTSBURGH, PA. .... 431 Fulton Bldg.
RICHMOND, VA.—Williamson & Wilmer, Inc., Mutual Bldg.
SAN FRANCISCO, CALIF.—Moore Machinery Co., 1699 Van Ness Ave.
ST. LOUIS, MO. .... 1598 Arcade Bldg.
SEATTLE, WASH. .... 500 First Ave., South
TOLEDO, OHIO .... 1922 Linwood Ave.
WASHINGTON, D. C.—640 Woodward Bldg., 15th and H Sts., N. W.

PRODUCTS: Heating and Ventilating Equipment including: Unit Heaters, Multiblade Fans, Pipe Coil Heaters, Buffalo Air Washers, Buffalo Unit Air Washers, Buffalo Unit Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Exhaust Fans, Blowers, Dust Collectors, Disc Fans, Spray Nozzles.

Buffalo Limit Load Fans

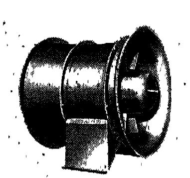


These fans are proving increasingly popular in the industrial field because of: 1. High efficiency. 2. Durable construction. 3. Non-overloading characteristic which means that no matter how much the fan load varies the motor will not overload and burn out. Fans are available with Buffalo Silent Floating Bases assuring extremely quiet operation.

Breezo Ventilating Fans

Use these efficient inexpensive fans wherever they may be installed to exhaust into the open air. Sizes from 8 in. to 36 in. in diameter. Prompt shipments from stock.

Buf-flow Axial Flow Fans

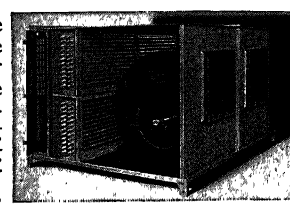


The design of this high pressure "Buf-flow" fan—with special directional guide vanes—propels the air stream in a *true axial direction*. No wasted energy—hence greater efficiency and marked power savings. What's more, this fan cannot overload and burn out motor.

Fans for Every Ventilating Need

For over 60 years we have specialized in designing and building fans for every conceivable use. Write for our fan catalogs.

Buffalo Comfort Conditioning Cabinets



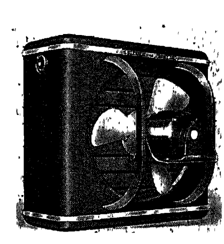
Available for simple cooling, for complete air-conditioning including winter heating. Neat, compact units that are easy to install and easy to service. Cooling capacities—3 tons up.

Buffalo Air Washers

For air washers whose design and performance has stood the test of time it will pay you to specify "Buffalo." Bulletin 3142-A gives you complete details.

Buffalo Unit Heaters

Buffalo Unit Heaters are built in a wide range of sizes and types for the efficient and economical handling of any heating problem. Operate with either low or high pressuresteam.



Breezo-Fin Steam Unit

Champion Blower & Forge Co.

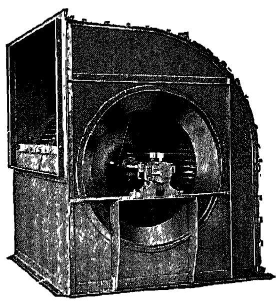
Manufacturers and Engineers

Plant and Offices: **Lancaster, Pa.**

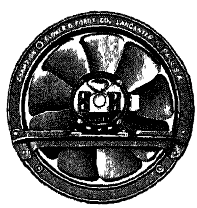
Address Correspondence to Div. 9

Manufacturers of Blowers, Ventilating Fans and Exhaust Fans for Air and Material; and Blast Gates

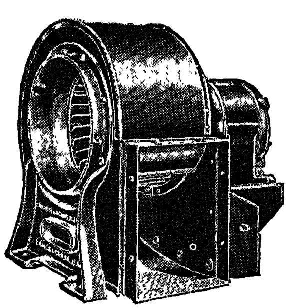
Representatives in Principal Cities



Type S Forward curve ventilating fans, single and double width, as well as direct motor drive.



Super Ventilating fans, direct motor drive up to 36 in. diameter. Motor belt drive up to 48 in. size.



Type BC Backward curve ventilating and exhaust fans, single and double width; belt driven and direct connected electric.

Ventilating Fans—For air conditioning systems and mechanical draft. Manufactured in the forward curved type for slow speed, and extremely quiet operation; also in the backward curved type with its flat horsepower curve characteristics and higher speeds suitable in the smaller sizes for direct connecting to synchronous speed motors. Ventilating Blowers manufactured in sizes up to 60 in. diameter wheel, in single and double width. Belt driven blowers equipped with either ball or high-grade babbit bearing. Direct motor drive can be equipped with any type or characteristic motor desired, in any arrangement.

Disc Fans—Super Ventilating Fans made in direct connected type up to 36 in. diameter, totally enclosed, ball thrust type motors. Slow speed motor belt driven type manufactured in sizes up to 48 in. for attic exhaust work and wherever large volumes of air are to be moved against low static pressures. All disc fans are quiet in operation. Decibel ratings on all fans are available.

Forced Draft Fans—All sizes for use on the smallest to largest boilers. Fans can be furnished with inlet or outlet adjustable louvers for controlling air volume.

Blast Wheels—We are well equipped to manufacture single and double width blast wheels in forward or backward curve type for oil burner and stoker manufacturers, as well as manufacturers of air conditioning units and other ventilating equipment.

Vibration Dampener Sub-Bases—For blower and ventilating equipment. Made with heavy channel iron and rubber vibration eliminator pads to suit size and weight of fan or blower.

Special Fan Equipment—We are in position to engineer and build fans, blowers, or exhausting equipment to meet customers' special needs. A card addressed to Div. 9 will bring you complete catalog data or information on any particular problem confronting you.

DeBothezat Ventilating Equipment Division

American Machine and Metals, Inc.

(Main Office and Factory)

901 DeBothezat St., **East Moline, Illinois**

District Offices

Chicago **New York** **San Francisco**

Branches

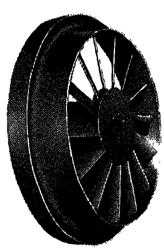
ATLANTA
BOSTON
CINCINNATI
CLEVELAND
DALLAS
DES MOINES
DETROIT
FORT WAYNE
HARTFORD

Branches

INDIANAPOLIS
LOS ANGELES
MILWAUKEE
MINNEAPOLIS
NEW ORLEANS
PITTSBURGH
PROVIDENCE
SAGINAW
ST. LOUIS

AXIAL-FLOW
PRESSURE FANS

NON-OVERLOADING POWER CHARACTERISTICS
CERTIFIED RATINGS—GUARANTEED PERFORMANCE



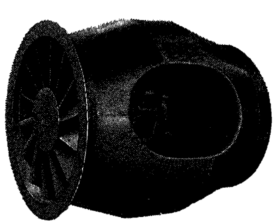
Ventilating Fan
Axial Flow

Axial Flow Ventilating Sets

A complete series of volume and pressure axial-flow fans of high mechanical and static efficiencies with a non-overloading power characteristic. These fans offer savings in space, weight and power. Axial-Flow Ventilating Sets are available in a wide range of capacities in sizes 8 in. through 10 ft in diameter, and may be had arranged for direct motor drive or belt drive.

Bifurcators

Designed for handling corrosive or high temperature vapors with direct motor driven fan. Motor is located in chamber open to atmosphere but isolated from gases handled by fan. Installed as integral part of duct system, in any position.



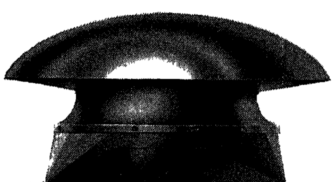
Bifurcator

Multi-Stage Impeller Blowers

Units can be furnished in 2, 4, 6 or 8 stages. Direct motor or belt driven, producing high capacities and static pressures, with non-overloading power characteristics.

"Power-Flow" Roof Ventilators

Designed to provide positive ventilation at all times regardless of temperature, humidity and wind velocity. Guaranteed performance ratings. Equipped with high-efficiency Axial-Flow Pressure Fan, these Power-Flow Roof Ventilators possess the greatest air moving capacity per horse power! Low fan tip speeds permit unusual quietness of operation. Work efficiently against resistance of duct systems. Have non-overloading power characteristic available in a wide range of sizes, speeds, and for all standard electric current.



"Power-Flow" Ventilators are aerodynamically correct in design. Note trim appearance—low height.

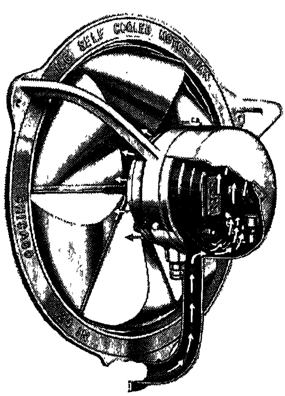
The above is only a partial list of the ventilating units DeBothezat builds. Our engineers will be glad to give you expert assistance in your ventilating problems—offering you a solution in space, weight and power saving equipment. Catalog on all products sent on request.

ILG Electric Ventilating Company

**Propeller Fans, Blowers, Unit Heaters,
Air Conditioning Equipment**

2880 North Crawford Avenue, **Chicago, Ill.**

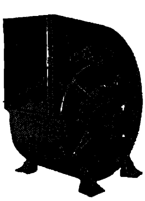
Offices in Forty-three Principal Cities



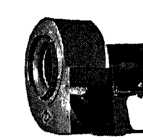
ILG PROPELLER FANS WITH SELF-COOLED MIRACLE MOTOR THAT "BREATHES"

Universally used for removal of stale air, fumes, heat, odors, etc. Rugged, heavy-duty framework stands up under years of continuous service. Accurate, electrically tested balancing of fan wheel assures quiet operation and smooth, effortless running, minimizing vibration and bearing wear, reducing maintenance and operating costs and increasing life of each unit. Direct connection of motor and fan avoids wasteful friction losses. Self-cooled motor combines protection of enclosed motor with low operating cost of open motor—constantly cooled by fresh, clean air, circulated internally—never "gums-up" from contact with foul air—saves 5 to 10 per cent on power costs. Every part of ILG units is designed, built and guaranteed by ILG, making possible the famous "ONE-NAME-PLATE" Guarantee. Sizes, 9½ to 72 in. Certified ratings. Write for Catalog F-482.

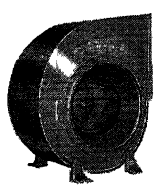
OTHER ILG PRODUCTS

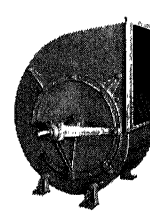


Ilg Type "B" Universal Blowers—Combine compactness, quietness and efficiency. Motor is recessed in side of blower, eliminating separate base. Multiblade wheel is direct connected to motor shaft. No inlet bearing. Available also for belt drive. Capacities 1750 cfm to 70,000 cfm, single and double width. Catalog EB-381.

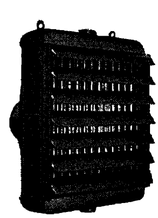


Ilg Type "B" Volume Blowers—A new design in small volume, low pressure blowers. Light weight, quiet running. Dynamically balanced multiblade wheel is direct connected to motor shaft. Motor and steel housing are supported by cast-iron base. Universal discharge. Available in 12 capacities, 180 cfm to 2100 cfm.

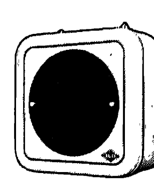




Ilg Type "BC" Universal Blower—Motor load remains constant over unusually large variation in air volume and static pressure. Type "BC" comes in direct connected drive (*left*); also belted drive with ball bearings (*right*). Blower wheel has backward curved blades riveted to side and back plates. Universal discharge. Available in 11 sizes.



Ilg Unit Heaters—Copper tube and fin construction and enclosed self-cooled motor. Ilg-built throughout. For steam or hot water. Tested with 500 lb hydrostatic pressure. Available in 26 capacities; also, Ilg self-contained gas or electric unit heaters for use where neither steam nor hot water is available. Catalog 128.

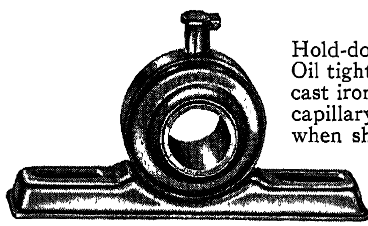


Ilg Cooling and Air-Conditioning Units—Ceiling type units used singly or in multiple with remotely located refrigerating machine circulating direct expansion "Freon" or methyl chloride; or with cold water. For cooling, dehumidifying and recirculating. Also, combination units for both cooling and heating. Write for catalog.

The Lau Blower Company

2007 Home Avenue, **Dayton, Ohio**

Manufacturers of Package Unit Furnace Blowers, Blower Wheels, Housings, Pulleys, Pillow Blocks, Complete Assemblies, Window Fans and Attic Fans

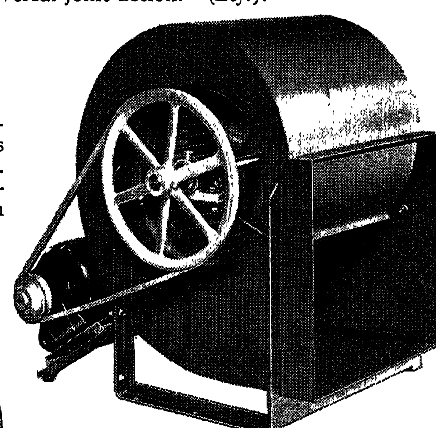


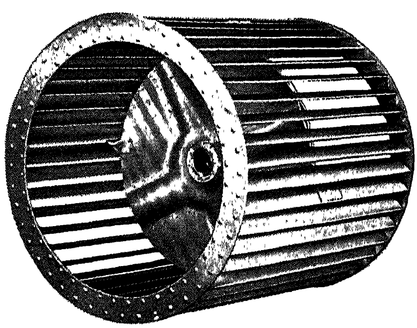
Self-Aligning Pillow Blocks

Hold-down bolts cannot affect the freedom of the bearing. Oil tight steel housing; reservoir holds twice as much oil as cast iron housings; Durex bushing feeds oil to the shaft by capillary action and maintains a constant oil film even when shaft is not rotating. Spherical surface of bearing conforms to contour of housing, providing a universal joint action. (*Left*).

100 Series Assembly

A complete blower assembly for manufacturers who fabricate their own casings (also available with top motor mounting). Variable speed drive; automatic belt-tightening device; automatic cut-out on motor; 23 sizes. (*Right*).



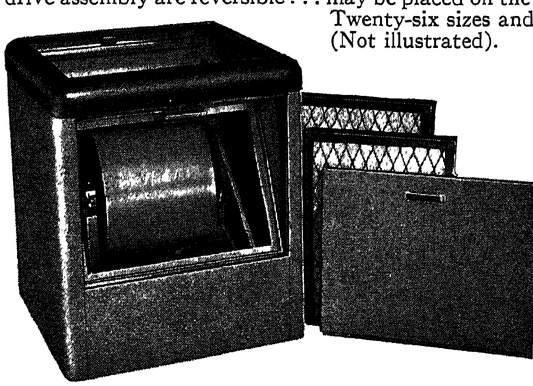


Blower Wheels and Housings

Squirrel cage, forward curve, multi-blade type wheels. Double inlet, double width (or single inlet, single width). Dynamically balanced; sizes 4½-in. to 30-in. Blower housings available in 10 standard sizes—special sizes on request. (*Left*).

700 Series Package Unit Blower

Steps-up efficiency of coal, gas and oil-fired furnaces. Complete with filters, blower cabinet, variable speed drive, blower and full size access doors on both sides. Motor and drive assembly are reversible . . . may be placed on the most convenient side—on the job. Twenty-six sizes and types; knock-down construction. (Not illustrated).



Blo-Ette Package Unit

A blower that will remedy the many thousands of existing unsatisfactory gravity jobs at a price so low that it is well within the reach of everyone. Top motor mounting; automatic cut-out on motor; leak-proof filter frames; 16 x 25-in. filters; high speed—low pressure; large size access door. Four sizes. (Illustrated at left).

Akron, O.
Albany, N. Y.
Atlanta, Ga.
Baltimore, Md.
Boston, Mass.
Buffalo, N. Y.
Camden, N. J.
Chicago, Ill.
Cincinnati, O.
Cleveland, O.
Columbus, O.
Denver, Colo.
Des Moines, Ia.
Detroit, Mich.
El Paso, Tex.
Greensboro, N. C.
Hartford, Conn.
Indianapolis, Ind.
Jacksonville, Fla.
Kansas City, Mo.
Little Rock, Ark.
Los Angeles, Cal.
Mansfield, O.
Memphis, Tenn.
Milwaukee, Wis.

B. F. Sturtevant Co.

Hyde Park, Boston, Mass.

Sturtevant REG. U. S. PAT. OFF.

Puts Air to Work

PLANTS LOCATED IN

CAMDEN, N. J. HYDE PARK, MASS. FRAMINGHAM, MASS.
STURTEVANT, WIS. GALT, ONT. BERKELEY, CALIF.

Minneapolis, Minn.
Montreal, P. Q.
Newark, N. J.
New York, N. Y.
Phoenix, Ariz.
Pittsburgh, Pa.
Portland, Ore.
Richmond, Va.
St. Louis, Mo.
Salt Lake City, U.
San Francisco, Cal
Seattle, Wash.
Spokane, Wash.
Springfield, Mass.
Syracuse, N. Y.
Toledo, O.
Toronto, Ont.
Washington, D. C.

A. M. Lockett & Co.
New Orleans, La.
Houston, Texas
Dallas, Texas
Galveston, Texas

The Cooling and Air Conditioning Division of B. F. Sturtevant Company

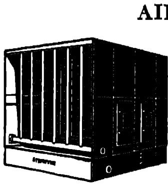
HYDE PARK BOSTON, MASS.

ATLANTA CAMDEN CLEVELAND GREENSBORO LOS ANGELES NEW YORK

DATA ON HEATING, VENTILATING, AIR CONDITIONING AND VACUUM CLEANING EQUIPMENT FOR ARCHITECTS, ENGINEERS, CONTRACTORS

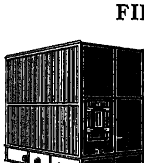
The Publications listed below, and on the following page, have been prepared to aid the architect, engineer and contractor in the selection of proper equipment for industrial, public, and private buildings of all types and sizes. We will gladly send copies upon request.

AIR WASHER



Built in several types to meet varying requirements in cleaning, cooling, dehumidifying and humidifying air.
Catalog No. 295.

FILTERWASHER

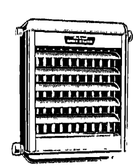


Airwasher employing filter pads in conjunction with water sprays. Humidifies and cleans air, removing all dust and dirt down to micronic fineness.
Catalog No. 453.

COOPERATION

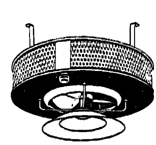
Sturtevant Engineers, located at each of the offices listed, are always ready to cooperate with architects, engineers and contractors in the selection of equipment suitable for any prospective installation.

SUSPENDED SPEED HEATERS



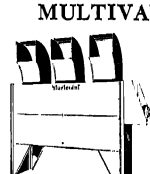
Propeller fan type, for wall or ceiling installation. Fin type heating element. For steam pressures up to 200 lbs, capacities up to 300,000 Btu.
Catalog No. 396.

DOWNBLAST SPEED HEATER



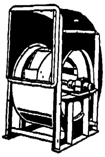
High Velocity unit heater for high ceiling installation. Fin type heating element. Steam Pressures up to 200 lbs. Capacities up to 400,000 Btu.
Catalog No. 454.

MULTIVANE SPEED HEATERS



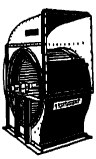
Centrifugal fan type for floor, wall, or ceiling installation. Fin type heating element. For steam pressures up to 200 lbs, capacities up to 2,010,000 Btu. **Catalog No. 452.**

SILENTVANE FAN



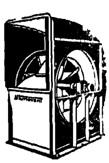
Backwardly curved blade type. A ventilating fan to meet the most exacting specifications, where very high efficiency and exceptionally low power consumption are required. **Catalog No. 451.**

MULTIVANE FAN



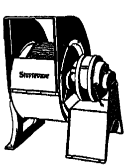
Forwardly curved blade type. A highly efficient centrifugal ventilating fan of sturdy construction to meet the general run of ventilating requirements. **Catalog No. 271.**

REXVANE FAN



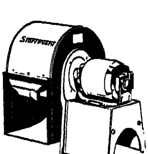
The modern paddle wheel. Correct inlet blade curvature and stream line shrouding retain all good features of the old paddle wheel and greatly increase capacity.
Catalog No. 414.

VOLUME FANS—CONVERTIBLE



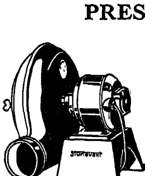
Silentvane, Multivane and Rexvane fans shown above can all be furnished with convertible and reversible housings, permitting quick conversion to any horizontal or vertical discharge.

REXVANE VENTILATING SET



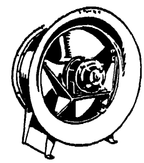
Direct connected; motor driven. For use with ducts, in ventilating small rooms, laboratory hoods and, in general, any room up to 10,000 cu ft contents.
Catalog No. 406.

PRESSURE BLOWER



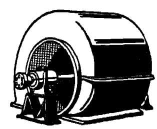
A small, compact forced draft fan, with direct connected electric motor, for coal burning heaters in schools and other buildings. **Catalog No. 297.**

PRESSURE FANS



Designed to operate against resistance of wind, ducts, etc., this rugged axial flow fan combines economical operation cost with high mechanical efficiency. **Catalog No. 444.**

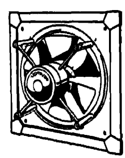
MECHANICAL DRAFT FANS



Forced and induced draft fans to meet any need. Can be furnished with Sturtevant reduction gears and steam turbine, motor or engine drive.

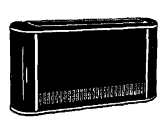
Catalogs No. 409, 445, 447, 448.

PROPELLER FANS



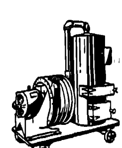
Built in fan sizes from 12 up to 45 in. Electric motors are available for both alternating and direct current, in wide variety of voltages. **Catalog No. 400.**

UNIT VENTILATOR



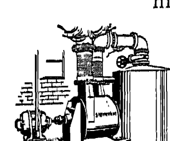
A combined steam heating and ventilating unit widely used in schools and other places. Finished in duco of any standard color. Stainless steel trim. **Catalog No. 377.**

VACUUM CLEANERS



Portable

Sturtevant Vacuum Cleaners are made in both portable and stationary Central System types and in a variety of sizes to meet every school building need. **Features:** Quiet operating vacuum producers, high suction for rapid cleaning under all conditions; tools designed individually for specific cleaning operations. Central Systems include piping recommendations to insure correct operation of the system.



Central Systems

Catalog No. 413 (Portable Cleaners).
Catalog No. 397 (Central Systems).

The NEW YORK BLOWER COMPANY

GENERAL OFFICES
32nd STREET & SHIELDS AVENUE · CHICAGO
FACTORIES at LAPORTE, IND and CHICAGO, ILL.

Experienced Air Engineers

Representatives in Principal Cities

BUYERS' KEY TO Products

Products . . . for all industrial processes, ventilation and heating. Specify them for every industrial, commercial and institutional requirement. Check these nine classifications which give thumb nail specifications:

- **CENTRIFUGAL FANS AND BLOWERS**
Type ME Housed Centrifugal Wheels—Quiet Operating, Slow Speed Type and/or High Speed Wheels with Non-overloading Horsepower Characteristics. Range of Standard Wheel Sizes 15 in. to 80 in. Wheel Diameter. Sizes 15 inch to 80 inch Wheel Diameter.

- **JUNIOR CENTRIFUGAL BLOWERS**
Type ME Junior Fans, direct connected Motor Driven. Range of Sizes 6 inch to 12 inch Wheel Diameter.

- **DISC TYPE (or PROPELLER) PANEL FANS**
Comet EXHAUSTAIR Ventilating Fans, Automatic Shutters, Power Roof Ventilators, direct connected Motor Driven. Size 10 inch to 30 inch Wheel Diameter. Heavy Duty Type, Pulley Driven, GIANT Disc Type Fans, Regular Sizes 36 inch to 108 inch Wheel Diameter with Round Body Frames.

- **INDUSTRIAL UNIT HEATERS**
Disc or Propeller Fan, Ceiling Suspension Type, NYBCo COMET Unit Heaters with Molybdenum Alloy Corrosion-Proof and Freeze-Proof, Extra Heavy Welded, Longlife Ferrous Heating Element. Suitable for low or high-pressure (unlimited) Steam Pressures. Capacities 24,000 to 300,000 BTU's per Hour. Excel AIR-FLOW Centrifugal Type Factory Unit Heaters. Blower-Type Unit Heaters with Encased Centrifugal Fans with NYBCo Molybdenum Alloy Welded Steel Heating Element (either Blow-through or Draw-through Type). Floor Type, Side-wall or Ceiling Suspension. Capacities 169,000 to 1,000,000 BTU's per Hour.

- **MECHANOVENT UNIT VENTILATORS**
CLASSROOM Unit Ventilators, highly refined in design and appearance. A De-Luxe Product in every sense. Suited for fully Automatic Temperature and Humidity Regulation. Capacities 750 CFM to 1560 CFM for Classroom Duty.

- **MECHANOVENT UNIT VENTILATORS (Continued)**
AUDITORIUM Unit Ventilators, Fully Encased Centrifugal Type Units, with or without Fresh Air and/or Recirculation Damper Assemblies, for use in Auditoriums or other places of large public gatherings. Capacities 2,000 CFM to 10,000 CFM.

- **AIR WASHERS**
A completely engineered line of PEERLESS Air Washers, Air Cleansing, Air-Conditioning and Cooling. Complete with Single- and Double-bank Atomizing Spray Systems, Marine Type Doors, Eliminators, Entering and Back-spray Louvres, Water Strainers, Pumps and Motors, and with or without Humidity Flooding Provisions. Sizes and Capacities ranging from 3,600 CFM to 76,000 CFM.

- **HOT BLAST HEATING SURFACE**
NYBCo "STEELFIN" Longlife High Pressure Molybdenum Alloy Steel, All Welded, Extra Heavy Duty, Homogeneous Fin-and-Oval-Tube Hot-Blast Heating Surface. Hot-dip Overall Metallic Coating, Including Headers. A Super-quality Product—proof against faults common to Surfaces constructed of Non-ferrous and Cast Iron Materials. An Engineered Product of Sizes and Capacities for Steam Pressures (or Hot Water Equivalents) from 2 lbs. to 150 lbs. duty, High Temperature or Low Temperature Fin Spacings, and a Range of Air Velocities from 400 to 1000 Ft. per Min.

- **VENTO, AEROFIN, and OTHER HEATING AND COOLING SURFACES**
The various types and makes of Heat Transfer or Heat Exchange Surface, as regularly sold through the outlets of manufacturers of Fan-system Apparatus. These types of Surface are offered in various combinations and sizes, together with full and complete engineering recommendations.

- **MISCELLANEOUS**
Dust and Shavings Exhausters and Material Conveyor Blowers (Centrifugal Fans with Special Housings and Wheels); Engines, Motors, and V-Belt and Other Drives; Air Filters—Automatic and Cartridge or Renewable Types; Control Devices, including Pressurestats, Thermostats, Humidistats; Turbine Ventilators; Gas-fired Unit Heaters; Specialties; and Other Apparatus for use in conjunction with Complete Blower Systems. Complete data and descriptive matter furnished on specific request.

MEMBER NAFM AND IUHA

Certified Ratings

Write for Catalogues

Full Catalog Matter, Descriptive Bulletins, Performance Tables, Engineering Data and Technical Presentation, Prices, and Complete Information with Illustrations will be furnished upon request of District Representative or by Home Office.

BUILT RIGHT RUN RIGHT

Since 1889

The Torrington Mfg. Co.

50 Franklin Street, **Torrington, Conn.**

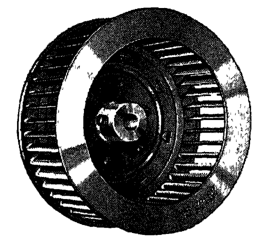
Manufacturers of Blower Wheels and Propellor Type Fan Blades.

AIRISTOCRAT *Quiet Propeller Fan Blades* AIROTOR *Blower Wheels* AUTOCRAT *Fan Blades*

Torrington Aluminum Blower Wheels produce the smooth, quiet performance which is essential in modern heating and air conditioning units because the unique patented construction breaks up resonance and minimizes noise. Made of aluminum, they resist corrosion and their light weight facilitates quick starting—saves power. Every wheel is statically balanced.

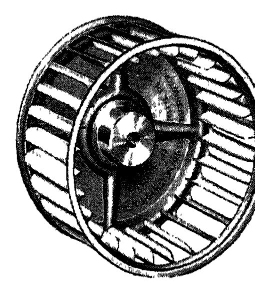
Bulletin lists 34 sizes of single inlet single width and 34 sizes of double inlet double width wheels, including guaranteed capacities for each. Also gives detailed dimensions for all wheels and table of dimensions for housing scrolls. We do not manufacture housings.

Sizes 3 in. to 15 in. diameter in all standard widths.



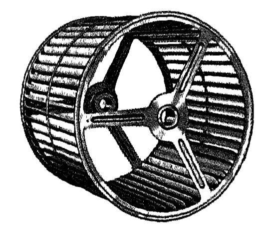
Single Inlet Aluminum Blower Wheel

Torrington Airotor Blower Wheels are light, sturdy and inexpensive—incorporate new principles of design and construction, which insure rigidity and concentricity. **Single Width—Single Inlet** wheel is of simple four-piece construction. **No rivets or welds are used;** concentric rib serving as backing for blade strip is formed at same time as hub socket, insuring trueness of wheel. Rigid radial ribs prevent deflection by thrust. Three thicknesses of metal in rims make for maximum strength. Manufactured in both aluminum and steel in 4½ in., 5 in., 6 in. and 9 in. sizes. Same size available in DA type double width, double inlet wheels.



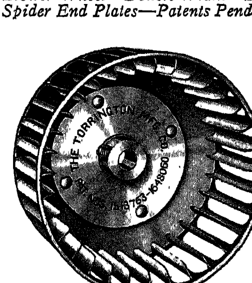
Airotor Blower Wheel—Single Width—Single Inlet
Patents Pending

Torrington Airotor Blower Wheel—Double Width—Double Inlet, Spider End Plates has blades punched and formed in a single strip, rigidly held by flanged single piece end rings. Hubs are rigidly mounted by peening. Wheels of 3⅝ in., 10½ in., 12 in. and 16 in. diameter are available at present; 4½ in., 5 in., 6 in., 7½ in., 9 in. and 20 in. sizes are being developed.



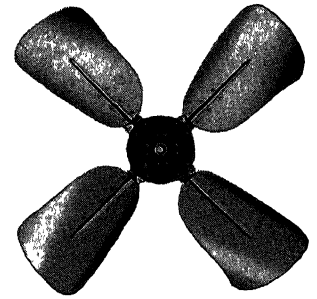
Airotor Blower Wheel—Double Width—Double Inlet
Spider End Plates—Patents Pending

Torrington Cup Type Wheels—Used for automobile heaters, gun type oil burners, windshield defrosters, small hair dryers, hand dryers, ice box and refrigerator circulators, window ventilators, exhausters, etc. Made for either clockwise or counter clockwise rotation, of steel, in the following sizes: 3 in. to 9 in. inclusive.



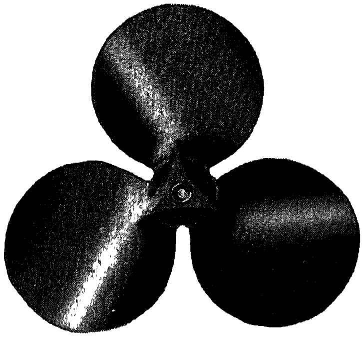
Cup Type Blower Wheel. Pats. 1,513,763; 1,648,060

AIRISTOCRAT "A" Series Attic Fan Blades are the result of extensive study and experiment to produce blades having extraordinary efficiency, to sell at lower than average prices. LOW COST is possible because tools are interchangeable for production of either 2, 3 or 4 blade models in any diameters from 24 in. to 48 in. inclusive (sizes 24 in., 30 in., 36 in., 42 in. and 48 in. are standard). Construction approved only after severe breakdown tests. The extremely high EFFICIENCY is attained by the application of correct principles of design. Blades, spiders and hubs are of steel. Available in the following finishes: 1. Plain. 2. Aluminum lacquered blades, black lacquered spider and hub. 3. All one color lacquer. Bulletin gives detailed dimensions and specifications; also performance data.



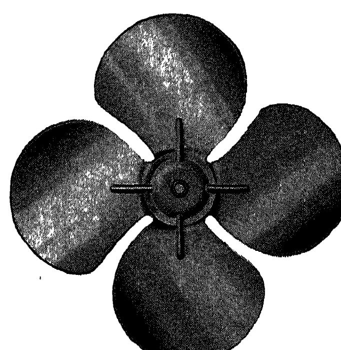
4-Blade Airistocrat Attic Fan "A" Series (also made in 2 and 3 blade models)

3-Blade "One-Piece" Series Propellor Fan—An attractive, inexpensive one-piece blade incorporating the **Airistocrat** features for quiet operation. Available in both steel and aluminum. Sizes 8 in. and 10 in. diameters.



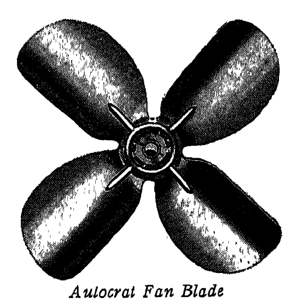
Airistocrat
3-Blade "One Piece" Series

4-Blade "One-Piece" Series Propellor Fan—An exceptionally rigid model blanked from one piece of metal with four wide blades. Quieter than narrow blade types. Made in both steel and aluminum. Clockwise rotation only (viewing air delivery side). Sizes 8 in., 9 in., 10 in. and 12 in. diameters. Available in the following finishes: 1. Plain. 2. Lacquered. 3. Nickel or cadmium plated (steel only).



4-Blade "One Piece" Aristocrat Fan

AUTOCRAT Fan Blades—For auto heaters, windshield defrosters, etc. Have been standard ever since these devices were first marketed. Made in sizes 3 in., 4 in., 4½ in., 5 in., 5¼ in., 5½ in., 6 in., 6½ in., all four blades, also 7 in. 5-blade, in one piece of cold rolled steel or aluminum with brass hubs, complete with set screw. ¼ in. bore is standard. Either clockwise or counter clockwise rotation (expressed when looking at air delivery side of fan). White nickel is standard finish for steel blades. Bulletin gives complete specifications and ratings.



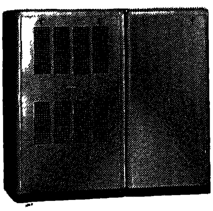
Autocrat Fan Blade

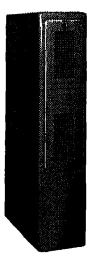
Utility Fan Corporation

4851 S. Alameda St., **Los Angeles, Calif.**

Utility Gas-Fired Heating Equipment, Evaporative Coolers, Blowers and Fans

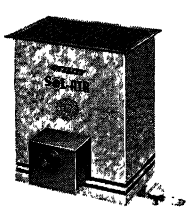
FORCED AIR FURNACES





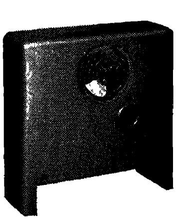
Basement and closet types . . . Compact design . . . Multiple-fin element with air-cooled hollow baffle . . . Element guaranteed against burnout forever . . . Automatic controls . . . Filters . . . Built-in motor overload protection . . . Dynamically balanced blowers . . . full-floating blower and motor.

FLOOR FURNACES



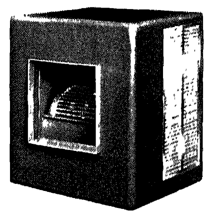
All-welded construction . . . Die-stamped grilles . . . heavy cast-iron burner . . . removable inside jacket . . . interlocking gas valve. Floor or dual registers. 25,000, 37,000 and 50,000 Btu input.

CIRCULATING HEATERS



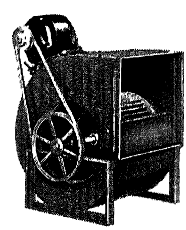
Fan sends stream of air through nozzle-shaped outlet to hold warm air in living zone. Built of heavy furniture steel . . . all die-formed and electric welded. Vented and unvented models.

EVAPORATIVE COOLERS



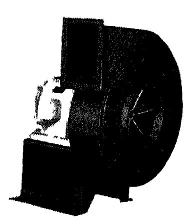
Comfort cooling—residential, commercial, industrial. Exlcusive feature—Uni-flowmeter (Pat. Pend.) for uniform water distribution. Patented No-Sag cooling pads. 14 Models.

BLOWERS



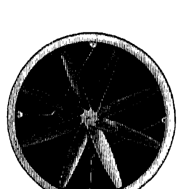
Complete line. Dynamically balanced, multiple-vane centrifugal blowers—standard and heavy duty. Wheel diameters 6 to 66 in.; widths 6 to 66 in.

EXHAUSTERS



High flow and high pressure designs in four drive arrangements, for exhausting many materials. High efficiency . . . decreased sound. Available with acid-proof plastic surfacing.

PROPELLER FANS



Airplane propeller type for high efficiency, economy and maximum air delivery. 2, 4 and 8 blade models—12 in. to 30 in. diameter; 300 to 250,000 cfm.

Utility Blowers are tested in accordance with the A.S.H.V.E. Code.

Write for complete information, catalogs and prices.

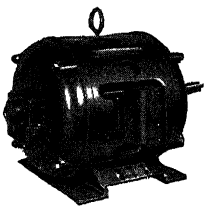
Wagner Electric Corporation

6400 Plymouth Avenue, **Saint Louis, Mo., U. S. A.**

Wagner motors are built in a wide range of mechanical and electrical types to meet the varied requirements of the air-conditioning industry. These motors are carefully designed and skillfully constructed to make them quiet in operation and completely reliable.

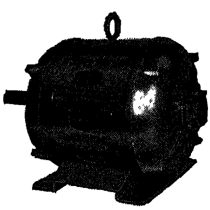
WAGNER POLYPHASE MOTORS

Single-Speed Squirrel-Cage Motors



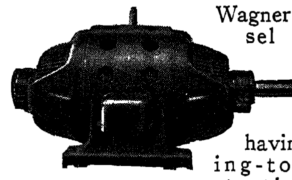
Normal starting-torque—normal starting-current for driving radial compressors, fans and blowers. High starting-torque—low starting-current for driving reciprocating compressors. 2 and 3 phase; ⅙ to 400 hp.

Multi-Speed Squirrel-Cage Motors



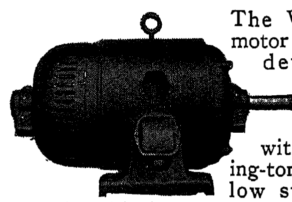
To use where several constant speeds reduce operating costs. Can be furnished with normal starting-torque or high starting-torque, with two, three or four speeds. Variable torque characteristics for fan service, constant torque characteristics for compressor service. 3 phase; ½ to 125 hp.

Fynn-Weichsel (Synchronous) Motors



Wagner Fynn-Weichsel motors are leading-power-factor motors, having high starting-torque, low starting-current, high pull-in torque, constant speed at all loads, and ability to carry heavy intermittent overloads. Especially desirable where power-factor improvement or constant speed is required. 2 or 3 phase; 7½ to 200 hp.

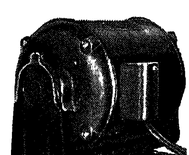
Special Compressor Motors



The Wagner RT motor was specially developed to meet the demand for a motor with high starting-torque and very low starting-current. 2 and 3 phase; 40 to 150 hp.

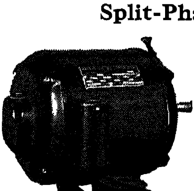
WAGNER SINGLE-PHASE MOTORS

Repulsion-Start-Induction Motors



Brush-lifting (assures quiet operation, long brush and commutator life). For high-starting-torque heavy-duty applications. Open, totally-enclosed, and drip-proof; rigid or resilient-mounted. ⅛ to 15 hp.

Split-Phase Motors

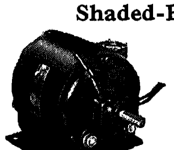


Long-life switch and unbreakable steel frames. Open, drip-proof and totally-enclosed; rigid or resilient-mounted. 1/20 to ⅓ hp.

Capacitor Motors

Condenser-start induction-run, ⅛ to ¾ hp; drip-proof or totally-enclosed end-plates; rigid or resilient-mounted. Condenser-start condenser-run, 1/20 to 1 hp; single or multi-speed for direct connected fans and blowers.

Shaded-Pole Fan Motors



Single-phase induction of simple construction requiring no complicated starting equipment, ideally adapted to fan and blower drives in which the fans are mounted directly on the motor shaft. Totally-enclosed and open type, rigid base round frame, or resilient mounted, with or without 3-speed regulator. 1/250, 1/125, 1/80, 1/60, 1/40, and 1/30 hp ratings. M40-31

GENERAL ELECTRIC COMPANY

SCHENECTADY, N. Y.

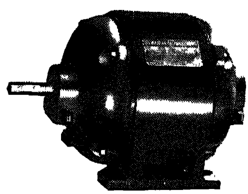
SALES OFFICES, WAREHOUSES, SERVICE SHOPS AND DISTRIBUTORS IN PRINCIPAL CITIES

For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc., Address—APPLIANCE AND MERCHANDISE DEPARTMENT, BRIDGEPORT, CONN.

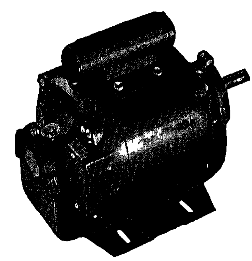
HEATING, VENTILATING, AND AIR-CONDITIONING MOTORS

The complete line of motors manufactured by the General Electric Company offers you a motor with electrical and mechanical characteristics best adapted to your compressor, fan, or pump application. The most frequently used applications are listed below. Complete information on other types of motors, vertical, enclosed, etc., with various electrical and mechanical modifications, may be obtained from our nearest sales office.

A complete line of motors, designed and tested especially for quiet operation for use in schools, hospitals, commercial buildings, and also a complete line of special sound-isolating bases for these motors are available when using V-belt drive.



TriClad induction motor. Type K, polyphase



Capacitor fractional-horse-power motor. Type KC

SOME G-E MOTORS AND THEIR USES

Application	Speed	Type Winding	Type	Horsepower Range	Classification
Fans and Centrifugal Pumps	Constant or Adjustable	Shunt	B & CD	1/8-200	Direct Current
Reciprocating Pumps and Compressors		Compound	B & CD	1/8-200	
	Constant	High Torque Capacitator	KC & KCJ	1/4-3	Single Phase Alternating Current
Small Direct Connected Fans	Constant	Resistance Split Phase	KH	1/40-1/3	
		Reactance Split Phase	KX	1/6-1/3	
	Constant or 3-Speed	Low Torque Capacitor	KC	1/50-10	
Belted Fans, Centrifugal Pumps		General Purpose	KC	1/4-10	
		Capacitor	KC	1/8-10	
		Repulsion Induction	SCR	1/8-10	
	Constant or Multispeed	Squirrel Cage	K or KB	1/4-1000	Polyphase Alternating Current
		(Low Starting Current)	KF	7½-75	
Reciprocating Pumps and Compressors		(High Starting Torque)	K & KG	Y 4-100	
Pumps, Compressors, Fans	Constant or Adjustable	Wound Rotor	M & MB	½-1000	
	Constant	Synchronous	TS	25-2000	

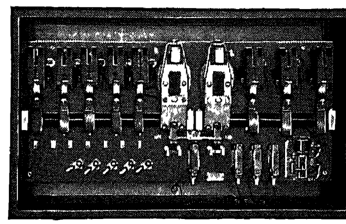
This Company will gladly assist in the solution of any electrical problems in relation to heating and ventilation

GENERAL ELECTRIC COMPANY
SCHENECTADY, N. Y.

SALES OFFICES, WAREHOUSES, SERVICE SHOPS AND DISTRIBUTORS IN PRINCIPAL CITIES

For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc., Address—APPLIANCE AND MERCHANDISE DEPARTMENT, BRIDGEPORT, CONN.

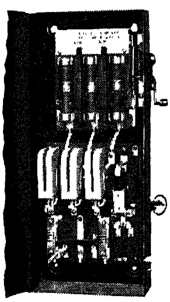
CONTROL FOR HEATING, VENTILATING, AND AIR-CONDITIONING MOTORS



CR7107 controller (cover removed) for use with multispeed Squirrel cage motors

The General Electric line of standard control offers manual or automatic equipment for compressors, fans, or pumps driven by any type motor which you require, providing full protection for your motor, especially those listed on the preceding page.

For special applications General Electric controllers can be designed to meet your exact requirements.



CR 7008—Magnetic Control for induction motors, with cover removed

CR 7008 provides these features:—

(1) Fuses or a circuit breaker protect against short circuits,

(2) Undervoltage protection prevents unexpected restarting after a voltage "dip," and

(3) Isothermic overload relays protect the motor but prevent unnecessary shutdowns.

The following control equipments are applicable to all motors on the preceding pages:

Full-voltage automatic starters for thermostatic control of fans, or pumps.

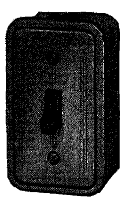
Automatic reduced- or full-voltage starters for synchronous motors driving compressors.

Manual or automatic speed-regulating controllers for wound-rotor motors driving fans.

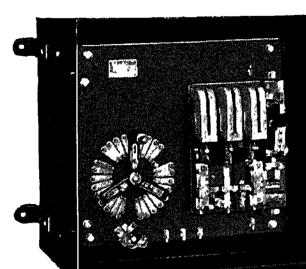
Manual full-voltage starters for pump motors.

Manual speed-regulating switches for small capacitor motors driving fans.

CR7006—full voltage magnetic switch for use with induction motors



CR1061 fractional-horsepower-motor starting switch for wall mounting



CR7764 Speed regulating controller for wound rotor motor (cover removed)

ACCESSORIES

Thermostats.
Float switches.
Electrically operated valves.
Pressure switches.
Indicating Push buttons.

Motors and control of one manufacture insure perfect operation and good service, and simplify installation.

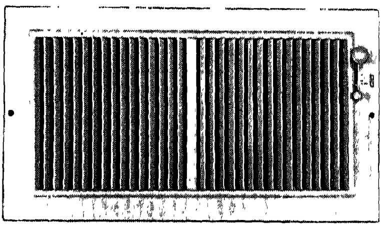
This Company will gladly assist in the solution of any electrical problem in relation to heating and ventilation

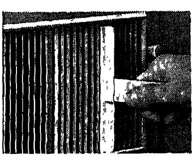
Air Control Products, Inc.

Muskegon, Michigan

AIR CONDITIONING REGISTERS AND GRILLES
ADJUSTABLE GRAVITY REGISTERS
FLOOR REGISTERS AND FACES
ATTIC-LOUVERS ● DAMPER CONTROL SETS

Air Conditioning Registers

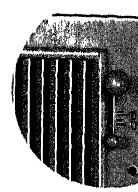


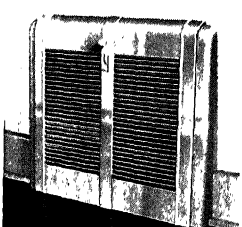


Each fin may be easily adjusted, even after the registers are installed, by means of a special tool that is provided.

No. 10 Series Registers assure uniform air distribution for they offer a combination of both vertical and horizontal control of the air stream. Adjustable vertical fins plus adjustable horizontal louvers—immediately behind the register face—give positive air deflection in both planes. A small stop below the operating knob can be set for any deflection from 15 degrees upward (for cooling) to any desired downward deflection (for heating). See illustration below. This easy adjustment also makes the No. 10 Register suitable for Year-'Round Air Conditioning. Registers are regularly supplied in a tough, moisture resistant buff priming coat and are equipped with a sponge rubber gasket seal. This design furnished in baseboard and sidewall types—also available as grille without horizontal valve.

No. 110 Series Registers—an attractive low-priced register with a single shutter valve mechanism. A quality register in every detail—adjustable vertical fins provide directional flow—sponge rubber gasket eliminates streaked walls. The close similarity in appearance to the No. 10 Series makes it possible to use both on the same installation.





Adjustable Gravity Type Registers

No. 20 Registers offer modern styling; ample free area, for gravity installations, and are adjustable for forced air. A universal register—efficient for new gravity installations or for converted forced air installations. Self-sealing sponge rubber gasket, supplied on all registers, makes an air tight seal between wall and register. Beautiful Metalescent Finish furnished as standard. Also available in sidewall types.

Floor Registers and Faces are of grid type construction—resulting in a rigid floor register with ample free area and heel-proof mesh; shallow valve is also adaptable for sidewall installations. Available in a complete range of sizes.

Attic Louvers for attic ventilation and **Damper Control Sets** for forced air dampers are also included in the complete AIR CONTROL Line.

Write for catalog giving complete information on **Air Control's** entire line of products.

Anemostat Corporation of America

10 East 39th Street, **New York City, N. Y.**

THE ANEMOSTAT HIGH VELOCITY AIR DIFFUSER

The Anemostat High Velocity Air Diffuser is a ceiling outlet consisting of a series of circular diverging metal cones opening outward from a central circular neck which may be attached directly to the main or branch duct.

The Anemostat assures draftless distribution of air at any duct velocity. Various standard sizes from 2 in. to 38 in. neck diameter will distribute volumes of air between 10 cfm and 15,000 cfm and will handle any velocities between 300 fpm and 2000 fpm. When introducing large quantities of air into a room air motion results. The series of cones which form the Anemostat discharge the air in definite proportions in all directions in a series of planes. This diffusion together with the aspiration (suction) effect causes prompt horizontal equalization of temperature and humidity throughout the room, and definitely prevents air pockets and dissipates the evaporation aura around the human body.

The air-mixing effect of the Anemostat causes the predetermined room temperature to be established at a point well above the breathing level, which permits the use of higher temperature differentials. This results in smaller volumes of air to be conditioned which in turn results in smaller plants, reduced operating expenses and smaller ducts. The high velocities which may be employed because of the draftless diffusion, result in further reduction of duct sizes and simplification of duct layouts.

At the different velocities recommended for rooms used for different purposes the increase in decibel ratings through the use of Anemostats is negligible.

The Anemo-lite which is an Anemostat combined with a built in lighting unit is an ideal solution to the combined problem of Air Distribution and Lighting. (See Figure 1).

Pendent lighting fixtures may be hung directly from the center cone of the Anemostat if desired. (See Figure 2).

Particularly suitable for theatres and auditoriums is the Anemostat combined with the indirect lighting unit. (See Figure 3). With this combination unusual and effective results are easily obtained.

Complete technical information on the Anemostat combined with lighting fixtures is available upon request.

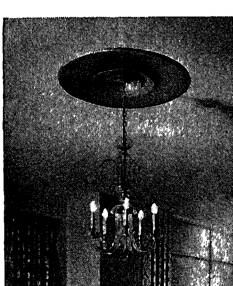


Fig. 2

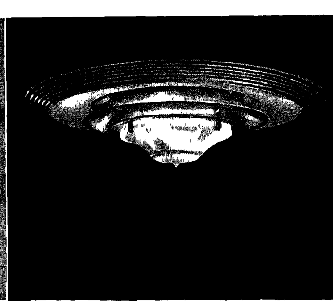


Fig. 1

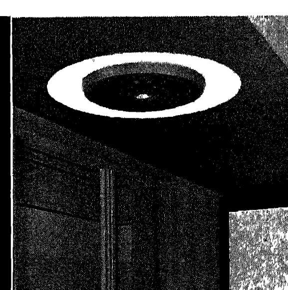


Fig. 3

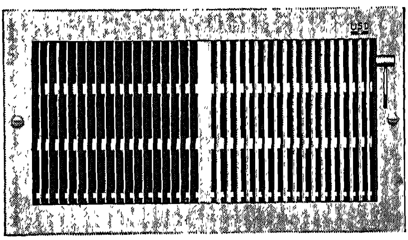
"No Air Conditioning System is better than its Air Distribution"

The Auer Register Co.

3608 Payne Avenue, **Cleveland, Ohio**

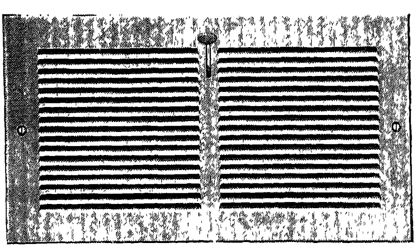
Manufacturers of Registers and Grilles for Gravity and Air Conditioning Systems; Wrought Metal Grilles for Concealing and Protecting Radiation

AIR CONDITIONING REGISTERS AND GRILLES



Airo-Flex No. 4432 Register—No Frame

Multi-louvres adjustable up, straight or down. Grille bars adjustable for right or left flow. Adjusting tool furnished. Grille to match.



Airo-Flex No. 7032 Register—No Frame

Grille bars set to direct air downward at 22½ deg., but adjustable for other angles. Single louvre. Grille to match.

The Auer line of registers and grilles for heating and air conditioning systems is modern and complete, offering a wide choice of styles for every purpose. Only a few representative models are shown on this page.

Airo-Flex "4000" Registers have *vertical* grille bars adjustable with tool, as shown, for combination or single current, straight, right or left. For up-and-down flow, Multi-louvre Back Blades, operated by lever, direct current at desired angle up, straight, or as much as 22½ deg. down.

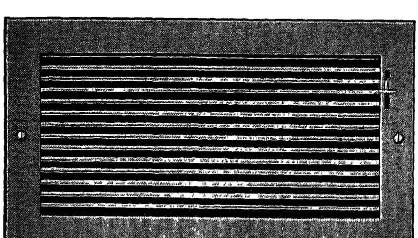
Airo-Flex "7000" Registers have *horizontal* grille bars adjustable in same manner as "4000" and are equipped with *single* blade louvre. A high quality *economy* register.

Fin-Flex Registers and Grilles are made with either vertical or horizontal fins which are easily adjusted at time of installation for single or multiple air current in any direction.

Dura-Flex Register and Grilles are also furnished with blades adjustable for any desired air flow. Other Dura-Flex designs are available with fixed blades.

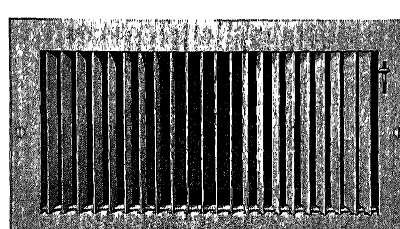
DuraBilt Floor Registers and Cold Air Faces are assembled with steel cross-bar construction, all cross joints locked and mortised. These should be specified wherever extra strength is required. They come in medium or narrow mesh.

All Auer models are designed with due regard for air capacity, and supplied in all required sizes and finishes. Complete Catalog 40, showing all types for air conditioning and gravity heating, furnished on request.



Fin-Flex No. 5030 Register with Band Iron Frame

Flexible fins ⅜ in. on center offer satisfactory one-time adjustment. Adjusting tool furnished with every order. Horizontal fins, furnished as standard. Vertical fins furnished if specified. Same design furnished also without valve, as a return.



Dura-Flex No. 8132 Register—No Frame

Adjustable bars ½ in. on center.

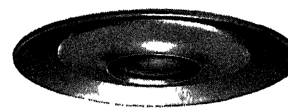
Also furnished with horizontal bars (adjustable). Small, convenient adjusting tool furnished with each order. Same design furnished also without valves, as a return.

Barber-Colman Company

Rockford, Illinois

ENGINEERED AIR DISTRIBUTION OUTLETS

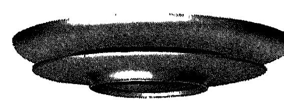
Venturi-Flo



Venturi-Flo—Flush Type

Venturi-Flo is a ceiling type outlet of modern design which mixes conditioned air and room air and distributes it over the desired area. The units are made in flush and surface types and in sizes to permit handling volumes of air from 6 cfm to 15,000 cfm.

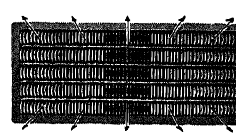
Comparatively high velocities may be used without increasing the noise level or causing objectional drafts in the room.



Venturi-Flo—Surface Type

Units are available in standard finishes. Either type may be obtained with adjustable dampers or in combination with lighting fixtures.

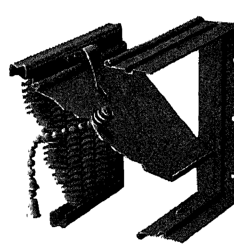
Uni-Flo



Uni-Flo Grille

Uni-Flo grilles and registers are designed and prefabricated with directional flow aspirating fins especially for air conditioning installations. Therefore **Uni-Flo** grilles and registers assure proper air distribution and obviate the necessity of adjustment at the installation.

Dimensions and core arrangement may be selected to give desired directional flow and throw without increasing the noise level or causing drafts. Various sizes and shapes are available including curved surfaces.

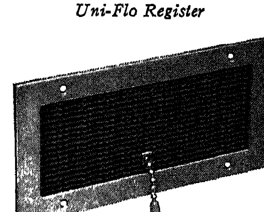


Uni-Flo Register

Registers are similar in construction to grilles, but with the addition of spring loaded, positive closing, chain or key-operated dampers.

Electroplated Finishes: Gunmental, brushed bronze, plain zinc, buffed zinc, and satin copper. Also available in plain metal, gray prime coat, or clear lacquer.

Uni-Fin

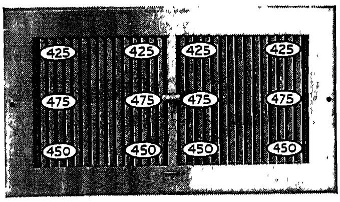


Uni-Fin Register

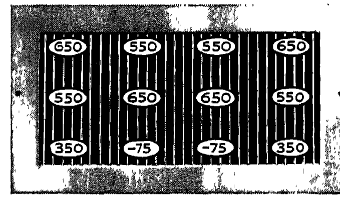
Uni-Fin grilles and registers are designed especially for residential warm air installations. Available in standard sizes and prime coat or electroplated finishes.

THE CORRECT TYPE OF OUTLET FOR EVERY REQUIREMENT

(*See also Page 940*)



Velocities with No. 75 Design



Velocities with Conventional Register

EVEN DISTRIBUTION OF AIR OVER ENTIRE FACE

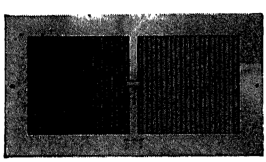
The turning blade valve distributes the air evenly with a uniform velocity over the entire face, as shown in Figs. 1, 2, and 3 on the preceding page. Note how the air rushes through the upper part of the face with a conventional register, as shown in Fig. 4. Since the entire face of No. 75 Design register is utilized for discharge of air, smaller and in some cases fewer registers can be used without causing excessive velocities.

Prevention of Streaked Ceilings—With either UP, STRAIGHT, OR DOWN deflections the air does not strike the ceiling immediately in front of the register; streaked ceilings are thus avoided.

Excellent Concealment of Duct—The depth and close spacing of the vertical bars, combined with the valve, provide almost complete concealment of the duct, adding considerably to the pleasing appearance of the register face.

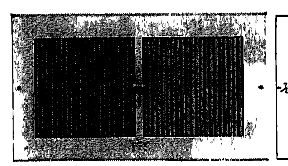
Special Settings—No. 75 Design functions equally well when located at the end of a horizontal duct or, by installing it upside down, when the air is delivered to it from above.

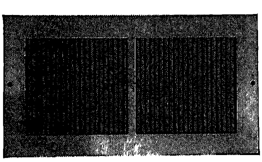
AVAILABLE IN FOUR TYPES



With Turning Blade Valve

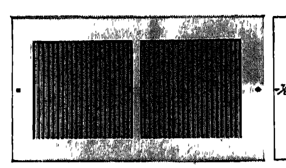
No. 751 Register (Left) has Sponge Rubber Gasket and $\frac{3}{16}$ in. turndown. No. 754 Register (Right) is similar except has $\frac{7}{8}$ in. projection.



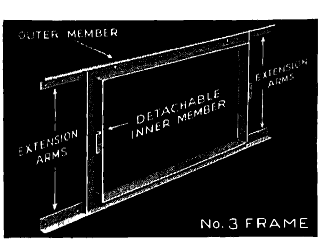


Without Valve

No. 750 Grille (Left) has Sponge Rubber Gasket and $\frac{3}{16}$ in. turndown. No. 757 Intake (Right) has $\frac{7}{8}$ in. projection.



FOUR TYPES OF INSTALLATION FRAMES AVAILABLE



No. 75 Design items can be used with or without installation frames. No. 3 Sidewall Stud Frame (illustrated), fastens directly to stud, forming a solid, streak-proof foundation for register. No. 8 Frame is similar for baseboard use. No. 5 Baseboard Stack Frame provides inexpensive, streak-proof installation. No. 2 Band Iron Frame provides for connecting register to stackhead.

CATALOG 40 AC, containing 10 pages of useful engineering data and showing the complete H & C line, available upon request.

Hendrick Manufacturing Company

Hendrick Perforated Metal Grilles

48 Dundaff Street, **Carbondale, Pa.**

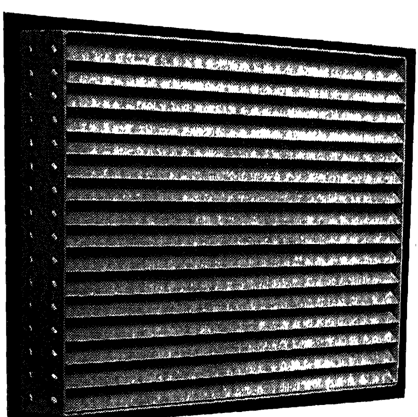
SALES OFFICES IN PRINCIPAL CITIES—CONSULT TELEPHONE DIRECTORY

PRODUCTS—Hendrick Perforated Metal Grilles; also Mitco Open Steel Flooring, Mitco Armorgrids and Mitco Shur-Site Treads.

Hendrick Perforated Metal Grilles

The Hendrick line of perforated metal grilles is a large and varied one, offering the architect, contractor, building owner, etc., literally hundreds of designs from which to select the grille or grilles best adapted to specific installations. In addition to standard patterns, Hendrick offers a number of exclusive grille designs, many of them covered by design patents.

All Hendrick Grilles are characterized by clean-out perforations and fine finish.



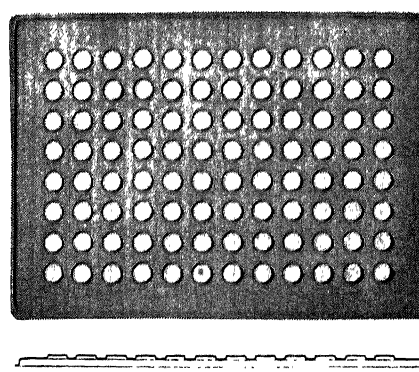
FIXED LOUVRE GRILLE

Hendrick Fixed Louvre Grille; ideal for doors in hospitals, hotels, bathrooms. Permits free circulation of air but prevents vision through the grille from any angle. Easily installed in any door. Regularly furnished in No. 18 U. S. Gauge Steel. Can be furnished in aluminum, bronze, or 0.05 in. thick stainless steel.

La Crosse

They are given a special flattening operation which makes for easy installation.

Hendrick Perforated Grilles are furnished in aluminum, brass, bronze, copper, Everdur, Monel, nickel-silver, stainless steel, steel, zinc and other commercially rolled metals. They are supplied unpainted, or with prime coat; with lacquer or duco finish in any color; with natural polish or with any standard electroplate finish. Furnished from 16 gage to 5/16 in. thick, up to 90 in. wide and almost any length. They come with invisible access doors, angle frames, hinges, etc.



NOZZLE GRILLE

For air conditioning systems utilizing high velocities; most efficient in minimizing the noise of air passing through a grille opening. Hendrick Nozzle Grille is generally furnished with 5/8 inch diameter perforations but can be had in a large variety of sizes, in aluminum, bronze, stainless steel, and steel, in gauges up to .078 in. thick and in sizes up to 48 in. by 120 in.

Design Patent No. 89,624.

Write on your letterhead for a copy of 194-page handbook; "Hendrick Grilles."

Plandaire, Inc.

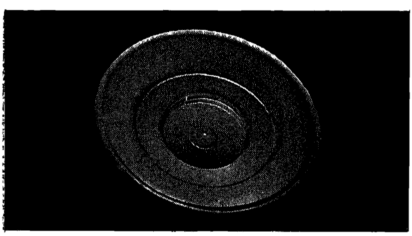
3223 Kennett Square, Box 7350, **Pittsburgh, 13, Pa.**

Sales Representatives

Baltimore, Md.....W. I. Collier & Company, 522 Park Ave.
Chicago, Ill.........Air Products Company, 9 S. Kedzie Ave.
Cincinnati, Ohio, Allonier Engineering Co., Union Central Bldg.
Cleveland, Ohio.................F. K. Minor Co., Bulkley Bldg.
Des Moines, Iowa............Electrol of Iowa, 533 Seventh St.
Detroit, Mich., Louis T. Ollesheimer, 2539 Woodward Ave.
Grand Rapids, Mich., Hero D. Bratt, 228 Ottawa Ave., N.W.
Harrisburg, Pa..................Myles Standish, 210 Muench St.
Jackson, Miss..........................Karl J. Hart, 903 N. State St.
Knoxville, Tenn......................E. Lloyd Widner, Box 2012
Lake Charles, La...Albert K. Newlin
Los Angeles, Calif.....Joe S. Earhart, 1709 W. Eighth St.
Missoula, Mont...............W. M. Walterskirchen, Box 1163
New Orleans, La.....Percy L. Luck, 1936 General Taylor St.
New York, N. Y., Air Conditioning Utilities Co., 8 West 40th St.
Norfolk, Va.........Laurence Trant & Co., 113 W. Main St.
Philadelphia, Pa.......Hucker-Pryibil Co., 1700 Walnut St.
Phoenix, Ariz.....Capitol Engineering Co., 426 E. Jefferson
Portland, Ore........................H. W. McKenzie, Lewis Bldg.
Rochester, N. Y.............A. L. Haskins, 1624 Monroe Ave.
St. Louis, Mo. Edwin H. Shutt, 4 Algonquinwood, Webster Groves
Saginaw, Mich., W. A. Witheridge Company, 746 S. Fourth Ave
Salt Lake City, Utah, Rushby C. Midgley, 201 Bennett Bldg.
San Antonio, Texas, E. G. Langhammer Co., 436 Main Ave.
San Francisco, Calif.......Edgar O. Kaup, 926 Natoma St.
Seattle, Wash., Heating Service Co., Inc., 326 Columbia St.
Shreveport, La............................C. H. Cox, 515 Market St.
Spokane, Wash., Heating Assurance, Inc., 124 E. Augusta Ave.
Tampa. Fla...Pat Flanagan, Box 445
Toledo, Ohio, Refrigeration Sales & Engrg. Co., 4112 N. Haven Ave.
Washington, D C.........Gleggo Thomas, 1426 G St., N.W.

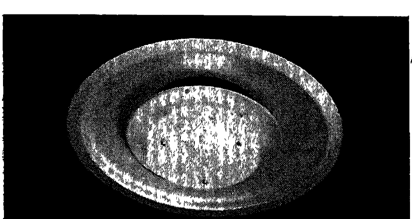
PRODUCTS: Plandaire KNO-DRAFT High Velocity Spun Aluminum Ceiling Type Air Diffusers. Plandlites and Pan Plaques.

MODEL F KNO-DRAFT DIFFUSER



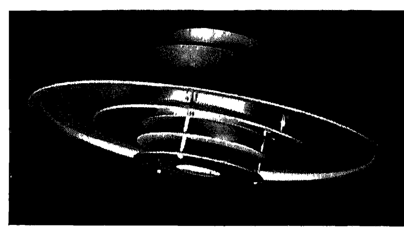
The Model "F" KNO-DRAFT is adjustable and can be used on either low or high ceilings and for various air throws. It can be installed in the ceiling or may be mounted on exposed duct work. Anti-smudge rim prevents streaked ceilings. Sizes 2½ in. to 42 in.

MODEL L PLANDLITE



Plandlites available in sizes from 6 in. to 42 in. neck diameters. Combines an Air Diffuser with a Lighting Fixture. Light distribution curves, resistances and selection data given in complete catalog. Model "L" is a direct light unit and Model IDL-A has indirect lighting equipment.

MODEL SR KNO-DRAFT DIFFUSER



The combination supply and return unit is available in sizes from 6 in. to 42 in. neck diameters. At 1400 fpm will handle up to 7700 cfm supply and 5800 cfm return. Available with indirect lighting equipment. Provides draftless air distribution with resultant temperature equalization.

MODEL FT KNO-DRAFT DIFFUSER

The Model "FT" KNO-DRAFT is of extra heavy gauge aluminum for use in buses, trolley cars and marine craft. Sizes 2½ in. to 42 in. May be equipped with volume control devices. All Plandaire products are Union made.

Write for Complete Catalog.

Tuttle & Bailey, Inc.

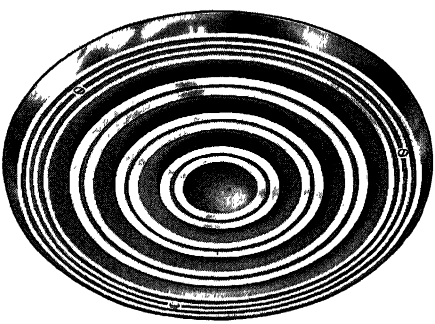
New Britain, Conn.

Branch Offices: New York, Chicago, Philadelphia

Ceiling Diffusers
Grilles, Registers and Intakes
Air Control Devices

Ornamental Grilles
Cast or Wrought Metals
Copper Convection Heaters

AEROFUSE OUTLET A Truly Flush Type Ceiling Diffuser

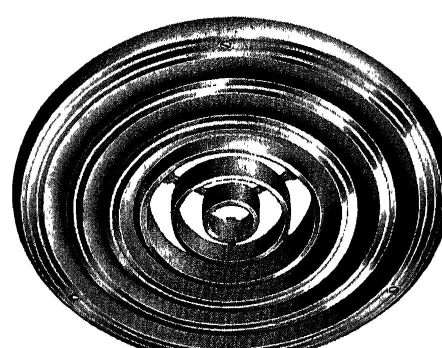


The Aerofuse Outlet is a perfected combination of real beauty and functional efficiency. Flush with the ceiling line and of simple attractive design it harmonizes unobtrusively with any style of interior decoration. Most important, however, is its superb performance. Because of unique construction, the supply air is brought into contact with room air over the largest possible area immediately after leaving the outlet. This results in a high rate of temperature equalization and eliminates the possibility of drafts.

1 Efficient Air Mixture . . . 2 Rapid Temperature Equalization . . . 3 Complete Air Distribution . . . 4 Total Elimination of Drafts

COMBINATION SUPPLY AND RETURN UNIT

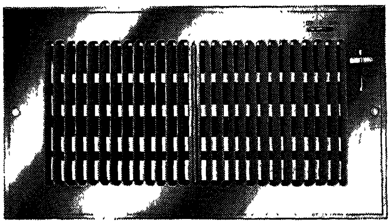
Particularly useful in installations where simplification of the duct layout is of primary importance, since the return (exhaust) duct may be run to the same point as the supply duct instead of to a grille at some other location. The removable center section provides Free Area for the return or exhaust of at least 60 per cent of the supply volume.



THE AEROFUSE OUTLET GIVES YOU BEAUTY
PLUS FUNCTIONAL EFFICIENCY

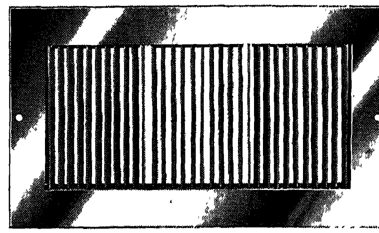
Tuttle & Bailey, Inc.

New Britain, Conn.



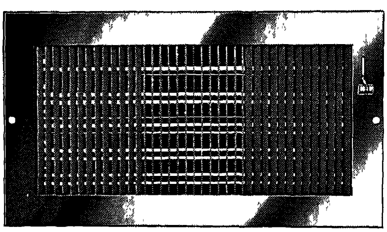
ADJUSTIBLADE REGISTER

A very inexpensive register. The air flow may be deflected sideways by the individually adjustable face vanes, and up and down by back blades. Also available with Flexair design (sectionally adjustable) face.



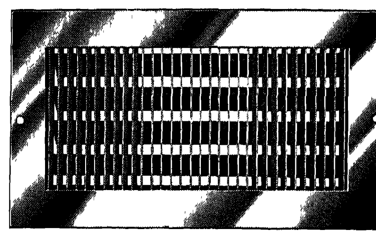
AIR CONDITIONING GRILLES

Furnished in a fixed deflection (Airline design) and a sectionally adjustable (Flexair design) type with bars running either vertically or horizontally.



DOUBLE CORE OUTLET

The vertical front bars are sectionally adjustable for sideways deflection. The horizontal back bars are either sectionally adjustable (Flexair design) or are of fixed deflection (Airline design).

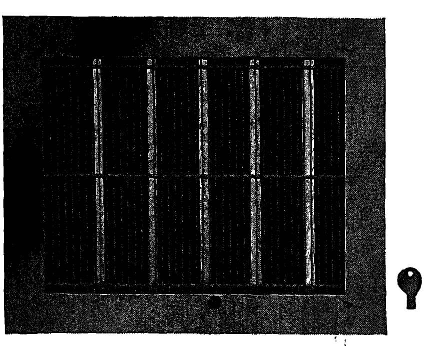


DOUBLE DEFLECTION OUTLET

Made with front bars of fixed deflection (Airline design) or with front bars sectionally adjustable (Flexair design) and individually adjustable back blades.

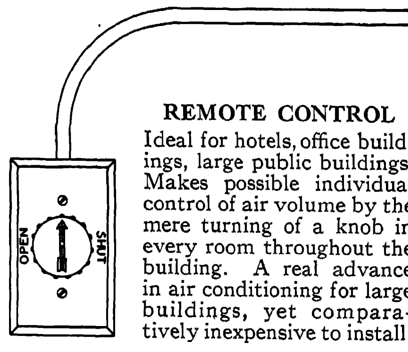
SANTROLS

A simple device to provide positive control of air volume throughout an entire duct system and insure even distribution of air over entire outlet face.



McKNIGHT REGISTER

Provides positive control of air volume at the outlet. Scientifically designed volume control louvers are operated by means of a special key furnished with each register.

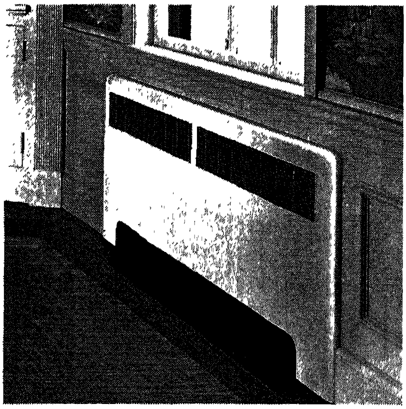


REMOTE CONTROL

Ideal for hotels, office buildings, large public buildings. Makes possible individual control of air volume by the mere turning of a knob in every room throughout the building. A real advance in air conditioning for large buildings, yet comparatively inexpensive to install.

Tuttle & Bailey, Inc.

New Britain, Conn.

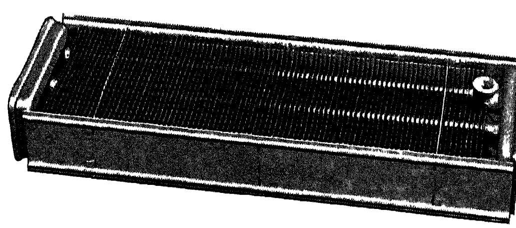


STANDARDIZED ALL COPPER CONVECTOR

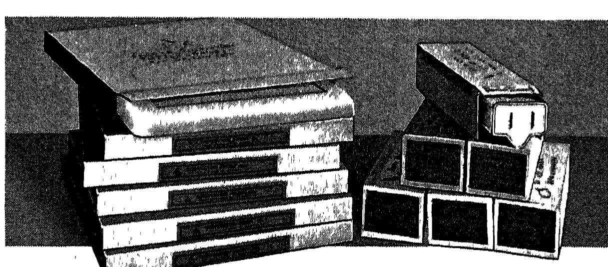
An all copper convector which has been completely standardized to meet every present day condition of steam, vapor or hot water heat. Front panels made with pleasing rounded corners and equipped with the efficient and unobtrusive Airline Grille. Furnished in a Recessed, Free Standing, or Semi-Recessed Type.

TUTTLE & BAILEY HEATING ELEMENT

All copper throughout. Fins and tubes are permanently fixed by a metallic bond uniting them into an integral unit. Headers are also of wrought copper requiring a minimum of space and allowing the use of the greatest number of fins for a given length of element. Copper tubes are scientifically shaped to allow minimum resistance to air flow and maximum contact with fins.



PACKED IN CARTONS FOR PROTECTION AND EASE IN HANDLING



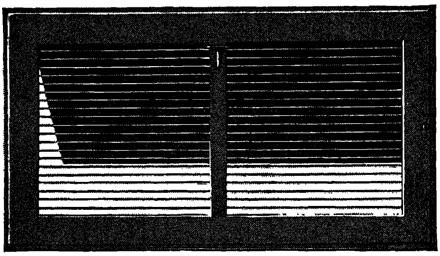
Tuttle & Bailey All Copper Convectors are packed in strong corrugated cartons for full protection from factory to installation and for ease in handling. They are useful during completion of the job as the elements can be installed before plastering and the carton tacked up across the opening, causing the elements to deliver their full rated heat and insuring the fronts against damage during plastering and finishing.

United States Register Company

General Offices: **Battle Creek, Mich., U. S. A.**

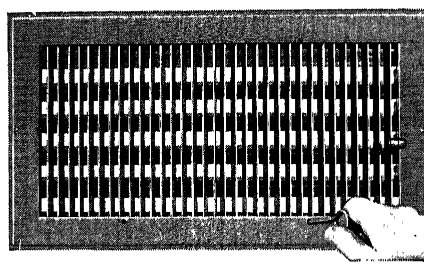
Branches: MINNEAPOLIS, MINN., KANSAS CITY, MO., ALBANY, N. Y., NEW YORK, N. Y., SAN FRANCISCO, CALIF.

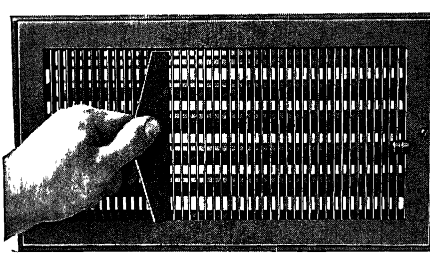
Air Conditioning Registers, Vents and Grilles



Style 153LF—Louver-Type Air Conditioning Register-Bars ¼ in. deep—Spaced 4 openings to the inch affords Non-Vision. Can be supplied in Directional Flow in either Horizontal or Vertical Bar Styles. Can be furnished with all styles of Setting Frames and with INSET PANELS which conveniently afford Multi-Flow.

Style 249LF—Duo-Deflection Air Conditioning Register. Gives complete Air Control. Vertical Front bars—Key-pin adjusted to provide 45 deg Right and Left or Two-way Side Flow. Lever operated Horizontal Back-valves give from Full Closed to any degree of Upflow and to 45 deg Down-flow. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads.





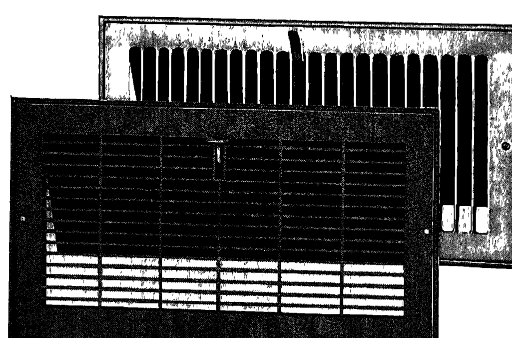
Style 256LF—Flex-bar Air Conditioning Register. Vertical Front Bars set 22 deg Right and Left. Side Flow Deflection attained by setting of Grille Bars with bending wrench to accommodate room condition. Back-valves give same Up and Down control of air flow as 249LF above. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads.

All of above Styles can be supplied with either Lever or Individually adjusted Multiple Valves or Louvers. I. E. 153VVI—Vertical Valves Individually adjusted. 145VVL—Lever operated Vertical Valves.

Style 103N-LF—Horizontal Lattice Perforated Register for Forced Air Systems. Not directional flow.

Style 102LF—Vertical Embossed Bar Design—Not directional flow.

Grilles and Vents in Matching designs.



For Complete Information Write for Latest Catalog

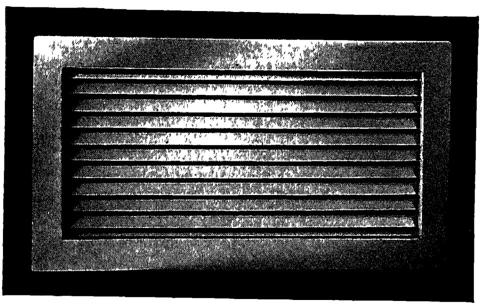
In Canada, United States registers, vents and grilles are manufactured and distributed by the CANADA REGISTER & GRILLE CO., Ltd., Toronto, Ontario

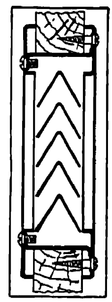
Waterloo Register Company

Waterloo, Iowa **Seattle, Wash.**

Established 1902

Representatives in Principal Cities





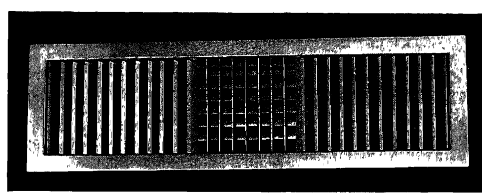
NO-VE-U Door Ventilator

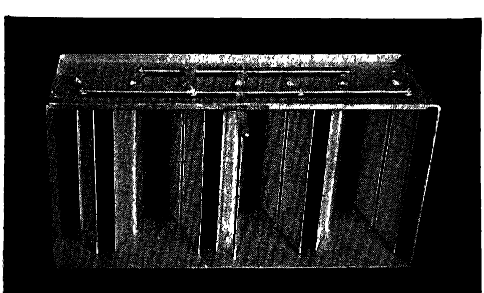
A sight-proof ventilator of rugged construction, either of fixed or adjustable design, for installation in doors of 1⅛ in., 1⅜ in. and 1¾ in. thickness. Opening in door must be exact and measurable in even inches. This unit is available with several different styles of frames, and the louvres can be furnished of 19 gauge steel, as standard, or of 16 gauge steel at extra cost where special strength is required. Not made to fit openings wider than 36 in. Available in various finishes.

FG-75 GRILLE

FGV-75 Front blades parallel to short dimension.
FGH-75 Front blades parallel to long dimension.

Individually adjustable double directional grille with streamline blades 1 in. deep spaced on ¾ in. centers. A removable wrench provides adjustment after installation. Depth of grille 2¼ in.





TECHNI-TROL VOLUME DAMPER

Made to fit inside dimensions of duct. The blades of this damper are arranged to open and close on rust proof bearings so that the volume of air may be reduced or increased without deflecting the air stream from a straight course. Can be arranged with locking mechanism for operation through grille or can be attached to various remote control wires or rods.

Descriptive matter and catalogs available from main office or representatives.

The American Rolling Mill Company

Executive Offices, **Middletown, Ohio**

Atlanta, Ga., 1437 Citizens and Southern National Bank Bldg.	Indianapolis, Ind. .... 1106 Fletcher Trust Bldg.
Berkeley, Calif .... Seventh and Parker Sts.	Kansas City, Mo. .... 7100 Roberts St.
Boston, Mass. .... 201 Devonshire St.	Middletown, Ohio .... 703 Curtis St.
Buffalo, N. Y. .... 504 Seventeen Court St. Bldg.	Minneapolis, Minn. .... 171-27th Ave., S. E.
Chattanooga, Tenn. .... 712 Chattanooga Bank Bldg.	New Orleans, La. .... 3501 S. Carrollton Ave.
Chicago, Ill. .... 310 S. Michigan Bldg.	New York, N. Y. .... 120 Broadway
Cleveland, Ohio .... 1516 B. F. Keith Bldg.	Philadelphia, Pa. .... 1808 Lincoln-Liberty Bldg.
Dallas, Texas .... 1111 Santa Fe Bldg.	Pittsburgh, Pa. .... 1626 Oliver Bldg.
Detroit, Mich. .... 5-261 General Motors Bldg.	St. Louis, Mo. .... 1725 Ambassador Bldg.

Choose the Correct ARMCO Grade

These grades of ARMCO sheet metal are recommended for the air conditioning applications shown. For detailed information get in touch with the nearest district office or write direct to The American Rolling Mill Company, Middletown, Ohio.

ARMCO Ingot Iron

(Galvanized)

Ducts
Washer Chambers
Plenum Chambers
Steam Line Casings
Furnace Casings
Spray Towers
Drip Pans
Housings
Machine Guards
Unit Conditioners (Industrial)
Roof Ventilators
Eliminator Blades

ARMCO Paintgrip

(Galvanized)

Recommended for all applications listed above that need immediate painting. Send for complete information.

Hot Rolled

(Sheets and Strip)

Fan Blades
Blower Casings
Fuel Oil Tanks
Unit Conditioners
Stoker Hoppers

ARMCO Zincgrip

A special galvanized sheet that can be severely formed without peeling or flaking of the tightly adherent zinc coating.

Cold Rolled

(Sheets and Strip)

Furnace Casings
Room Unit Casings

Plates

(ARMCO Ingot Iron)

Smoke Stacks
Coal Hoppers
Breeching
Unfired Pressure Vessels
Low-fired Boilers
Tanks

ARMCO High Tensile

A low alloy, high tensile steel possessing great strength. Used with proper design it results in weight reduction of framework, tanks and similar items. Under atmospheric service conditions it has four to six times the resistance of regular steel.

Stainless Steel

(Sheets, Strip and Plate)

Combustion Chambers
Heat Flues and Tubes
Furnace Casing Trim
Grilles
Corrosion Resistance
Fan and Blower Blades
Heat Resistance without destructive scaling up to 1600° F. or higher.

Other ARMCO Products

The grades for these applications are only a few that ARMCO makes. Others include copper-bearing sheets and plates and open-hearth steel, either galvanized or uncoated.

Bethlehem Steel Company

General Offices: Bethlehem, Pa.

BETHLEHEM STEEL COMPANY, GENERAL OFFICES: BETHLEHEM, PA. DISTRICT OFFICES: AKRON, ALBANY, ATLANTA, BALTIMORE, BOSTON, BUFFALO, CHICAGO, CINCINNATI, CLEVELAND, COLUMBUS, DALLAS, DENVER, DETROIT, HONOLULU, HOUSTON, INDIANAPOLIS, JOHNSTOWN, PA., KANSAS CITY, MO., LOS ANGELES, LOUISVILLE, MILWAUKEE, NASHVILLE, NEW HAVEN, NEW ORLEANS, NEW YORK, PHILADELPHIA, PITTSBURGH, PORTLAND, ORE., ST. LOUIS, ST. PAUL, SALT LAKE CITY, SAN ANTONIO, SAN FRANCISCO, SAVANNAH, SEATTLE, SPRINGFIELD, MASS., SYRACUSE, TOLEDO, TULSA, WASHINGTON, WILKES-BARRE, YORK. EXPORT DISTRIBUTORS: BETHLEHEM STEEL EXPORT CORPORATION, NEW YORK.

COPPER-BEARING BETH-CU-LOY FOR RUST RESISTANCE

The charts at the right show conclusively the superior rust resistance of copper-bearing steel. Sheets of the same composition as Beth-Cu-Loy, Bethlehem's copper-bearing steel, outlasted ordinary iron and steel by a wide margin when exposed to atmospheric corrosion.

Beth-Cu-Loy, available in the form of sheets, pipe and plates, offers 2 to 3 times longer life as indicated by these three corrosion tests—and Beth-Cu-Loy costs only 3 to 5 per cent more than ordinary steel, much less than open-hearth or copper-bearing iron.

Heating, ventilating and air conditioning engineers, architects and contractors are finding it pays to specify Beth-Cu-Loy wherever moisture or corrosion is a factor. Beth-Cu-Loy sheets are easily workable, durable and low in cost.

BETHLEHEM MAKES:

Sheet Steel—all grades, hot-rolled (black), cold-rolled, and galvanized—available in Beth-Cu-Loy.

Steel Pipe—In sizes up to 3 inches, now made by the Continuous-Weld Process and sold under the trade name Beth-Co-Weld.

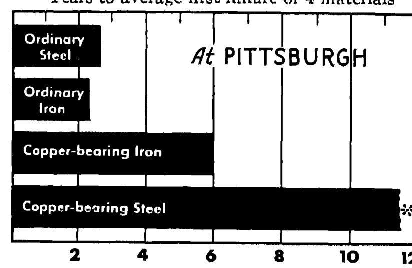
Ammonoduct—A new kind of pipe that has an outstanding advantage in its unusual ductility. It can be bent cold, without need for annealing, without danger of fracturing. Recommended for ammonia piping and for heater coils, water legs in furnaces and similar uses where pipe must be bent.

Boiler Tubes—Charcoal iron and steel.

Plates—all sizes; flanged and dished heads. Available in Beth-Cu-Loy.

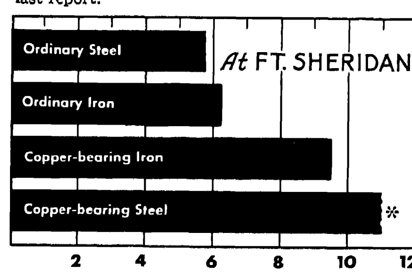
Literature and further information on any of these products can be secured from the nearest district office or from the general offices in Bethlehem, Pa.

A.S.T.M. Test of 22-gage black sheets
Years to average first failure of 4 materials



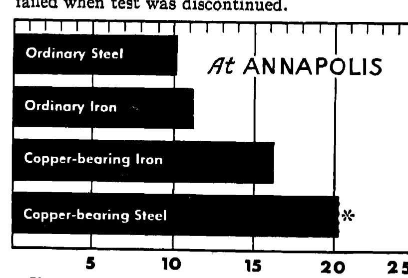
Sheets exposed April 21, 1926; tests still under way (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 38).

*No failures in copper-bearing steel sheets at last report.



Sheets exposed April 19, 1917; test discontinued April 16, 1928 (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 28).

*Only 10 of 61 copper-bearing steel sheets had failed when test was discontinued.



Sheets exposed October 17, 1916; tests still under way (see Proceedings of *A.S.T.M.*—Committee A-5, Vol. 38).

*Only 9 of 78 copper-bearing steel sheets had failed at last report.

A booklet "Beth-Cu-Loy Sheets," gives the story of these tests. A copy is yours for the asking.

United States Steel Corporation Subsidiaries

Carnegie-Illinois Steel Corporation, Pittsburgh and Chicago

Columbia Steel Company, San Francisco

Tennessee Coal, Iron & Railroad Company, Birmingham

United States Steel Export Company, New York

District Offices in all Principal Cities

U. S. S. COPPER STEEL

For Superior Rust Resistance at Low Cost

Corrosion resistance and cost are two determining factors of the type of metal to be used for various air conditioning jobs.

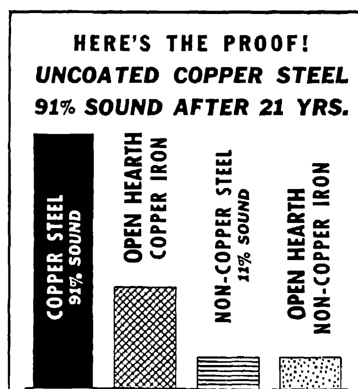
Copper Steel has 2 to 3 times the atmospheric corrosion resistance of plain steel or pure iron as shown in the results of unbiased tests made at Pittsburgh, Ft. Sheridan and Annapolis by the American Society for Testing Metals.

The cost of U. S. S Copper Steel is less than that of pure iron or copper-bearing pure iron and only slightly more than plain steel. Thus there often is a dividend of 200 per cent to 300 per cent longer life and a saving in the first cost as well.

When galvanized, U. S. S Copper Steel produces a sheet that is rust resistant all the way through—not just on the surface. It should be used for all ducts carrying humidified air or placed in damp locations such as basements, shower rooms, etc.

U. S. S. PAINTBOND

When painting is desirable for galvanized steel, U. S. S Paintbond should be used. This makes it possible to paint the job immediately. Paintbond offers a much better surface for painting, prevents flaking of the paint and retards corrosion. It is used for duct work, furnace housings and outdoor sheet metal work.



Corrosion test of A.S.T.M. on 22 gage black sheets exposed at Annapolis, Md., October, 1916. The copper steel sheets outlasted all others in the test.

OTHER U. S. S. PRODUCTS INCLUDE:

Black Sheets—All grades, hot rolled, cold rolled in a number of different finishes.

Stainless—Heat resisting steel for various uses where temperatures are high and corrosion severe.

Cor-Ten—High Tensile steel—greater strength, greater atmospheric corrosion resistance for smokestacks, hoods, etc.

Efficient performance of the Air Conditioning apparatus and Air System Equipment shown in the two preceding sub-divisions is dependent upon proper adjustment and control. For these purposes there are many types of instruments for testing and adjusting the apparatus, and devices for control of its operation.

The functions and methods of using controls and instruments are described in Chapters 33 and 34 of the Technical Data Section. In the following sub-division of the Catalog Data Section—Controls and Instruments—these devices are illustrated and described, and factual data given by the manufacturers.

Following the data on Controls and Instruments is a sub-division on Steam and Hot Water Heating Systems, including the many parts required to make up the complete systems. This type of apparatus too, requires testing, adjusting, and control; many of the same devices used with air systems are also suitable for use with steam and hot water systems.

CONTROLS AND INSTRUMENTS

•

Automatic controls form an essential part of modern heating, ventilating and air conditioning equipment, and for the refrigerating equipment which performs important functions in many air conditioning operations. Their use makes possible accurate maintenance of desired physical conditions, with an operating efficiency and economy which are not obtainable with manually operated controls.

Instruments of many types and for many uses are available for determining the capacity and operating efficiency of apparatus. These instruments are designed to obtain results in conformity with adopted test methods and operating standards.

CONTROLS (p. 938-965)

Thermostats—room, immersion, insertion and surface types; humidity controls, pressure controllers, damper motors, control valves, solenoid valves, relays, etc.

For control of air, gases, temperatures, humidity and liquids; for automatic fuel burning apparatus; for all types of heating, ventilating and air conditioning apparatus operating as separate units, or as integral parts of central systems.

The various types of automatic controls include electric, pnuematic, and self-contained control systems—two-position, or on-and-off, and the modulating or graduated control. They are adaptable for individual room control, or for zone control in large buildings, and also for industrial process control.

Technical data on automatic controls will be found in Chapter 33.

INSTRUMENTS (p. 938-965)

For measuring, indicating and recording air velocity, temperature, humidity, pressure, flow and liquid levels; and for testing and rating heating, ventilating and air conditioning equipment.

They include gauges, meters, recorders and indicators, hygrometers, pyrometers, psychrometers, thermometers, velometers.

Technical data on instruments is contained in Chapter 34.

Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

Alco Valve Company

ENGINEERED REFRIGERANT CONTROLS

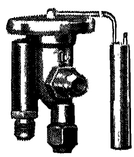
2638 Big Bend Blvd., St. Louis, Mo.

NEW YORK OFFICE: 381 Fourth Ave. CHICAGO OFFICE: 433 East Erie St.

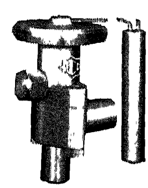
A complete line of Engineered Refrigerant Controls

THERMO EXPANSION VALVES

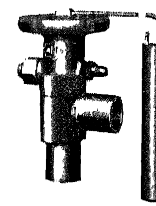
For automatic control of liquid refrigerant on all types of air conditioning and refrigeration systems.



Type TK



Type TJL



Type THL

CAPACITIES—From fractional tonnage to 100 tons Methyl Chloride, 50 tons Freon-12.

MAGNETIC STOP VALVES

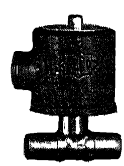
For all types of service

Magnetic Liquid Stop Valves

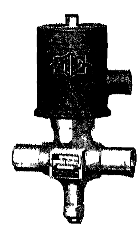
Freon—up to 75 tons, Methyl Chloride—up to 150 tons.

Magnetic Suction Stop Valves

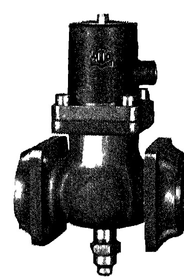
Freon—up to 1½" or 8.8 tons Methyl Chloride—up to 1½" or 17 tons.



Type S1



Type M3



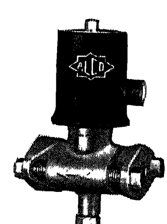
Type R2

AMMONIA CONTROLS

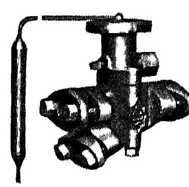
Magnetic Liquid Stop Valves — up to 172 tons.

Magnetic Suction Stop Valves— up to 1½" or 28 tons.

Thermo Expansion Valves— from fractional tonnage to 60 tons.



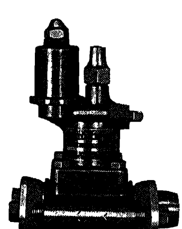
Type M5



Type TGS

EVAPORATOR PRESSURE REGULATORS

For Freon, Methyl Chloride and Ammonia, with port sizes up to 2 in., and a wide variety of connection sizes.



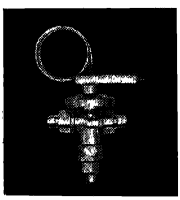
ALCO also offers Magnetic Stop Valves for brine, water, gas, air and steam; specially designed Magnetic Compressor Discharge Valves and Magnetic Pilot Check Suction Stop Valves (for lines subject to reverse Flow).

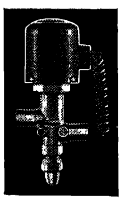
In addition, the Alco line of Engineered Refrigerant Controls includes Float Valves, Float Switches, High Pressure Float Valves, Constant Pressure Expansion Valves and liquid and suction line Filters.

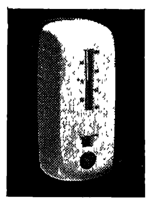
AUTOMATIC PRODUCTS COMPANY

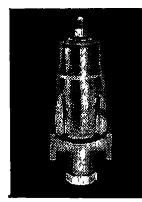
2450 NORTH THIRTY — SECOND STREET
MILWAUKEE WISCONSIN

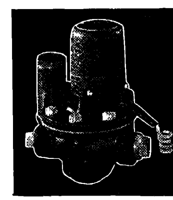
A-P DEPENDABLE CONTROLS
For Heating, Refrigeration and Air Conditioning


1.


2.

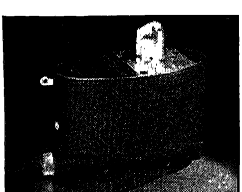

3.


4.

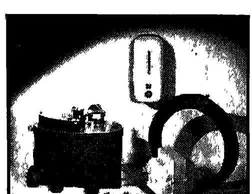

5.

1. **A-P Thermostatic Expansion Valves.**
   Several models and sizes, for capacities up to 16 tons Freon or 32 tons Methyl Chloride.
2. **A-P Solenoid Refrigerant Valves.** Capacities up to 50 tons Freon.
3. **A-P Thermostats.** For Cooling or Heating.
4. **A-P Water Regulating Valves.** Capacities up to 1440 Gallons per hour.
5. **A-P Solenoid Operated Water Valve.** Made especially for Deep Well Cooling.

A-P Controls for Oil Burning, Gravity-Feed Heating Plants.



A-P Constant Level Oil Control Valve— With Fuel Compensator. Used on Gravity Oil Burning Appliances.



A-P Complete Furnace Control Set— Made in variety of types for Gravity-Feed Oil Burning Furnaces.

A-P Fuel Oil "Trap-It"—Traps dirt and water in fuel systems. Improves operation of all oil burning devices.

A-P Valve DEPENDABILITY

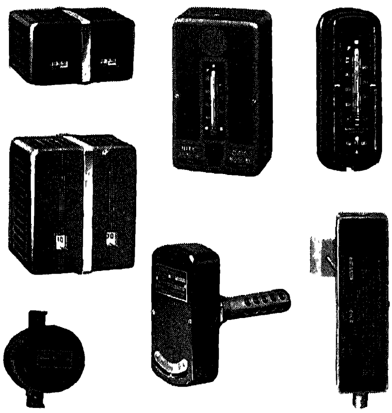
is widely recognized in Refrigeration, Air Conditioning and Heating. This reputation is born of close adherence to a rigid standard of perfection—in materials used, careful testing and inspection, simplicity of construction, and many unusual features.

Barber-Colman Company

Rockford, Illinois

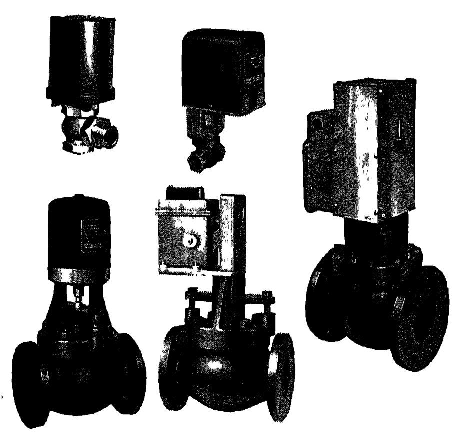
TEMPERATURE and HUMIDITY CONTROL EQUIPMENT

Barber-Colman Controls are all electric. Precision built to insure long, continuous, and dependable service. Easy to install in either new or existing installations. Ready for instant service at all times, even after long shut down periods.



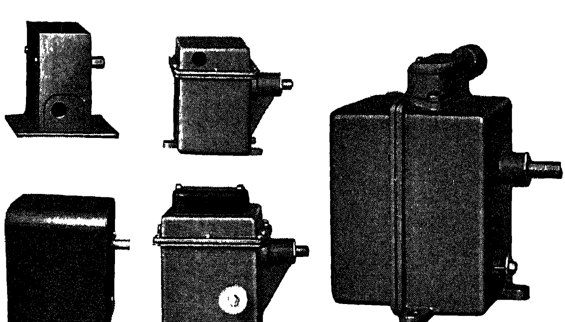
Thermostats, Hygrostats, and Comfortstats. Instruments for controlling dry-bulb, wet-bulb, effective temperature, and relative humidity. For snap-action, floating, or proportioning control. Supertherm has a horizontal bimetal element and is more sensitive to changes in room temperature. The Comfortstat takes into consideration relative humidity as well as temperature and therefore regulates in accordance with effective temperatures.

The Econostat (not illustrated) is a complete self-contained thermostatic unit for the automatic regulation of the heat supply of a building in accordance with outdoor temperatures.



Motor-Operated Valves. Packless, packed, single seat, pilot piston, vee-ported, balanced, three-way, four-way, and butterfly. For shut-off, throttling or proportioning service. Also solenoid valves for air, oil, water, gas, and refrigerants. Motor-Operated Valves are powered with Barcol motors which have only one moving part and require no attention except oiling; oil submerged operators require no attention.

All gears are machine cut and steel gears heat treated. All exposed steel parts are zinc plated. Motor-operators are detachable as units.



Damper Control Motors. Unidirectional or reversible, fixed or adjustable speed. For positive or proportioning operation of dampers in heating, ventilating, air conditioning, or industrial applications.

Oil-submerged models have the motor and gear train entirely submerged in oil, thus insuring quiet operation and long dependable service.

Automatic cam-operated switches provide accurate stopping at the desired limits.

(*See also Page 921*)

The Bristol Company

Waterbury, Connecticut

Branch Sales and Service Offices

Akron, Ohio	Chicago, Ill.	New York City	San Francisco, Cal
Birmingham, Ala.	Detroit, Mich.	Philadelphia, Pa.	St. Louis, Mo.
Boston, Mass.	Los Angeles, Cal.	Pittsburgh, Pa.	Seattle, Wash.

The Bristol Company of Canada, Ltd., 64 Princess Street, Toronto, Ontario

Bristol's Instrument Co., Ltd., North Circular Road, London, N.W. 10, England

A COMPLETE LINE OF INSTRUMENTS FOR RECORDING, INDICATING AND CONTROLLING

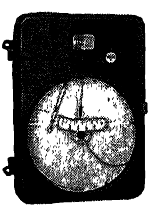
TRADE-MARK

BRISTOL'S

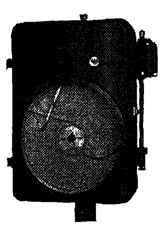
REG. U. S. PAT. OFFICE

FLOW METERS, ELECTRIC AND MECHANICAL

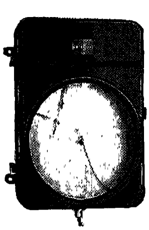
For indicating, recording, automatically controlling, and integrating flow of steam, air, gases, liquids. Electric Flow Meters telemeter measurements of flow any distance up to several hundred miles. *Catalog No. 1051.*



Electric Flow Meter, Model M1040 M



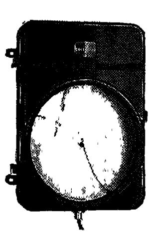
Mechanical Flow Meter, Model 1140M



Recording Pressure Gauge, Model 40M

RECORDING PRESSURE GAUGES

For securing continuous records of pressure, vacuum or liquid level for steam, air, gas, liquids. For ranges from full vacuum to 12,000 lb per sq in. *Catalog No. 1025.*



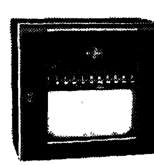
Recording Thermometer, Model 240M

RECORDING AND INDICATING THERMOMETERS

Recording Thermometer: For all commercial ranges from 60 F below to 1000 F above zero. For wall, flush, or panel mounting. *Catalog No. 1251.*

RECORDING AND INDICATING RESISTANCE THERMOMETERS

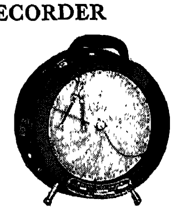
For securing, at a central point, readings of a number of temperatures at distant points. As many as eight temperature records can be made on one chart simultaneously. *Catalog No. 1452* and *Bulletin No. 997.*



Eight-point Wide Strip Resistance Thermometer, Model 427

DIRECT READING RELATIVE HUMIDITY RECORDER

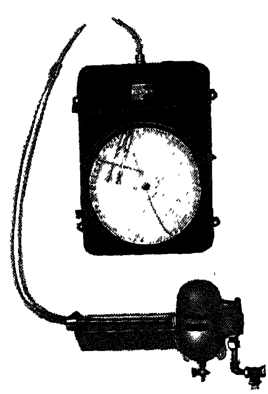
Shows trend of humidity and humidity conditions. No calculations or humidity tables needed. Requires no water, no fan. Portable case. *Bulletin No. 554.*



Thermo-Humidigraph, Model 4069

WET- AND DRY-BULB PSYCHROMETER

Records wet- and dry-bulb temperatures in kilns, drying rooms, air ducts. Self-contained and distance types. Moisture-, fume-, and dust-proof case. For wall, and flush panel mounting. *Catalog No. 1251.*

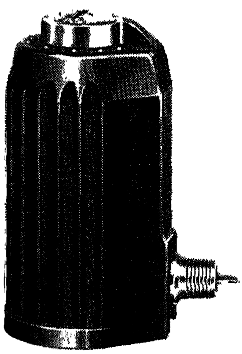


Wet- and Dry-Bulb Psychrometer, Model 4240M

IDEAL FAST-VENTING SYSTEM

For Automatically Fired One-Pipe Steam Jobs

An automatic, oil, gas or coal burner operates on a pronounced *on* and *off* cycle. Therefore, on automatically fired one-pipe steam jobs, all venting and all heating must be accomplished during the limited *on* period—the venting first. The time factor in venting is of utmost importance, especially when using the compensated or "preheater" type thermostat which causes heating cycles even shorter.



New No. 300 Multiport

About three years ago, the Detroit Lubricator Company brought out the Ideal Fast-Venting System which consists of No. 300 Multiport for radiators and the No. 861 Hurivent for mains which have very large ports, and allow both radiators and mains to vent in only a small fraction of the time needed when conventional valves are used.

Experience has demonstrated the outstanding advantages of large port fast venting for automatically fired one-pipe steam systems. The more important of these advantages are:

1. Quicker response to the thermostat's call for heat.
2. System balance—all radiators heat up simultaneously.
3. Definite elimination of cold rooms.
4. Material reduction in fuel costs.

The New 300 Multiport

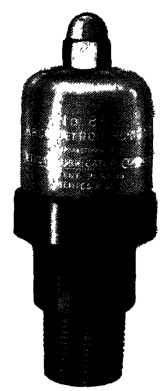
With the principle of large port fast-venting definitely proven, Detroit engineers have taken the next logical step in the progress of fast-venting. A step that carries this practice to the full limit of its possibilities. They have redesigned the No. 300 Multiport adding several new and very valuable features.

The new No. 300 Arco-Detroit Multiport has a venting capacity even greater than the old No. 300, a greater range of adjustment, and permits even, fast heat delivery as the burner comes on.

Boiler pressure, in nearly all cases, need never exceed a few ounces—still further reducing firing cycle and fuel cost. Recommended only for automatically fired systems controlled to a maximum operating pressure of 3 lb.

Other Important Changes

1. Minimum venting rate is automatically controlled which makes it possible to secure the absolute minimum venting rate for oversized radiators or radiators very near the boiler.
2. A siphon tongue replaces the old siphon tube so that the new valve can be used on thin radiation without the siphon conflicting with the wall opposite the connection.
3. Together with a simple adapter, the new valve can be used on installations which otherwise would require a straight shank valve.
4. Dealers and contractors stock only one model, that illustrated herewith.
5. New Inner-shell construction prevents both noise and spitting.
6. Size reduced and more pleasingly styled.



New No. 861 Hurivent

No. 861 Arco-Detroit Hurivent for Mains

In order to take full advantage of fast-venting, large port vent valves are necessary to vent the mains at the same time air valves are venting risers and radiators. The No. 861 has a full ½ in. port and has for the past several years proved its marked efficiency—the No. 861 will vent more than 260 ft of two-inch main in the short period of 60 seconds at only 4 ounces of pressure.

In addition, the Detroit line includes the new No. 5000 Airid Variport, an adjustable large port valve for installations operating at more than 3 lb. pressure; the No. 500 Airid, an inexpensive non-adjustable air valve; and the No. 841 Ideal Quick Vent for mains. For a vacuum operated hand fired coal job there is the No. 510 Vac-Airid, an air valve, and the No. 862 Vac-Hurivent, and the No. 842 Ideal Vac-Vent, both for the mains.

The Fulton Sylphon Company

Manufacturers of Sylphon Automatic Temperature Controlling Instruments and Packless Expansion Joints

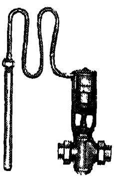
Sales Representatives in Principal Cities

FULTON Sylphon FOR TEMPERATURE CONTROL

Knoxville, Tenn.

HOT WATER SUPPLY

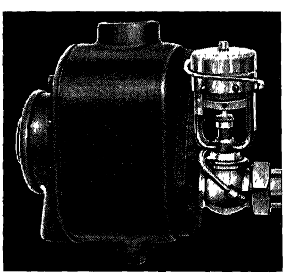
No. 923 Temperature Regulator—For controlling water temperature in heaters, open or closed tanks and other equipment. Operation unaffected by temperature fluctuations at the valve, either above or below bulb temperature. All parts, except steel adjustment spring, made of non-ferrous metals. May be installed in any position. Ranges from 40 −80 F to 290 −330 F. Bulletin HVG-20.



No. 923 Temperature Regulator

Sylphon Thermostatic Water Mixers

Utilize hot water from any storage tank or instantaneous heater, and effectively regulate the amount of cold water required to temper it to the desired degree, actually mixing the hot and cold water together before delivery. Temperature remains constant in spite of fluctuations in supply water temperatures or pressures.

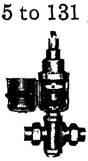


No. 902 Sylphon Thermostatic Water Mixer—14 to 131 gpm depending on water pressure

Four sizes with capacities ranging from 5 to 131 gpm. Bulletin HVG-40.

REFRIGERATION CONTROLS

Adaptable wherever brine is used as the refrigerant. Latest development is a "freeze-proof" valve (illustrated at left on the popular Sylphon No. 945-Z Regulator). Bulletin HVG-20.



No. 945-Z Regulator

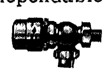
PACKLESS EXPANSION JOINTS

The Sylphon Packless Expansion Joint eliminates useless building height, expensive construction and non-revenue producing space. No costly leaks and repairs, no repacking, always tight, allows heating system to operate at full efficiency. Write for Bulletin HVG-140.

No. 110 Sylphon Expansion Joint

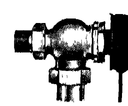
SPACE HEATING CONTROL

No. 885 Automatic Radiator Valve—For exposed radiation. Small, neat, finely finished, adjustable to room temperature desired. Simply replace ordinary radiator valves with these Sylphon Automatic Regulators—no wiring, piping or auxiliary equipment are required. These valves answer the demand for an inexpensive means of providing accurate, dependable space temperature control in rooms, sections or throughout large buildings, new or old. Similar type valves for concealed radiation — get Bulletin HVG-80.



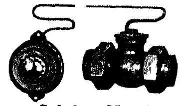
Sylphon No. 885 Automatic Radiator Valve

No. 890 Electric Radiator Control Valve—For either exposed or concealed radiation. Similar in appearance and action to Sylphon Automatic Valves, but operated by an electric wall thermostat. The closing of the thermostat circuit energizes a low voltage electric heater coil surrounding a bulb containing a volatile liquid. This liquid expansion causes pressure on a bellows in the valve head operating the valve. This provides radiator valve control from a remote location, permits regulation of several radiators from a single thermostat, enables a time switch to be installed, if desired, offers effective zone control of large areas at a fraction of the cost of conventional motor-operated valve systems. Bulletin HVG-70.



Sylphon No. 890 Electric Control Valve

No. 7 Temperature Control—A self-contained, self-powered regulator for controlling unit heaters, wall or ceiling type radiators, heating coils in duct-type heating systems, etc. Quickly installed, holds temperatures within close limits. Valve placed in steam line to one or a battery of heaters, thermostat mounted on wall or column. For use on regular heating pressures up to 15 lb. Similar regulators, Nos. 7-2 and 7-3 for 50 and 75 lb pressure and temperatures up to 170 F. Bulletin HVG-50.



Sylphon No. 7 Temperature Control (Self-operating)

HEATING AND AIR CONDITIONING CONTROL

Almost any type of heating, ventilating or air conditioning system can be advantageously controlled wholly or in part by Sylphon Regulators. Basic advantages of Sylphon Controls are:

Modulating—Maintains ideal conditions —not continually correcting too hot, too cold, too humid or too dry conditions.

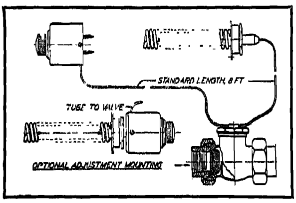
Compensating—Many Sylphon Regulators offer compensating control, automatically raising their low limit setting at a predetermined rate as outside temperatures fall.

Sensitive—Close operating temperature differentials. Quick response.

Simple—in design.

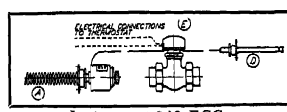
Rugged Construction—To give years of satisfactory service.

Adaptable—Any one of many combinations of Sylphon Instruments can be arranged to control any air conditioning system and to provide exactly the conditions desired. *Write for Bulletin SAC—820.*



Regulator 928-C

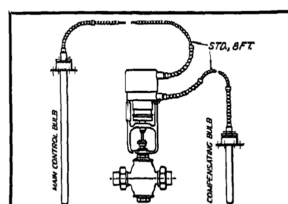
The No. 928-C Regulator—Simple, compact yet highly sensitive. Suitable for modulating control of air temperatures in ducts. Bulb is constructed of numerous coils of copper tubing giving sensitivity to the slightest temperature variation. Packless valve eliminates service problem and makes this regulator ideal for installation in inaccessible locations. Suitable for steam pressures up to 15 lb; other types available for pressures up to 75 lb.



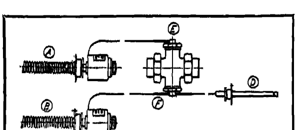
Instrument 928-ECC

The No. 928-ECC Sylphon Instrument—Room control and low-limit control in a single valve regulator for modulating control of ventilating systems. Main control from an electric room thermostat operating through the electric head "E" on the valve. Low-limit control by Bulb "A" located in discharge duct from the heater. Bulb "D", located in inlet side of the duct to the heater, compensates Bulb "A". Compensating thermostat can be furnished to raise low-limit setting at predetermined rate with falling outside temperature. Suitable for steam pressures up to 15 lb.

Sylphon No. 971 Differential Regulator—For controlling room temperature on the cooling cycle, where chilled water or brine is used as cooling medium and where it is desired to have a gradual increase in room temperature as outside temperature increases. This regulator is modulating in action, thereby affords better control over humidity than is procured when usual on-and-off type control is employed.



Regulator 971

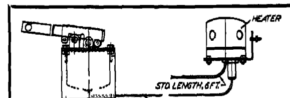


Control 889-C

The Sylphon No. 889-C Control—A modulating, dual-function regulator for control of duct heating and ventilating systems—two independent valves in a single body.

Adjustable Thermostat "A" governing Valve "E" functions to maintain room temperature from temperature of recirculated air. Adjustable Thermostat "B" acts as a low-limit ductstat controlling Valve "F" to maintain minimum discharge air temperature. Bulb "D" compensates Bulb "B" to maintain even discharge air temperature irrespective of demand. Compensated Thermostat "B" can also be furnished to raise its setting at a predetermined rate with falling fresh air temperatures, if desired. Suitable for steam pressures up to 15 lb.

Sylphon No. 371 Positive Type Damper Motor—On-and-off control of dampers. Operation controlled by room thermostat, by hand-operated switch, by motor starting switch, etc. Advantages include: (a) motor returns to closed or safety position in event of current failure; (b) heat-motor bulb and motor separate enhances convenience of installation; (c) damper motor lever adjustable; (d) positive, powerful operation.



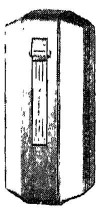
Damper Motor 371

Julien P. Friez & Sons

(Division of Bendix Aviation Corporation)

Baltimore **Maryland**
Established in 1876

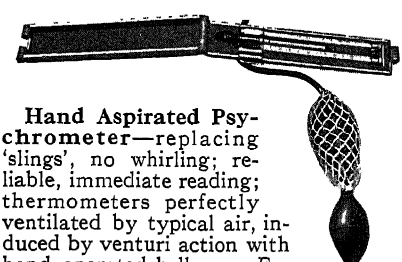
Manufacturers of a Complete Line of Automatic Electric Controls for Industrial and Comfort Applications. Also a Complete Range of Recording and Accurate Measuring Instruments for Indoor and Outdoor Applications



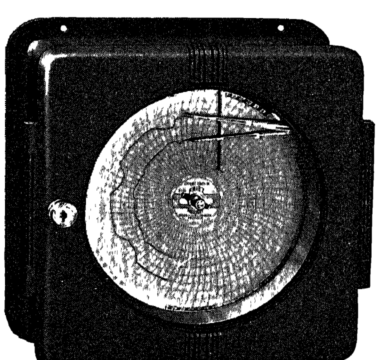
Humidstat—accurate for long periods and over complete range: double length human hair element. **Bulletin AA.**

Thermostat — sensitive, accurate for highest grade work. **Bulletin TT.**

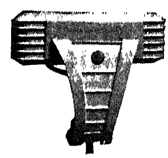
Comfortrol — effective temperature Thermostat resetting itself as prevailing humidity varies, using human hair compensating element. *Exclusive.* **Bulletin E.**



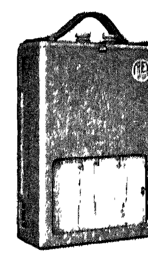
Hand Aspirated Psychrometer—replacing 'slings', no whirling; reliable, immediate reading; thermometers perfectly ventilated by typical air, induced by venturi action with hand operated bellows. *Exclusive.* **Bulletin S.**



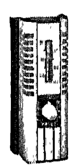
Remote Reading Temperature and Humidity Recorder—Electrically operated; humidity uniquely recorded from distant location directly in percent relative. *Exclusive.* **Bulletin R.**



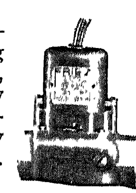
Windowstat—placed at window indoors, positively preventing condensation from excess humidity. Controls humidity supply just below critical point as outdoor temperature varies. *Exclusive.* **Bulletin W.**



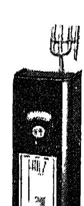
Portable-Recorder—for surveys or tests of humidity and temperature. Inked records on charts the size of filing cards (3″ x 5″). *Exclusive.* **Bulletin G.**



Microstat—Small sized thermostat, featuring powerful Alnico magnets. Though priced with the lowest, unsurpassed for accuracy and fine appearance. **Bulletin TM.**



Magnetic Gas Valve—New principle, free floating disc, no diaphragm; quiet, durable; range of sizes; low priced, low voltage, especially suited for control by Microstat pictured above. *Exclusive.* **Bulletin VG.**



Hydraulic Action

A new and complete line of controls for warm air, hot water and steam systems, industrial ovens, space heating, air conditioning and refrigeration. Permanently accurate, rugged, at competitive prices. **Bulletins CR, LC and TS** and Data Sheet 225.

Write for Bulletins

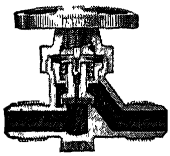
MODERN ADVANCED CONTROLS FOR MODERN NEEDS

Henry Valve Company

1001-19 North Spaulding Ave., Chicago, Ill.

Manufacturers of

Complete line of Dryers, Strainers and Line Valves for Freon, Methyl Chloride and Sulphur Dioxide. Also Ammonia Valves and Forged Steel Fittings.



Balanced-Action Diaphragm Packless Valves

VALUE OF BALANCED ACTION

Regardless of operating conditions or the differential in the pressure above and below the valve seat, "balanced-action" assures positive and instantaneous opening. The balancing action is really the equalizing of the pressures on the two sides of the valve seat at the instant of opening. This is accomplished by a channel in the axis of the valve stem. When the valve is closed, the upper port of this channel is sealed by the diaphragm assembly itself. At the instant of opening, the pressure above the seat forces the diaphragm assembly upward, exposing the upper port of the balancing channel. The pressure is released through the channel to the region below the seat, equalizing the pressures, thus assuring positive opening. A spring-tensioned ball check seals the channel for diaphragm inspection.

Other important features are Ovaline handwheel, ports-in-line, non-rotating bearing plate to protect diaphragm from rotating friction of stem, and use of multiple puncture and fracture-proof diaphragms designed to resist wear and corrosion. Available in a complete range of sizes with flare and solder connections.

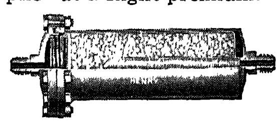
ABSO-DRY PRESSURE SEALED DRYERS

For Refrigeration and Air Conditioning

The exclusive Henry vacuum process first removes every trace of moisture, then the dryer is charged with dehydrated air. Loosening a seal cap prior to installation produces hissing sound, a guarantee of original factory dryness and freedom from leaks.

OTHER FEATURES OF HENRY DRYERS—*Perforated Dispersion tube* is connected to inlet port and exposes entire volume of dehydrant to penetration by refrigerant. *Minimum* pressure drop. No channelling. *Compression Spring* maintains uniform tension on dehydrant at all times and compensates for changes in volume. *Soldered or Flanged Shells*—models are available with either soldered cap or flanged end shells. Flange is distortion-proof. Shells not exceeding 5½ in. in length are drawn in dies, so that they have only one joint.

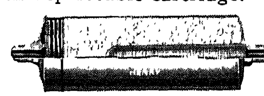
FOUR DEHYDRANTS—Choice of following dehydrants at same price: Activated Alumina, Calcium Chloride, Calcium Oxide and Drierite. Silica gel is supplied at a slight premium.



Type 744 Cartridge Dehydrator

A flanged shell dehydrator with replaceable cartridge.

Type 712 Dehydrator

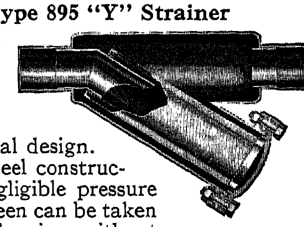


Soldered brass shell dehydrator with dispersion tube and compression spring.

HENRY STRAINERS

There is a size and type of Henry Strainer for every installation requirement.

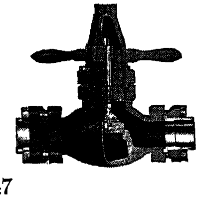
Type 895 "Y" Strainer



With solder fittings for use with copper pipe. Exceptional design. Welded steel construction. Negligible pressure drop. Screen can be taken out for cleaning without removing strainer from line. Very large screen area. Light weight. Baffle prevents heavy particles injuring screen.

WING CAP VALVES

Designed especially for Freon and Methyl Chloride. Have patented rotating self-aligning stem disc. Special resilent packing. May be repacked under pressure. Wing cap can be inverted and socket used for operating valve. Screw end, soldered and flanged connections.



Johnson Service Company

AUTOMATIC TEMPERATURE AND AIR CONDITIONING CONTROL

General Offices and Factory

Milwaukee, Wis.

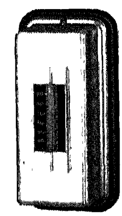
Branch Offices in all Large Cities

JOHNSON TEMPERATURE REGULATING CO. OF CANADA, LTD., 113 SIMCOE ST., TORONTO, ONT.
HALIFAX, N. S. MONTREAL, QUE. WINNIPEG, MAN. CALGARY, ALTA. VANCOUVER, B. C.

PRODUCTS AND SERVICES

Manufacturers, Engineers, and Contractors—For automatic temperature and humidity control systems applied to all types of heating, cooling, ventilating, and air conditioning installations.

Space Control—Automatic control of room temperatures and humidities, applied to radiators, unit ventilators, unit heaters, and heat delivery ducts. Johnson "Duo-Stats" to maintain the proper relationship between outdoor and heating system temperatures for groups of radiators, or "heating zones." A complete line of devices for automatic control of air conditioning systems, heating, cooling, humidifying, dehumidifying.



Single Room Thermostat

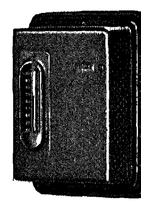
Process Control—Automatic temperature and humidity control devices for manufacturing and industrial processing, applied to tanks, dryers, vats, kettles, curing rooms, coolers, kilns, etc.

Nation-wide Service—Johnson sales engineers, technicians, and trained installation men are available at all branch offices. None of the men in the nation-wide Johnson organization are agents, jobbers, or part-time representatives. All are salaried employees, devoting their entire efforts to the interests of the JOHNSON SERVICE COMPANY and its customers.

Send for Bulletins describing the detailed characteristics of any of the Johnson devices.

JOHNSON THERMOSTATS

Room Thermostats—Intermediate (gradual) or positive (snap) action, maintaining temperatures accurately within one degree above or below point of setting. "Dual" (two-temperature) and "Summer-Winter" types, as well as standard instruments. Various types of covers allow wide selection of adjusting features, guards, and method of mounting. Red-reading thermometers with magnifying tube attached to each cover.

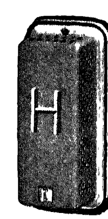


Dual Room Thermostat

Insertion and Immersion Thermostats—Sense temperatures in ducts, tanks, and similar locations. High grade insertion or immersion thermometers for mounting adjacent to the thermostats, including the distinctive Johnson "Sunrise" insertion thermometer, with red-reading mercury column in heavy lens glass tube and 9-in. scale with patented adjustable tilting feature.

Extended Tube Thermostats—Mercury or vapor tension type, to sense temperatures at a point remote from the location of the operating mechanism. Various types of bulbs. Connecting tubing up to 50 ft in length for vapor tension, 75 ft for mercury actuated systems.

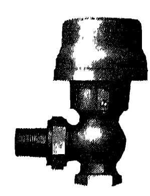
Special Thermostats—For applications encountered in industrial control, including the "Record-O-Stat," combination extended-tube temperature controller and recorder. Full 10-in. chart and vapor tension or mercury actuated systems. Single or duplex type, the latter controlling and recording both wet and dry bulb temperatures.



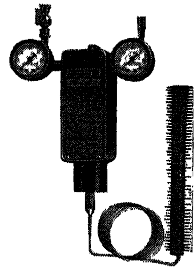
Room Humidostat

Remotely Adjusted Thermostats—A distinctive Johnson feature, applied to various types of instruments where readjustment must be accomplished from a remote point, such as another thermostat or a manual switch.

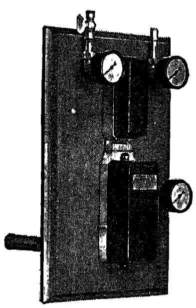
Johnson Sensitivity Adjustment—An important development in automatic temperature and humidity control for air conditioning. A unique and convenient means of adjusting the sensitivity of Johnson thermostats and humidostats, on the job, balancing "time-lag" with respect to the capacity of conditioning apparatus. "Hunting" and temperature fluctuation prevented. Available on all Johnson gradual action insertion and immersion thermostats, insertion humidostats, and certain room type thermostats and humidostats.



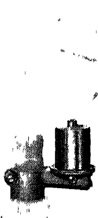
"Sylphon" Radiator Valve



Extended Tube Thermostat



Remotely Adjusted Duct Thermostat



Modulating Attachment for Expansion Valves

JOHNSON HUMIDITY CONTROL

Johnson Humidostats—Automatically control the supply of moisture delivered by a humidifier or by other means, maintaining a constant percentage of relative humidity. Available in both room and insertion patterns and with various types of elements as determined by requirements, the most sensitive controlling within 1 per cent at relative humidities as high as 95 per cent at 100 F. Humidostatic elements are wood-strip, human hair, animal membrane, or other suitable substances as selected.

Johnson Humidifiers—"Steam grid" type (perforated pipe supplied with low pressure steam) or pan type with copper evaporating pan, brass heating coils, and float control.

JOHNSON VALVES

Johnson Diaphragm Valves—Simple and rugged. Seamless metal bellows and heavy spring operate the valve stem. Available, if desired, with diaphragms of special molded rubber, resistant to aging, heat deterioration and oxidation. No complicated moving parts. Made in all standard sizes and patterns. Direct acting (normally open) or reverse acting (normally closed). Also, three-way mixing and by-pass valves. For steam, water, brine, and freon.

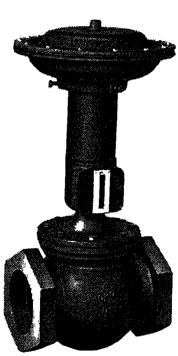
Johnson "Streamline" Diaphragm Valves—Modulating discs and special internal construction, insure superior gradual control . . . Where maximum power is required for repositioning at the slightest demand of controlling instruments, Johnson molded rubber diaphragm valves are fitted with Johnson's dependable pilot feature, for smooth gradual operation, independent of friction and pressure variations.

JOHNSON DAMPERS AND SWITCHES

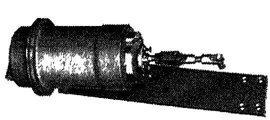
Standard Johnson Dampers—Steel blades in flat steel frames with adequate bracing to form a rigid assembly. Finished in two coats of black lacquer. Special corrosion-resisting finishes on order. Angle iron frames optional. **Special Dampers**—Galvanized iron, monel metal, aluminum, copper, rust-resisting steel, etc. Brass pins in steel bearings or ball bearings.

Johnson Damper Motors—In principle, similar to valves. Seamless metal or specially molded rubber diaphragm operates damper through suitable linkage. Various types of brackets. Distinctive Johnson "Piston-type" damper motors afford long travel at full power, a feature not found in other such devices. With or without pilot mechanism, as described above under "Valves."

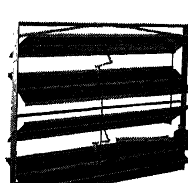
Johnson Pneumatic Switches—Various patterns for operation of dampers and for placing thermostats and other devices in and out of service, as required, from remote points. Standard switchboards are oiled slate. Ebony asbestos, polished oak, and genuine or imitation marble on order.



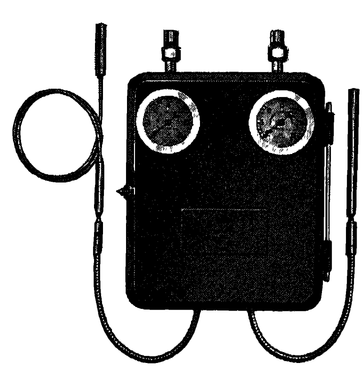
Rubber Diaphragm Coil Valve



Piston Type Damper Motor



Louvered Damper



Johnson "Duo-Stat"

Illinois Testing Laboratories, Inc.

422 N. LaSalle Street, **Chicago, Illinois**

TESTING ENGINEERS AND MANUFACTURERS

Pyrometers—Portable—Wall Type—Surface Temperatures
Distant Reading Resistance Thermometers
Automatic Temperature Controllers—Air Velocity Meters

"ALNOR" VELOMETER

The Only All-Purpose Air Velocity Meter

Velometer with averaging jet used for checking velocity from supply grille.

The Velometer is a versatile direct reading air velocity meter which gives instantaneous readings of the speed of air measured in feet per minute.

Anyone can use the Velometer. No mathematical calculations, no leveling—no timing.

As its movement is actuated by the pressure or impact of the air against a swinging vane, it is essentially a pressure instrument—thus it can be scaled to not only read velocities directly, but also to read static or total pressures when using suitable jets.

Not only is the Velometer ideal for measuring duct Velocities and pressures, velocities at grilles or registers it also offers a convenient and satisfactory instrument for checking drafts, leaks around doors or windows or in ducts, velocities of ceiling outlets and similar air diffusers.

Made in standard ranges for velocity readings from 20 fpm to 6000 fpm and 3 in. static or total pressure. Special ranges available as low as 10 fpm and up to 24,000 fpm velocity and 20 in. pressure.

Jets—Several standard jets are offered providing a wide application.

Spot jets—for velocities over very small areas.

Averaging jets—for obtaining average velocities over a definite area or grille face.

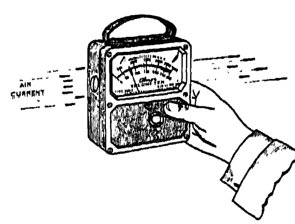
Duct jets—for determining velocities directly within ducts or pipes.

Static pressure jet—for static pressures in inches of water.

Total pressure jet—for total pressure in inches of water.

Other jets—Standard jets offered in several lengths and sizes.

Special jets—can be designed for unusual applications.



View shows how instrument is used to read low velocities without jets.

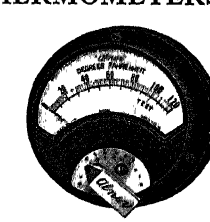
Ask for Bulletin No. 2448-D

"ALNOR" DISTANT READING ELECTRIC THERMOMETERS

The use of "Alnor" multi-point resistance type thermometers is rapidly increasing in air-conditioning, as well as for heating and refrigerator installations.

The instrument can be located in the machinery room or boiler room with the elements located on various floors in any part of the building, or outdoors, thus providing the engineers with constant and convenient temperature readings.

"Alnor" thermometers are made in several styles and sizes, both portable and mounted types.



"Alnor" round type multi-point resistance thermometer with built-in switch.

Ask for Bulletin No. 2451-A

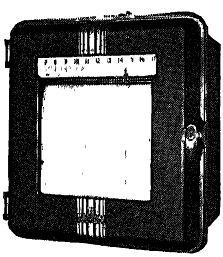
Leeds & Northrup Company

General Office and Works: 4941 Stenton Avenue, **Philadelphia, Pa.**

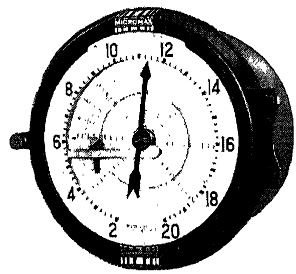
Branch Offices:

BOSTON
BUFFALO
CHICAGO
CINCINNATI
CLEVELAND
DETROIT
HARTFORD
HOUSTON
LOS ANGELES
NEW YORK
PITTSBURGH
ST. LOUIS
SAN FRANCISCO
TULSA

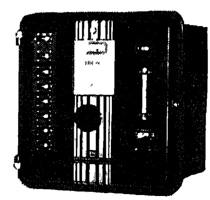
RUGGED, ELECTRICAL-BALANCE INSTRUMENTS



Model S Micromax Recorder
Records from 1 to 16 points on a single strip-chart. Extremely open record. Can operate signals. (About 1/15th size)



Model R Micromax Recorder
Records 1 or 2 points on a round-chart. Has extremely readable dial. Can operate signals. (About 1/15th size)



Panel Indicator
Hand-operated. Can be connected through selector switches to any number of points. (About 1/12th size)

Electrical Thermometers for Air Conditioning

No method for measuring temperatures fits the specific needs of air conditioning as does the three-lead, null-type resistance thermometer method. It is independent of distance and disregards all temperatures except those right at points of measurement. Thermohms (electrical resistance thermometers) can be placed anywhere—in rooms, air ducts or water lines. They are connected by simple electrical wiring to instruments at a central location. Instruments may be: Micromax Recorders, Model S for up to sixteen Thermohms, Model R, for related pairs such as wet and dry bulb; indicators with switches for any number of Thermohms; or indicating and recording combinations.

Sound in principle, this equipment is reliable in operation. Instruments and Thermohms are highly responsive, yet rugged in construction. A complete system is easy and economical to install, regardless of distances. It is easy to operate and demands minimum maintenance. Thermohms and instruments are interchangeable, and can be replaced without disturbing wiring or returning anything to the factory.

L&N Resistance Thermometers make it possible to operate efficiently; to maintain comfort or correct process atmosphere constantly . . . so that maximum return is realized on the conditioning investment.

Electrical Instruments for the Steam Plant

The facts needed to operate a modern heating plant so as to save fuel, to protect equipment, and to operate efficiently at varying loads are provided reliably by rugged L&N instruments. Readings can be indicated or recorded or both. Recorders can be equipped to operate signals or alarms that warn the operator of extreme conditions. In some cases the instruments control automatically.

Micromax Model S provides a permanent record of conditions at from 1 to 16 points on one wide-scale chart. Micromax Model R concentrates on conditions at one point, provides a permanent record, has a giant indicating dial that can be read at a glance. A Panel Indicator provides intermittent checks on conditions at one or several points.

In the heating plant, L&N measuring, signalling or controlling equipment is used for:

Metermax Combustion Control
Furnace Pressure Control
Smoke Density Analysis
Flue Gas Analysis (Percent CO_2)
Flue Gas Temperatures
Steam and Water Temperatures
Boiler-Furnace Temperatures
Electrolytic Conductivity of Water

Jrl Ad-N-225(2)

The Mercoid Corporation

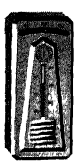
COMPLETE LINE OF AUTOMATIC CONTROLS AND MAGNETIC VALVES FOR HEATING AND AIR CONDITIONING

Main Office and Factory
4201 BELMONT AVE., CHICAGO, ILL.
Distributors and Jobbers in all Principal Cities

Branch Offices
NEW YORK, N. Y., 330 W. 34th St.
PHILADELPHIA, PA., 3137 N. Broad St.
BOSTON, MASS., 839 Beacon St.

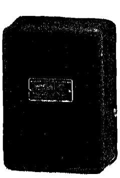
Mercoid Controls are noted for their accuracy, trouble-free service and long life. They are equipped exclusively with sealed mercury contact switches—the switch that cannot be affected by dust, dirt or corrosion. See Mercoid catalog No. 400AS showing the complete line with detailed description.

SENSATHERM



A very sensitive low voltage room thermostat. Operates on a total differential of 1 deg F. Various other types available: Two-stage Sensatherm offering new possibilities in connection with oil burners, stokers, gas burners, refrigeration, air conditioning and unit heater equipment. Dual Sensatherms for heating and cooling.

TRANSFORMER-RELAY

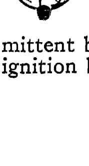


A noiseless low voltage mercury contact relay which also acts as a transformer inducing low voltage (24 volts) on the pilot circuit. Recommended for oil burners, stokers, coal blowers, unit heaters, air conditioning equipment, etc.

PRESSURE AND TEMPERATURE LIMIT CONTROL

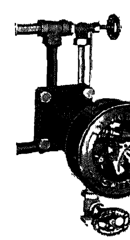
These instruments are of proven reliability and long life. The outside double adjustment with calibrated dial is a time saving feature when making adjustments. Available for steam, hot water and warm air.

VISAFLAME
The Mechanical Eye Actuated by Light



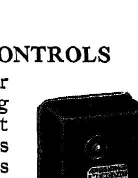
A control system for direct burner mounting. It represents a decided improvement in oil burner safety control. Operates direct from the light of the flame instead of from the heat in the stack. Used in conjunction with the K-2-I panel unit for intermittent burners and the K-2 for constant ignition burners.

COMBINED PRESSURE AND LOW WATER CONTROL



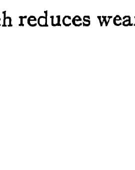
Type DA-131Q prevents firing into dry boilers and guards against building up excess steam pressure. Has quick hook-up fittings designed in accord with the A. S. M. E. Code. Instead of a packing gland, a flexible diaphragm is used which eliminates sticking or erratic operation. Has outside double adjustments. Other types of low water and boiler water feed pump controls available.

OIL BURNER SAFETY AND IGNITION CONTROL



Type JMI provides positive protection against flame or ignition failure on intermittent ignition oil burners. This control insures having ignition circuit closed before every starting operation of burner. Type JM is used for constant ignition burners.

STOKER TIMER CONTROLS



Type THV Stok-A-Timer maintains fire during periods when thermostat does not call for heat. Has heat motor which operates a gearless mechanism having but one rotating member that turns at rate of 1 rpm. The slow operating speed is a feature which reduces wear and tear.

Minneapolis-Honeywell Regulator Company

2711 Fourth Ave., So., **Minneapolis, Minn.** Cable Address: MINNREG, MINNEAPOLIS

Electric or Pneumatic Control Systems for Heating, Ventilating, Air Conditioning

BROWN INSTRUMENTS for Indicating, Recording, Controlling

Factories: MINNEAPOLIS, MINN., PHILADELPHIA, PA., WABASH, IND., CHICAGO, ILL.

Branch Offices or Distributors are located in all principal cities.

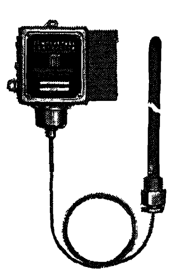
Albany, N. Y.	Dallas	Houston	New York	San Antonio
Allentown	Davenport, Ia.	Indianapolis	Oklahoma City	San Francisco
Atlanta	Dayton	Jackson, Mich.	Omaha	Scranton
Baltimore	Denver	Kalamazoo	Peoria	Seattle
Birmingham	Des Moines	Kansas City	Philadelphia	Sioux Falls, S. D.
Boston	Detroit	Los Angeles	Pittsburgh	Springfield, Mass.
Bridgeport, Conn.	East Orange	Louisville, Ky.	Portland, Me.	Syracuse
Buffalo	Eau Claire, Wis.	Lowell, Mass.	Portland, Ore.	Toledo
Butte	El Paso, Texas	Mason City, Iowa	Providence	Tulsa
Charlotte, N. C.	Fairhaven, Mass.	Milwaukee	Richmond, Va.	Washington, D. C.
Chicago	Fargo, N. D.	Minneapolis	Rochester, N. Y.	Wichita
Cincinnati	Hartford	Nashville	St. Louis	Worcester, Mass.
Cleveland	Harrisburg, Pa.	New Orleans	Salt Lake City	Youngstown
Columbus				

In Canada: Montreal, Toronto, Calgary, Vancouver, London, Winnipeg

In Europe: Amsterdam, Holland; London, England; Stockholm, Sweden

AUTOMATIC CONTROLS FOR EVERY APPLICATION

Minneapolis-Honeywell is ready to assume the complete responsibility for the supply and installation of automatic controls and instruments specified for any building that you design. M-H serivce is complete. Through our nationwide organization, we are prepared to make complete installation, supervise installation, provide periodic service or supply control equipment. Minneapolis-Honeywell can offer unbiased advice on your control requirements—manufacture and install complete electric control systems, complete pneumatic control systems, or a combination of the two.



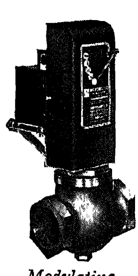
Duct Type Temperature Controller

Each Minneapolis-Honeywell office maintains a factory trained personnel. Your Minneapolis-Honeywell office will be glad to furnish you with recommended control layouts and cost estimates.

Any Minneapolis-Honeywell Office will recommend a control system before installation of equipment to produce control results after the installation has been completed.

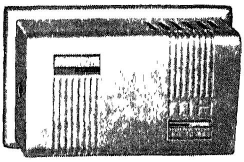
THE MODUTROL SYSTEM OF ELECTRIC CONTROL

The Modutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Electric Controls or Self-contained Automatic Valves used to govern the operation of air conditioning or heating systems other than the small domestic installations. A wide variety of both modulating and two position motors, controllers and valves are available thus making the Modutrol System extremely flexible as to the selection of control equipment to produce the desired results.



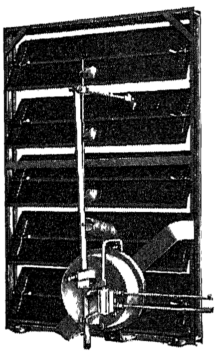
Modulating Motorized Valve

Complete electric control systems are available for those installations where precise, flexible and dependable results are required. Electric controls of the **Modutrol System** provide a dependable means of effecting modulation through the use of the "Series 90" control circuit.


"Gradustat" Pneumatic Thermostat

THE GRADUTROL SYSTEM OF PNEUMATIC CONTROL

The Gradutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Pneumatic Controls used to govern the operation of air conditioning or heating systems. Such features as infinite positioning with the Gradutrol Relay and accurate graduation of valve and damper motor makes the Gradutrol System a truly remarkable advance in pneumatic control of commercial air conditioning and space heating installations.

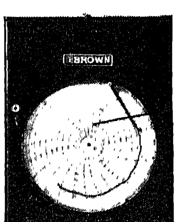

Gradutrol Motor and Damper

COMBINATION ELECTRIC AND PNEUMATIC SYSTEMS

The outstanding advantages of both the electric Modutrol System and pneumatic Gradutrol System of control may be combined in a single installation. Thus maximum flexibility and low installation cost are obtained. Minneapolis-Honeywell can offer either an electric or pneumatic system, or a combination of the two. This is your guarantee of an unprejudiced recommendation.

BROWN INSTRUMENTS

The extent to which air conditioning equipment is being used in office buildings, theatres, stores, industrial buildings, etc., has opened up a wide demand for indicating and recording resistance thermometers because the temperatures throughout these air conditioning systems should be checked periodically in order to obtain the best results at minimum operating cost. To obtain uniform conditions from modern equipment, it is necessary that the engineer in charge of operation have a visual picture of actual conditions.

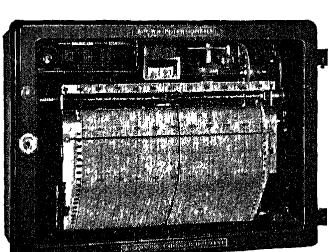


Brown Recording Thermometer and Pressure Gauge

Brown Resistance thermometers are available for indicating, recording, and controlling service and are applicable to all types of air conditioning and space heating installations.

In addition to Resistance Thermometers, The M-H Brown Instrument Division manufactures:

Thermometers	Flow Meters
Hygrometers	CO_2 Meters
Pressure Gauges	Tachometers
Vacuum Gauges	Liquid Level Gauges
Potentiometer Pyrometers	Protectoglo System


Brown Recording Resistance Thermometer

RESPONSIBILITY FOR ENTIRE CONTROL SYSTEM

Minneapolis-Honeywell Regulator Co. is equipped to assume the entire responsibility for any control installation, thereby eliminating the difficulties and misunderstandings which division of responsibility may create.

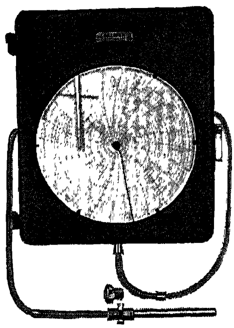
The Palmer Company

Main Plant: 2506 Norwood Ave., **Cincinnati (Norwood), Ohio**

Canadian Factory: King and George Sts., Toronto

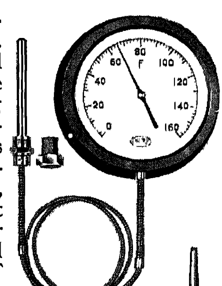
Manufacturers and Originators—"Red-Reading-Mercury" Thermometers

RECORDING THERMOMETERS

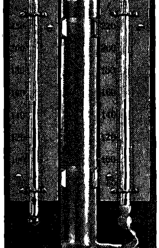


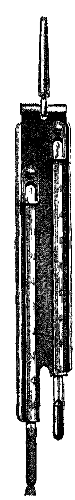
Mercury Actuated. 12 in. die-cast aluminum case. Wrinkle or Satin finish. All parts are rust-proof. Flexible armoured tubing and bulb of stainless-steel. Fittings: Plain, Union, Separable-socket and adjustable or union flange. All ranges up to 1000 F or 550 C. Guaranteed extremely accurate and sensitive. Every part strengthened for long and satisfactory service. *Write for Bulletin No. 1300.*

DIAL THERMOMETER



Mercury Actuated. 8 in. case. Black rubberized finish. Flexible armoured tubing and bulb of stainless-steel. All parts are rust-proof. Fittings: Plain, Union, Separable-socket and Adjustable or Union flange. All ranges up to 1000 F or 550 C. Guaranteed sensitive and accurate and to give long and satisfactory service. *Write for Bulletin No. 1500.*

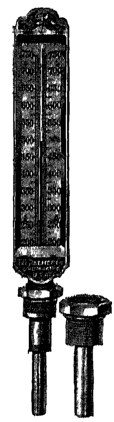




WALL HYGROMETER and SLING PSYCHROMETER

Wet and Dry bulb; Mercury tube, with RED column. Chart furnished. Guaranteed sensitive and accurate. *Send for Bulletin No. 500.*

"RED-READING-MERCURY"

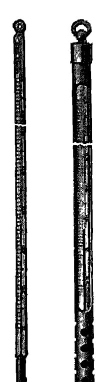


Industrial Thermometers—These mercury tubes will show a bright RED color, visible at a great distance. The color is reflected and cannot fade. (Patented by Palmer). Thoroughly annealed and guaranteed permanently accurate. Costs no more. STRAIGHT, ANGLE, SIDE - ANGLE, RECLINING AND INCLINING Case, OBLIQUE STEM, etc. 7, 9 and 12 in. case, with or without glass front. Standard 3½ in. stem and longer lengths. Fittings: Fixed Thread, Union, Separable-socket and Adjustable or Union Flange. All ranges up to 750 F or 400 C. For ranges up to 1000 F or 550 C, with plain mercury tube, borosilicate glass. *Write for Catalog No. 200-F.*

REPAIRS—To all makes of Industrial Mercury Thermometers, furnishing "Red-Reading-Mercury" tube, at no extra cost and replacing all worn or broken parts, making the thermometer as good as new. Guaranteed accurate. A trial order will convince you.

LABORATORY THERMOMETERS

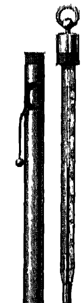
Glass engraved mercury tube; show bright RED column . . . so easy to see. With or without metal armour; Round or Lens glass; ranges to 750 F. or 400 C. Plain mercury tube borosilicate glass on ranges 1000 F. or 550 C. Correctly annealed and guarteed accurate.



POCKET THERMOMETERS . . . for quick tests. Reliable and accurate. With RED column.

− 20 + 120 F.
0 + 220 F.

Write for Catalog No. 300-D.



Penn Electric Switch Co.

Goshen, Indiana

Offices

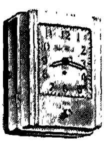
NEW YORK, BOSTON, PHILADELPHIA, DETROIT, DAYTON, CHICAGO, MOLINE (ILL.), ST. LOUIS, ATLANTA
Export—100 Varick St., NEW YORK CITY

Representatives—GARLAND-AFFOLTER ENGRG. CORP., San Francisco, Los Angeles, Seattle, Portland; SPECIALTY SALES CO., Salt Lake City; FORSLUND PUMP AND MACHINERY CO., Kansas City; VINCENT BRASS AND COPPER CO., INC., Minneapolis; D. J. BOWEN, Dallas.

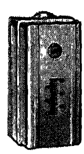
IN CANADA—POWERLITE DEVICES LTD., PENN ELECTRIC SWITCH DIV., TORONTO, ONT.

Distributors and Jobbers in All Principal Cities

Automatic Controls for Heating, Refrigeration, Air Conditioning, Pumps, Air Compressors



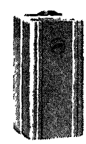
Tem-Clock



Temtrols

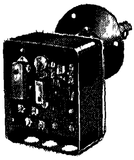
Temperature and Humidity Controls

For control of temperatures and humidity in heating, cooling and air conditioning equipment.

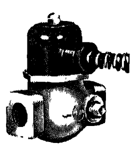


Humidistat

Heavy Duty Thermostats



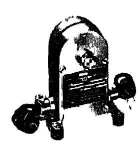
Oil Burner Stack Switches



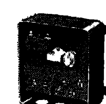
Solenoid Gas Valves

Combustion Controls for all Fuels

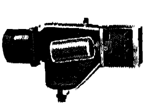
For automatic fuel burning equipment, and for stack combustion control.



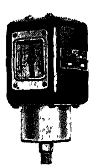
Damper Motors



Stoker Timer Relays



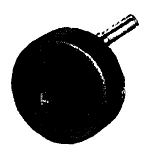
Cut-offs and Feeders



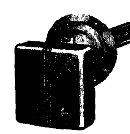
Steam Pressure Controls

Boiler and Furnace Controls

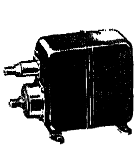
For feedwater, steam pressure, liquids and warm air.



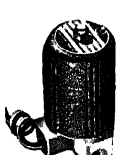
Liquid Immersion Temperature Controls



Warm Air Bonnet Controls



Refrigeration Compressor Controls

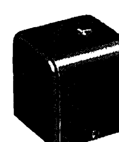


Water and Refrigerant Solenoids

Many Others

For control of refrigerants, water and air; and for pumps and compressors.

Water Valves



Pump and Air Compressor Controls

Write for catalog on Penn Controls to cover your particular applications, or 'phone the nearest Penn office or representative. Penn engineers always are available for consulation on control problems, without obligation, of course.

Penn control engineers have simplified design and production problems for others! Let them assist you.

THE POWERS REGULATOR CO.

50 Years of Specialization in Temperature and Humidity Control

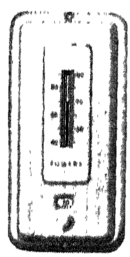
Offices in 47 Cities — See your phone directory.

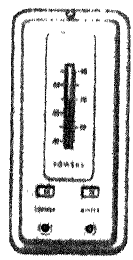
General Eastern Office
231 East 46th St., New York City

General Offices and Factory
2720 Greenview Ave., Chicago, Ill.

The Canadian Powers Regulator Co., Ltd.
195 Spadina Ave., Toronto, Ont.

Albany, Atlanta, Baltimore, Birmingham, Boston, Buffalo, Butte, Calgary, Chattanooga, Chicago, Cincinnati, Cleveland, Columbus, Dallas, Denver, Des Moines, Detroit, El Paso, Greensboro, Halifax, Hartford, Honolulu, Houston, Indianapolis, Jacksonville, Kansas City, Los Angeles, Memphis, Milwaukee, Minneapolis, Montreal, Nashville, New Orleans, New York, Oklahoma City, Philadelphia, Pittsburgh, Portland, Rochester, St. Louis, Salt Lake City, San Antonio, San Francisco, Seattle, Syracuse, Toronto, Vancouver, Winnipeg

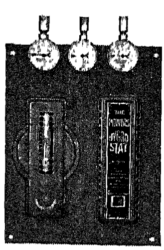


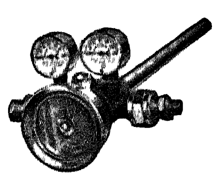


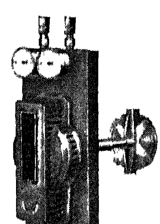
PRODUCTS—A very complete line of compressed air operated and self-operating temperature, humidity and air flow controls for automatically regulating heating, cooling, ventilating and air conditioning systems and industrial processes.

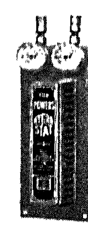
A complete line of self-operating and compressed air operated valves and regulators made for: Controlling steam heated hot water heaters and submerged type heaters; and for automatically mixing hot and cold water or steam and cold water delivering a mixture at a predetermined temperature. Dial Indicating and Recording Thermometers. Thermometer-Regulators. High pressure steam traps and pressure reducing valves.

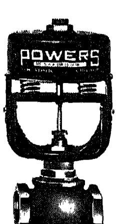
Powers Compressed Air Operated Apparatus

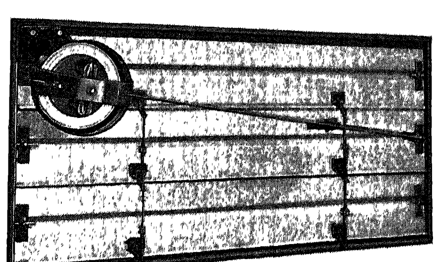


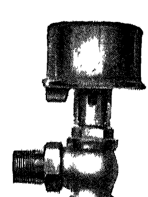




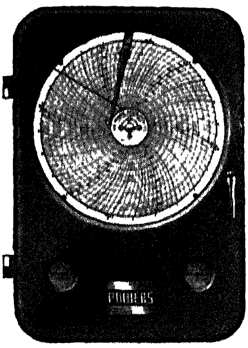


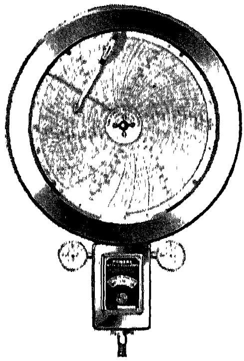


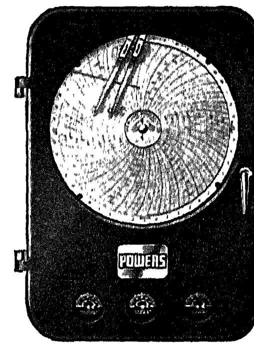


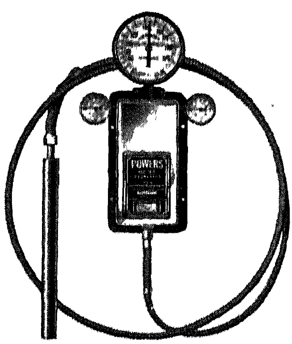


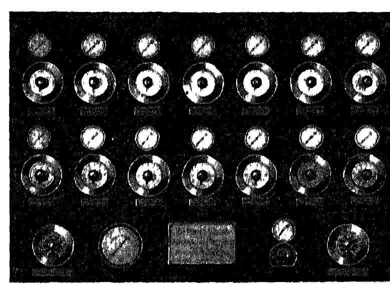
Compressed Air Operated Apparatus—Continued



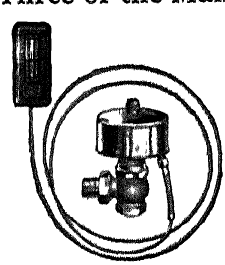


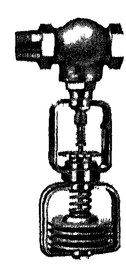


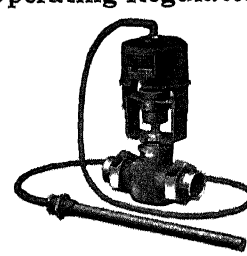




Three of the Many Types of Powers Self-Operating Regulators







POWERS ENGINEERING SERVICE

As the accurate performance of a heating, cooling, ventilating or an air conditioning system, or an industrial process is so dependent upon its automatic control equipment, and as the cost of such control is but a fraction of the entire system, the use of the proper type of regulation is always sound economy.

To secure the maximum return on the investment in automatic control equipment, it is exceedingly important that proper selection of control apparatus be made when each installation is being planned.

Forty-nine years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings have given us a wealth of experience from which you can draw in selecting the proper type of control for any purpose.

CATALOGS AND BULLETINS describing any or all of our products furnished upon request. Phone or write our nearest office. See your phone directory.

Simplex Controls

Division of The Herbusch Corporation

706 Chestnut St. **St. Louis, Mo.**

THE NEW ELECTRIC FILTER WATCHMAN

This announcement is vitally important to every purchaser, operator and owner of air conditioning equipment where filters are employed for dust removal. The air delivery and the rated capacity of heating or cooling equipment are based upon reasonably clean filters. The importance of keeping filters clean is fully recognized by manufacturers and engineers because air delivery falls off in proportion to the dust accumulation and operating costs also increase proportionately.

The new Electric Filter Watchman is a simple device which gives a visible reminder by a warning signal light on the outside of the air conditioning unit when it is time to clean or replace filters. At moderate cost it is possible to provide positive assurance that a comfort air conditioning or a forced warm air heating system is giving the performance guaranteed by the manufacturer.

Filters that are somewhat inaccessible will not be forgotten when the new Electric Filter Watchman is on the job and the operating cycle for the compressor or the gas or oil burner will be less frequent and will save many times the cost of the Electric Filter Watchman.

The Electric Filter Watchman is rapidly becoming standard equipment for many well known air conditioning units and is recognized as an essential device to assure continuous and satisfactory performance. Clean filters of the average throw-away type offer a resistance of only 1/16 to 1/10 inches S. P. but 30 days of service in many localities increases the resistance from 1/8 to 1/4 inches S. P. (See chart in Chapter 28 for change in Resistance Due to Dust Accumulation.) A few of the important industrial users of the Electric Filter Watchman are:

C. G. Conn., Ltd.
Eastman Kodak Co.
General Electric Co.
Radio Station WHAS
Sears Roebuck & Co.
Southwestern Bell Telephone Co.
S. H. Kress Company.
Western Electric Company.
Gelatine Products Co.
Wheeling Steel Corp.
Westinghouse Mfg. Corp.
Chesapeake & Potomac Telephone Co.

Engineers should include provision for the Electric Filter Watchman in their specifications and should strongly emphasize to the owner the economy of reduced operating costs and better system performance of the air conditioning or forced circulation warm air heating plant. Owners and operators of existing plants should apply the Electric Filter Watchman immediately and effect immediate savings. The Electric Filter Watchman will pay for its low cost quickly and serve indefinitely. Send for bulletins on construction details and instructions for installation and operation of the new Electric Filter Watchman.

HOW IT OPERATES

The Electric Filter Watchman is mounted outside of the air conditioning housing any place between the filter and the blower. A 5/8 inch hole is drilled in the housing and the metal tube of the Filter Watchman is inserted. One connection is then made to the fan side of the filter beginning on the upstream side by means of rubber tubing so that the pressure drop across the filter bank actuates the electric signal at a pre-set point. Setting can be made for any pressure encountered in the air conditioning field. The Electric Filter Watchman is primarily an indestructible bellows, operating by contact a simple warning light. No adjustments are necessary and there is nothing to get out of order. It will serve for the life of the air conditioning system and beyond.

Spence Engineering Company, Inc.

28 Grant Street, **Walden, N. Y.**

SPENCE METAL DIAPHRAGM "DEAD END" REGULATORS

Advantages of Spence Regulators

Dead-end Shutoff—Spence Regulators are guaranteed to hold a dead-end.

Single Seat—Spence design makes possible a balanced single seat even in large sizes.

Metal Diaphragms—Under normal conditions never require replacement.

Accurate Regulation—Regardless of fluctuations in either load or initial pressure.

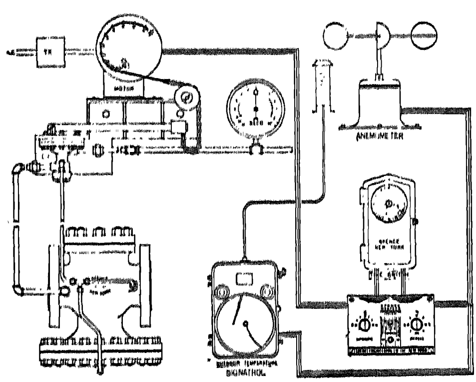
SECO Metal—Guaranteed to resist the wiredrawing action of steam.

Interchangeable Pilots—Any type of pilot will fit any size main valve.

Accessibility—Pilot is connected to main valve with unions.

No Stuffing Boxes—All main valves and most pilots are packless.

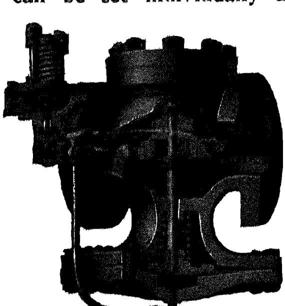
Spence Weather Compensator and Orifice Zone Control System



This simple, dependable Control, when installed on a properly designed orificed heating system, will show a substantial degree-day steam saving, at a low maintenance cost.

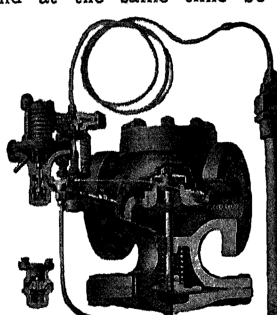
The delivery pressure of the Regulator is automatically adjusted in direct proportion to the building heat losses. In other words, as the losses become greater, steam pressure on the system is automatically increased.

Any number of zones can be controlled by one automatic Signatrol, automatic Wind Loss Compensator (Anemometer), Time Switch and Master Control Panel equipped with Manual and Automatic Dials for each zone. In this way each zone can be set individually and at the same time be under the Master Control.



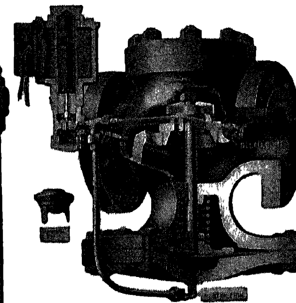
Pressure Regulator—Type ED

Designed to regulate a steady or varying initial pressure so as to maintain a constant, adjustable, delivery pressure. Applicable to heating systems, power plant operations, or manufacturing processes.



Combined Temperature and Pressure Regulator —Type ETD

Self-contained, pilot operated, dead-end. Designed to control flow of fluid to a heating or cooling element, so as to maintain a constant, adjustable temperature, and protect the element against excessive pressure.



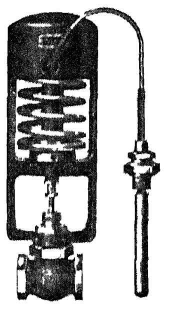
Electrically Operated Valve—Type EM

Can be opened or closed independently by an electrical switch.

Type ET—Same as ETD except pressure control is omitted.

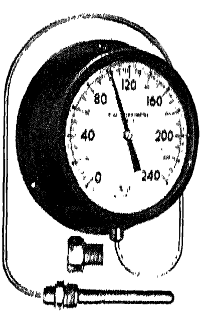
Order a SPENCE Regulator for 40 days' free trial.

Fall-O-Matic Universal Pipe Intersection Cutter.



Taylor Self-Acting Temperature Controller—Adapted for use on hot-water storage tanks, etc. Requires no auxiliary motive power.

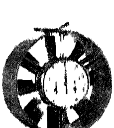
The valve can be closed at any desired temperature, or a throttling action can be obtained. Not practicable on pipe lines having steam pressure over 125 lb.



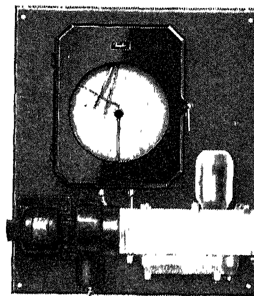
Taylor Dial Thermometers for air ducts or any application where it is desirable to have temperature readings at some distance from the thermometer bulb, as in a central control room. 8 in. dial can be read at a glance as easily as a steam gage.

Taylor Sling Psychrometer—The advantage of this form of Wet- and Dry-Bulb Hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as the whirling bulbs are subjected to perfect circulation. Two accurate etched stem thermometers are mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened.

These thermometers have scales of 20 to 120 F, graduated in ½-deg divisions. A copper case protects the tubes when not in use.

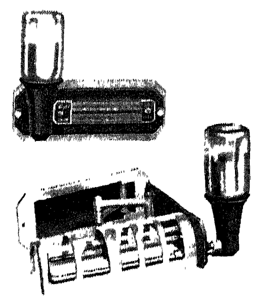


Taylor Biram's Anemometer—This instrument is ideal for measuring air velocities with the fan revolutions indicated on the dial. Available in various models for a wide range of air speeds and registration limits.

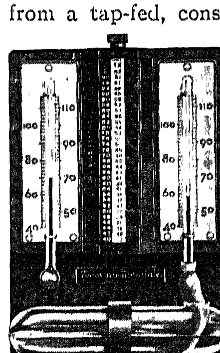


Taylor Recording Hygrometer—Records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.

Type shown has motor-driven fan for conditioned rooms or passages where circulation is poor. Furnished without fan for installations where circulation across bulb is good.



Taylor 10BG Hygrometer—For air ducts and closed compartments. Combines the accuracy of an etched stem thermometer with the ruggedness of an industrial thermometer. Complete assembly swings out on bracket for quick and easy inspection by loosening two thumb screws. Numerals etched on broad, flat tubes and pigment-filled. Tube scales mounted horizontally for quick reading. Specially designed guard allows maximum circulation with minimum risk of breakage. Water supply from beer bottle reservoir, as shown; or from a tap-fed, constant level reservoir.



Taylor Humidiguide—A handsome small hygrometer for the wall of the home, office, school or other building where a neat, easy-reading and inexpensive instrument is desired. It is self-contained, requiring no charts or separate tables. Frame is Mahogany Bakelite.

For complete information on above instruments and others designed for heating, ventilating and air conditioning, send to any of the offices listed on the previous page for new Taylor Catalog Number Five.

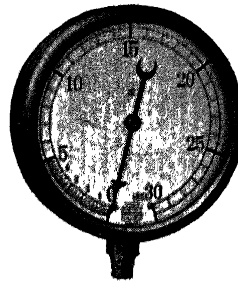
UNITED STATES GAUGE COMPANY

U. S. GAUGE CO. N. Y.

Indicating and Recording Pressure Gauges

44 BEAVER STREET · NEW YORK

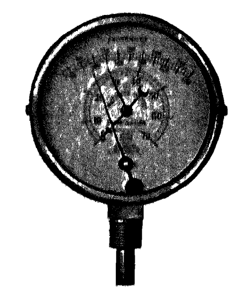
FACTORY · SELLERSVILLE · PENNSYLVANIA
BRANCHES · NEW YORK · CHICAGO · PHILADELPHIA
BOSTON · CLEVELAND · DETROIT · ST LOUIS
HOUSTON · SEATTLE · LOS ANGELES · MONTREAL



U. S. GAUGES—U. S. Gauges are made in all standard sizes from 2 in. to 12 in. dial inclusive for pressures up to 50,000 lb and for vacuum. Gauges may be supplied with cast-iron, cast-brass, drawn steel, or drawn brass cases for wall mounting or flush mounting. For severe service requirements we can supply long wearing hardened steel movements or bushed movements.

For service on Steam Heating Systems the following gauges may be supplied—

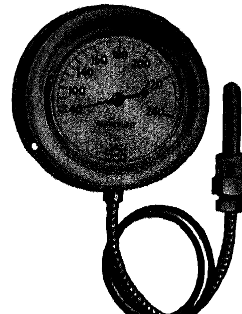
Steam Gauges . . . Compound Pressure and Vacuum Gauges . . . Retard Gauges . . . Compound Retard Gauges . . . Steam Gauges with Internal Siphons.



For service on Hot Water Heating Systems the following instruments and gauges may be supplied

Altitude Gauges . . . Tank-in-Basement Gauges . . . Altitude and Pressure Gauges . . . Combination Altitude Gauges, and (a) Bimetal Thermometers, (b) Glass Tube Thermometers, (c) Vapor Tension Distance Type Thermometers . . . Glass Tube Hot Water Thermometers.

U. S. RECORDING GAUGES—U. S. Recording Gauges are supplied in 8½ in., 10 in. and 12 in. sizes for pressures up to 50,000 lb and for vacuum. These Recording Gauges can be supplied with either cast-iron or cast-brass cases for wall mounting or flush mounting. Pen arms are made of non-corrosive metal. Especially designed clock movements are used. Charts can be furnished for customary time periods.



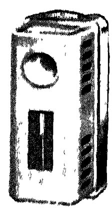
U. S. DIAL THERMOMETERS—U. S. Dial Thermometers are of the vapor tension type with open scale reading in the center and upper portion of the scale, or of the glass filled type with even scale reading. Cases may be cast-iron, cast-brass, drawn steel, or drawn brass for either wall or flush mounting. Supplied in all standard sizes from 2 in. to 12 in. dial inclusive, for temperature ranges from minus 40 deg F to 800 deg F. Furnished with rigid connection bulb or with bulb at end of flexible capillary tubing up to 100 ft long.

White-Rodgers Electric Company

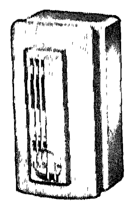
1293 Cass Avenue, **St.Louis, Mo.**

NEW YORK CITY SYRACUSE BOSTON CHICAGO CLEVELAND COLUMBUS DETROIT INDIANAPOLIS

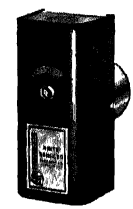
Distributors in Principal Cities



Line voltage Thermostat for Unit Heater and Air-Conditioning Installations.



Low Voltage Room Thermostat—anticipating type.



Steam Pressure Control—for safety limit service.

PUT "HYDRAULIC ACTION" TO WORK FOR YOU

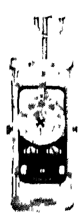
Around the new and basic principle of **Hydraulic Action** has been designed an outstanding line of temperature and pressure controls to fit every requirement.

Hydraulic action contributes these important new features to the field of automatic temperature control:

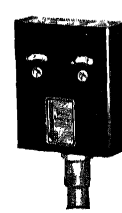
1. *Visible, uniformly calibrated dials.*
2. *Easily set differential adjustor.*
3. *Fast acting thermal elements.*
4. *Combination controls with independently adjustable switches.*

High capacity switch—the tremendous force available with Hydraulic Action has resulted in a sturdy switch with Underwriters' approved rating of 25 amp 120 v, 15 amp 240 v, 1½ hp (R.I.) 120-240 v AC.

Take advantage of Hydraulic Action on your next installation. Specify White-Rodgers Controls. The latest condensed control catalog is awaiting your request. Write for it today!

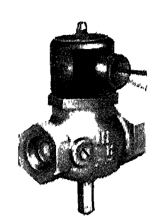


Single speed fan control-cover removed showing visible dial.

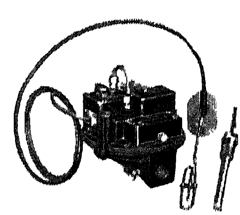


Dual Immersion Control—Limit-Circulator or Summer-Winter service

Stoker Timer—with or without night set-back or fused line switch.



Solenoid Gas Valve—High plunger torque and silent operation.



Diaphragm Gas Valve with puff bleed and built-in mechanical limit control.

International Heating & Ventilating Exposition

THE AIR CONDITIONING EXPOSITION

Permanent Address—Grand Central Palace, **New York, N. Y.**

EXPOSITION HELD

The first in Philadelphia, 1930.
The second in Cleveland, 1932.
The third in New York, 1934.
The fourth in Chicago, 1936.
The fifth in New York, 1938.
The sixth in Cleveland, 1940.
The seventh in Philadelphia, 1942.

Subsequent Expositions will be held on alternate, even numbered years.

These are held coincident with the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and are directed by the International Exposition Company, under the auspices of the A.S.H.V.E.

EXHIBITORS

Comprise leading firms in each phase of the industry; number has varied from 150 to 327 exhibitors.

EXHIBITS

These range from and comprise all the types of articles discussed or advertised in this copy of THE A.S.H.V.E. GUIDE.

1. *The* COMBUSTION *Group:*
   Furnaces, burners (coal, oil and gas), grates, stokers, boilers, radiators (various types), refractories and auxiliaries.
2. *The* OIL BURNER *Group.*
3. *The* HYDRAULIC *Group:*
   Water feeders, water heaters, pumps, traps, valves, piping, fittings, expansion joints, pipe hangers, etc.
4. *The* STEAM HEATING *Group:*
   Vapor heating and steam specialties.
5. *The* HOT WATER HEATING *Group.*
6. *The* AIR *Group:*
   Warm Air furnaces and stoves, registers and grilles, cooling towers, air filters, motors, fans, blowers, conditioning equipment, ventilators (room and industrial types), unit heaters, etc.
7. *The* AIR CONDITIONING *Group:*
   Equipment which circulates and filters the air, in summer dehumidifies and cools; in winter heats and humidifies, and does all these in proper season for complete, all year round air conditioning.
8. *The* CONTROL *Group:*
   Instruments of precision for indicating, controlling or recording temperature, pressure, volume, time, flow, draft or any other function to be measured.
9. *The* REFRIGERATING *Group:*
   Compressors, condensers, cooling apparatus, contingent apparatus and refrigerants.
10. *The* CENTRAL HEATING *Group:*
    Apparatus and materials especially designed or adapted to the uses of central heating and central heating station supplies.
11. *The* INSULATING *Group:*
    Structural insulators (refractory and cellulose materials), asbestos, magnesia clays and combinations thereof, pipe and conduit covering, etc., weather-stripping, etc.
12. *The* MISCELLANEOUS *Group:*
    Electric Heaters, boiler and pipe repair alloys, liquids and compounds, tools of all kinds, and equipment not specifically included in the above groups, but related thereto.
13. *The* MACHINERY AND GENERAL EQUIPMENT *Group.*
14. BOOKS AND PUBLICATIONS.

VISITOR ATTENDANCE

Comprises a registered attendance invited to the exposition and includes:

(Figures are 1940 analysis)

INDUSTRIES

Governmental	401
Distribution Channels	
Contractors, Dealers, Jobbers, Supply Houses	7,031
Home Owners	333
Industrial Users	9,371
Professional and Service Organizations	680
Public Utilities	900
Real Estate Management and Operation	630
Educational Institutions	500
Miscellaneous	765
TOTAL	20,652

OCCUPATIONS

Executive	11,433
Construction	2,632
Operation	2,353
Technical	2,688
Not Classified including Educators, Publishers, Home Owners, etc	1,546
TOTAL	20,652

Industrial Expositions in America lead the expositions of the world in style, business effectiveness, industrial influence and educational value. This Exposition stands among the leaders in Industrial Expositions in America. It is an educational institution which biennially brings together the research developments and improvements in equipment and materials for use in heating, ventilating and air conditioning all types of buildings.

HEATING SYSTEMS

•

Steam and hot water heating systems with their many parts and accessories are classified according to their specific type of design and the service required. These systems and their component parts include:

HEATING SYSTEMS (p. 968-991)

Combinations of parts forming steam vapor and vacuum systems and hot water systems.

Technical data on steam heating systems are contained in Chapter 13; hot water systems in Chapter 15. Other references to heating systems will be found in the Index to the Technical Data Section.

BOILERS (p. 992-1013)

Water tube, fire tube and firebox types; cast iron and steel construction; for coal, coke, gas or oil firing.

Technical data on heating boilers are given in Chapter 11.

In connection with steam and hot water heating systems various types of radiators and convectors are required. Complete manufacturers references will be found in the Index to Modern Equipment—pages 1097-1120.

Technical data is contained in Chapter 12.

BURNERS (p. 1014-1024)

Automatic fuel burning equipment suitable for use as an integral part of heating boilers and furnaces, and also for conversion of hand-fired heaters to automatic operation.

Technical data are given in Chapter 9.

PUMPS (p. 1025-1028)

For use in conjunction with heating systems, and other purposes in heating, ventilating and air conditioning service; and for handling air, gases, ammonia, brine and other refrigerants.

References to technical data on pumps will be found in the Index to the Technical Data Section.

PIPE AND FITTINGS (p. 1029-1045)

Iron, steel, wrought iron, copper, brass—seamless or welded.

Technical data will be found in Chapter 17.

SPECIALTIES (p. 1031-1041)

Feed water devices, pressure and draft regulators, combustion controls, strainers, traps, valves, etc.—all essential for efficient operation of heating equipment.

References to technical data on heating specialties may be found in the Index to the Technical Data Section, each indexed under its respective title.

Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

Barnes & Jones
INCORPORATED

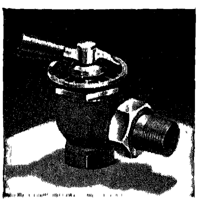
129 Brookside Avenue — Boston, Mass.

New York Office: 101 Park Avenue

Barnes & Jones Vapor and Vacuum Systems of Steam Heating; Modulation Valves; Adjustable-Orifice Radiator Valves; Packless Quick-Opening Radiator Valves; Thermostatic Radiator Traps; Thermostatic Trap Replacement Units; Condensators (Boiler Return Traps); Float and Thermostatic Traps; Strainers; Damper Regulators; Gages; Systems of Zone Control for Steam Heating.

Complete Catalog on Request

Modulation Valves, Type K · Packless Quick Opening Valves, Type F



Types K and F Valves have non-tarnishable indicating dial, non-rising stem, renewable disc seat. Tail piece extra heavy. Extra long to facilitate installation. Three models: lever handle, wheel handle, lock shield. Type F Valve furnished with wheel handle only.

Type K Valve

Size	½″	¾″	1″	1¼″
Cap. Sq Ft Rad.*	30	60	100	180

Type F Valve

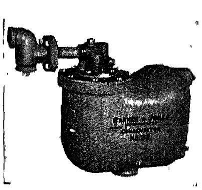
Size	½″	¾″	1″	1¼″	1½″	2″
Cap. Sq Ft Rad.*	30	60	100	180	270	400

*Based on 2 oz pressure differential.

Adjustable Orifice Valves, Type H

May be adjusted for different capacities after installation. At all times provides indication of the adjustment. Operation is quiet. Unauthorized tampering with adjustment is virtually impossible.

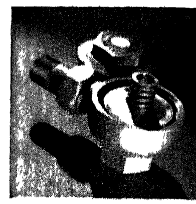
Condensators



For returning water of condensation to boiler from open return line systems independently of boiler pressure, without change in operating conditions, air binding, or admitting steam to the return side.

No.	31	32	33	34	35	36
Cap. Sq Ft Rad.*	700	1600	3500	6000	10,000	16,000

Thermostatic Radiator Traps

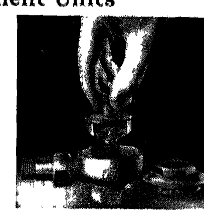


Sturdily made to precision standards. Sensitive in operation. Provide instant discharge of air and water, prevents passage of steam. Contains unique Cage Type Thermostatic Unit, which carries its own thermostatic element, valve piece and valve seat, factory calibrated and locked in correct adjustment.

Trap No.	120	12	123	124	134	13	14
Inlet Tapping	½″	½″	½″	½″	¾″	¾″	1″
Outlet Tapping	½″	¾″	½″	¾″	¾″	¾″	1″
Cap. Sq Ft Rad.*	200	200	400	400	400	700	1200

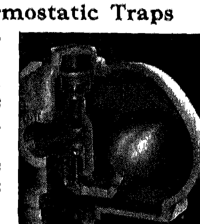
*Based on 1½ lb pressure differential.

Thermostatic Radiator Cage Replacement Units



Offer complete and reliable trap renewal in practically every make of thermostatic trap. You simply (1) remove the old cover and unit, (2) insert the new Barnes & Jones Cage Unit, (3) replace the cover, and the old trap will operate with its original efficiency.

Float and Thermostatic Traps



Handle large and sudden condensation loads. Large air and water capacity. Large float assures instant opening of the discharge valve. Cage Type Thermostatic Unit assures quick elimination of air.

Trap No.	41	42	43	43A	44B	45B
Inlet Tapping	¾″	1″	1¼″	1½″	1½″	2″
Outlet Tapping	¾″	1″	1¼″	1½″	1½″	2″
Cap. Lb. Water per Hour*	200	700	1200	1200	2400	5000

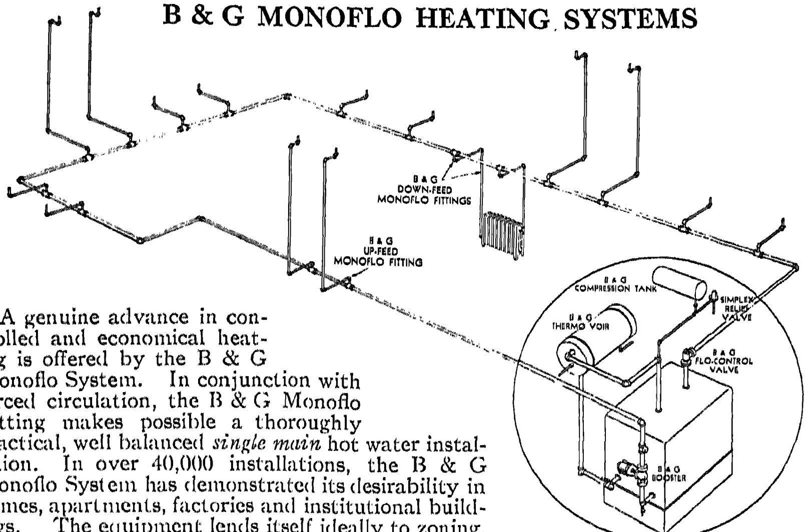
*Based on 2 lb pressure differential.

Bell and Gossett Company

3000 Wallace Street — **Chicago, Ill.**

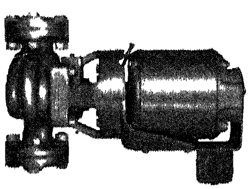
HOT WATER SYSTEMS AND SPECIALTIES

B & G MONOFLO HEATING SYSTEMS



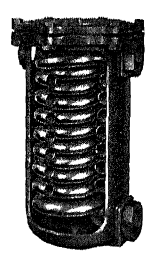
A genuine advance in controlled and economical heating is offered by the B & G Monoflo System. In conjunction with forced circulation, the B & G Monoflo Fitting makes possible a thoroughly practical, well balanced *single main* hot water installation. In over 40,000 installations, the B & G Monoflo System has demonstrated its desirability in homes, apartments, factories and institutional buildings. The equipment lends itself ideally to zoning, yet is exceedingly simple in application.

EQUIPMENT REQUIRED



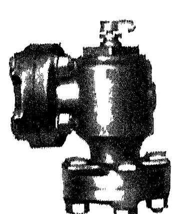
B & G Booster

An electrically-driven centrifugal pump, which mechanically circulates hot water through the system—distinguished by genuine oil lubrication, patented water-tight seal and precision manufacture throughout.



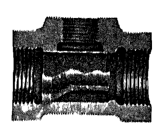
B & G Indirect Water Heater

Any one of five B & G Heater types can be installed to furnish year around domestic hot water at smallest possible cost.



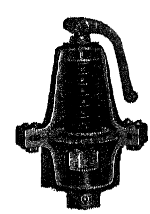
B & G Angle Flo-Control Valve

This valve, installed in the main, controls circulation of hot water to radiators, permitting summer operation of the Indirect Water Heater. It also helps maintain a uniform room temperature during the heating season.



B & G Monoflo Fitting

A correctly engineered fitting, installed in the main at radiator connections, which diverts water into the radiators. Its design assures a balanced distribution of water without introducing excessive resistance.



Simplex Relief Valve

For boiler protection.

See the B & G Handbook for Complete Engineering Data

C. A. Dunham Company

Administrative and General Offices

450 E. Ohio Street, **Chicago, Ill.**

Factories: Marshalltown, Iowa; Michigan City, Ind.; Toronto, Canada; London, England

C. A. Dunham Co., Ltd.,
1523 Davenport Road,
Toronto, Ont., Canada

COMFORT CONDITIONED by DUNHAM

C. A. Dunham Co., Ltd.,
(Of the United Kingdom)
Morden Road, London, S.W. 19,
England

See "Dunham Heating Service" in local telephone directory in all principal cities.

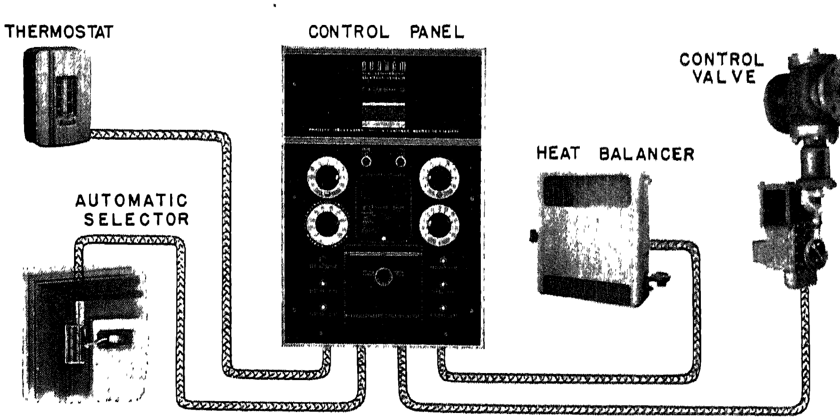
The accumulated experience of the entire Dunham Organization is put at the disposal of the Heating Ventilating and Air Conditioning Engineer. This cooperation is available for *Modernization Work*, as well as for new construction in industrial, commercial, housing and other projects.

The Dunham Sub-atmospheric Steam Heating System maintains desirable temperatures throughout a building by automatic control of both steam temperature and steam volume.

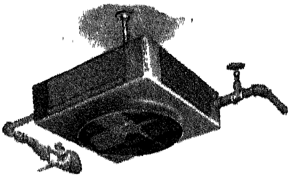
The system is a simple two-pipe system in which all the essentials of circulation, distribution and control are co-ordinated. Control of the temperature of the steam is accomplished by controlling the pressure or vacuum of the steam in the supply piping and radiators to balance exactly the heat input with building-heat-loss.

The Dunham Sub-atmospheric System distributes a varying supply of heat equally, automatically and continuously through the heated space. Desirable building temperatures are automatically maintained under varying weather conditions. A positive *continuous* circulation is maintained as a fundamental function of the system. This maintains unusually constant temperature levels throughout the building.

The Control is Fully Automatic. The RT (Resistance Thermometer) Control Equipment, which is an integral part of the System, is fully automatic. Beginning with a maximum radiator heat output obtained by steam circulation at a pressure of 2 pounds and a temperature of 218 F or more as required, the output is progressively reduced according to the demands of the weather, by a reduction in the rate of steam admission to the system, which automatically causes a reduction in steam pressure and temperature so that steam may be circulated at varying temperatures down to about 133 F. Further reduction in heat-output is obtained by partial filling of radiators with sub-atmospheric steam until the point is reached at which the need for heat ceases and the supply of steam is completely shut off. The distribution of the steam supply is automatically maintained under all variations in supply by the co-ordinated functioning of the Traps, Pump, Differential Controller and Regulating Orifices at radiator inlets.



DUNHAM UNIT HEATERS

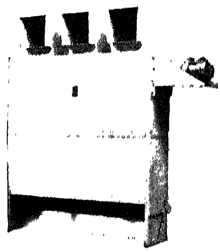


Type C

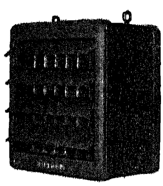
Type C—A specially designed heater for industrial and commercial structures. Designed for moving a large volume of heated air at low temperature down to working levels making possible a heated air stream discharging directly floorward. 10 sizes, 3 types of outlet diffusers, capacities 100 to 2000 sq ft EDR.

Type R—13 standard sizes. Capacities from 381 to 3355 sq ft EDR. Also in suspended type. Both types either direct connected or belt drive.

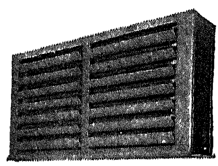
Type V—36 sizes. Capacities from 95 to 1100 sq ft EDR.
Type D—12 sizes. Capacities from 558 to 2360 sq ft EDR.
Type F—12 sizes. Capacities 1074 to 4550 sq ft EDR.



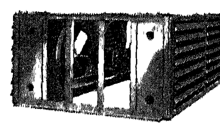
Type R



Type V

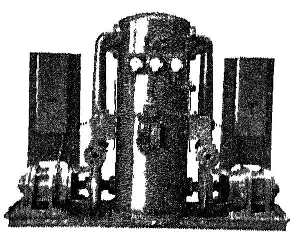


Type D



Type F

DUNHAM PUMPS



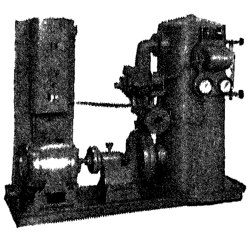
Type VRD

Tested and Rated with A.S.H.V.E. Code and Code of Vacuum Return Line Heating Pump Manufacturers' Section of Hydraulic Institute.

Vacuum Pumps

Type VRD—Designed for use with Dunham Sub-atmospheric Steam Heating. Capable of maintaining whole systems under vacuums as high as 25 in. Built in 9 sizes. Capacities 2500 to 65,000 sq ft EDR.

Type VR—Meets all code tests for air and simultaneous air and water handling capacities. No moving parts or close clearances in exhauster unit. Built in 11 sizes. Capacities 2500 to 150,000 sq ft EDR.



Type VR

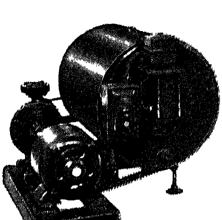
Condensation Pumps

Provide automatic return of water of condensation to boilers for gravity systems or steam process equipment. Pump and motor assembled on rigid cast iron base. Bronze fitted centrifugal pump has non-corrosive shaft. Enclosed type Impellor. Liberal size ball bearings. Receiver tank equipped with float switch and push button starting switch with over-load protection.

Type CII-B—66 sizes of varying capacities and discharge pressures. Capacities 2000 to 50,000 sq ft EDR; 60 cycle d.c. or a.c. 1750 rpm; 25 or 50 cycle a.c., 1450 rpm.

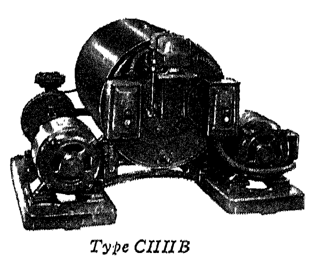
Type CIII-B—49 sizes of varying capacities and discharge pressures. Capacities 2000 to 50,000 sq ft EDR; 60 cycle d.c. or a.c. 3450 rpm; 25 or 50 cycle a.c., 2850 rpm.

Type CV—48 sizes of varying capacities and discharge pressure. Capacities 2000 to 25,000 sq ft EDR.

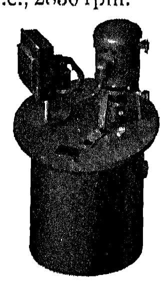


Single Unit Type CII or CIII Model B ←

Duplex Unit Type CII or CIII Model B →



Type CIIIB

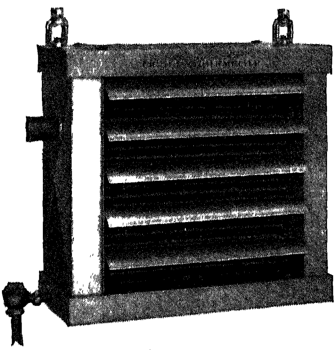


Type CV

THERMOLIER

Patented

THE GRINNELL UNIT HEATER



Industrial Type

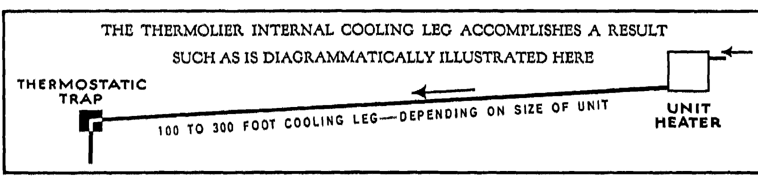
De Luxe, Industrial and Factory Types—125 Lb W.S.P.

Thermolier is a ruggedly built unit heater whose efficiency and dependability have been proved by actual performance in field service. Thousands of them are installed in industrial buildings and commercial structures of all types of occupancy.

Thermolier has 14 points of superiority, the most outstanding of which is the internal cooling leg built right into the unit, an exclusive Thermolier feature. See drawing below.

Radiation is from brass-finned seamless copper U-tubes rolled into a cast-iron tube sheet. No solder is used for strengthening joints and there are no flat horizontal surfaces to catch dirt.

Units may be controlled manually or automatically, singly or in groups. Installation and piping are extremely simple and inexpensive, hence the unit may be moved from one



location to another at small cost if found desirable on account of changes in building or occupancy. The complete line includes 27 Models in DeLuxe Type, and 35 Models each in Industrial and Factory Types.

Thermoliers provide maximum distribution of heat without objectionable drafts.

Specifications

Fan—Grinnell special of rugged construction. **Motor**—heavy duty, oversize, enclosed, moisture-proof. **Housing**—Art Metal Slate gray finish with chromium trim on DeLuxe Types. Copper on Industrial Type with rubbed lacquer finish; steel on Factory Type finished in gray lacquer. **Frame**—Heavy pressed steel, providing rugged support for motor and fan. **Special Features**—Adjustable swivel hanger rod couplings; louvers rigid, but easily adjustable: integral cooling leg insuring perfect drainage through one thermostatic trap for pressures up to 25 lb.

For pressures not exceeding 125 lb, a thermostatic trap of proper construction can be used and should be attached directly to the unit.

CAPACITIES

60 F Entering Air Temperature—2 lbs Steam Pressure

Model Nos.	Btu per Hour	Model Nos.	Btu per Hour	Model Nos.	Btu per Hour
21	35,600	46L	63,500	81L	143,600
21L	28,350	51	98,600	91	196,000
22	37,100	51L	73,700	91L	160,600
26	38,650	57	101,300	101	237,800
26L	31,750	57L	83,000	101L	205,400
31	48,700	61	102,100	111	275,300
31L	36,200	61L	77,500	111L	229,700
37	62,200	66	128,700	141	325,300
41	70,900	66L	110,500	141L	269,700
41L	56,600	71	151,700	181	373,300
44	84,800	71L	126,800	181L	303,200
46	86,400	81	174,900		

Data Book covering other pressures and temperatures, dimensions and complete installation information on application. Address GRINNELL COMPANY, INC., 277 West Exchange Street, Providence, R. I.

GRINNELL ADJUSTABLE PIPE HANGERS AND SUPPORTS

One of the chief advantages of Grinnell Adjustable Hangers is that they permi adjustment of pipe lines after installation, thus obviating the necessity of turnbuckles o the removal of hangers. Their time and trouble-saving qualities during installation ar equally exceptional. Below are shown a few Grinnell Hangers and Supports of par ticular interest to heating engineers. Send for Hanger Catalogue showing complete line

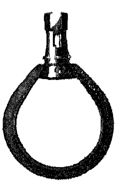


Fig. No. 101
Solid Ring

Adjustable Swivel Rings (Patented)

These Malleable Iron Adjustable Swivel Rings can be used with Coach Screw Rod or Machine Threaded Rod in connection with practically any type of Ceiling Flange, Expansion Case, Insert, etc.

Adjustment of at least 1½ in. is secured by turning Swivel Shank. Swivel Shank automatically locks, preventing loosening due to vibration in the pipe line.

The Split Ring permits adjustment either before or after Ring is closed. A wedge type pin is loosely but inseparably cast into the hinged section for fastening this section after pipe is in place.

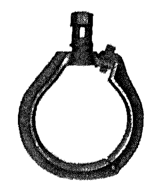


Fig. No. 104
Split Ring

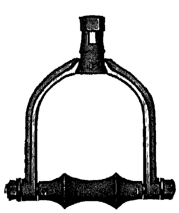


Fig. No. 174
Swivel Pipe Roll

Adjustable Swivel Pipe Rolls (Patented)

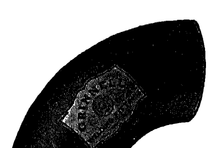
An adjustable type of pipe roll using a single hanger rod. Swivel Shank allows vertical adjustment and automatically locks, preventing loosening from vibration.

CB-Universal Concrete Inserts
(Patented)

Made of air furnace malleable iron, in one body size, to take a special removable nut, tapped for ⅜ in., ½ in., ⅝ in. or ¾ in. rod as required. Nuts automatically lock by means of V-type teeth on both insert and nuts.

Fig. No. 282
CB-Universal Insert

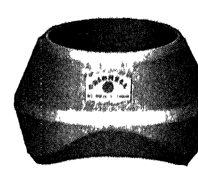
GRINNELL WELDING FITTINGS



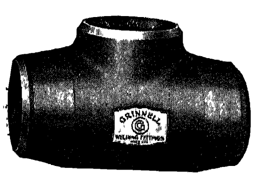
90° Elbow, Long Turn

Grinnell Welding Fittings are made from Seamless Steel Pipe or tubing and possess the same physical characteristics as standard, extra strong and o.d. steel pipe or seamless steel pipe of comparable size. They can be used under the same conditions, pressures and temperatures as the pipe itself.

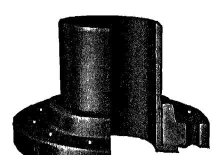
Welding faces for all plain circumferential Butt Welds are scarfed or beveled to the regulation 45 deg. angle with 1/16 in. square end on inside of fitting. Angles of bevel other than 45 deg. can be furnished on special orders.



Welding Outlet



Welding Tee



Lap Flanged Welding Neck

Threaded Outlet

William S. Haines & Company

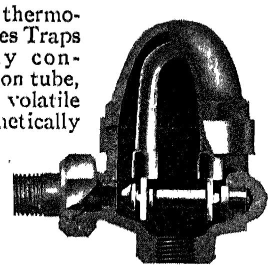
12th and Buttonwood Sts., **Philadelphia, Pa.**

Manufacturers of

EQUIPMENT FOR VAPOR AND VACUUM HEATING SYSTEMS

PRODUCTS—Haines Vento Radiator Traps, Medium Pressure and Blast Type Traps, Combined Float and Thermostatic Traps, Air Eliminators, High Pressure Thermostatic Traps, Boiler Return Traps, Packless Radiator Valves and Modulating Supply Valves.

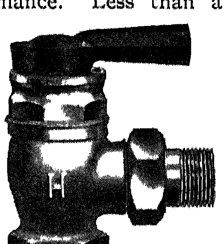
HAINES RADIATOR TRAPS



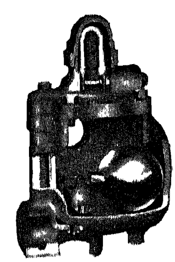
The operating thermostat in all Haines Traps is a specially constructed Bourdon tube, charged with a volatile fluid and hermetically sealed. It is the expansion and contraction of the fluid, under varying temperatures, that furnishes the operating power. The thermostatic element is outboard the valve seat closing the valve against the flow of steam.

HAINES MODULATING VALVES

The seat and carrying member construction assures positive leak proof performance. Less than a full turn of the handle completely opens or closes the valve. This valve is packless and made in sizes from ½ in. to 2 in. Can be furnished with wheel or lever handle or lock-shield.



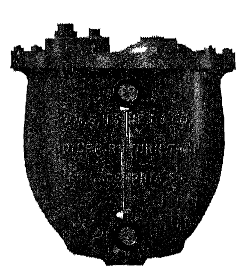
HAINES F & T TRAPS



This trap is designed for handling large quantities of condensation such as occur at main line drip points, unit heaters, hot water generators, etc. This trap cannot become air bound as it has a thermostatically controlled bypass. It is light enough to be supported in the pipe line.

HAINES BOILER RETURN TRAPS

For vapor and atmospheric heating systems. Prevents cracked boilers. Assures positive circulation by venting the air and returning the water of condensation to the boiler irrespective of boiler pressure. Weighted valve mechanism prevents wire drawing of valves. This trap has no stuffing boxes or packed joints to leak air or water.



All Haines material is ruggedly constructed to assure long life and accurately designed for economical operation.

Each device is individually tested, factory adjusted and guaranteed.

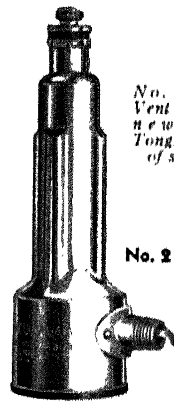
Hoffman Specialty Co., Inc.

Executive Office
77 Bedford St., **Stamford, Conn.**
Main Office and Factory: **Waterbury, Conn.**
Sales Representatives in Principal Cities

Manufacturers of Adjustable Port Radiator Venting Valves, Quick Vents and Air Eliminators for One and Two Pipe Steam and Vacuum Systems; Hoffman Supply Valves, Traps and Basement Specialties for Controlled Heat Systems, Air Conditioner Hoffman-Economy Vacuum and Condensation Pumps, and Hot Water Controlled Heat Equipment.

AIR VALVES

The Nos. 1, 1A and 40 are used for venting radiators on One and Two Pipe oil or gas automatic fired Steam Systems, and the Nos. 4, 4A, 75 and 75A are used in conjunction with these valves for venting steam mains, risers and other quick venting service.



No. 2 Vacuum Vent Valve with new "Short Tongue" in place of siphon tube.

No. 2

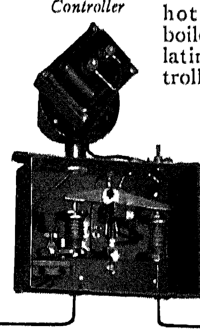
VACUUM VALVES

The Nos. 2, 2A Vacuum Air Valves feature the Hoffman Double Air Lock consisting of the vacuum check and vacuum diaphragm. These valves are for use on coal burning hand or stoker fired One Pipe Vacuum Systems; and for venting ends of steam mains or heating risers, where it is also desired to prevent the return of air into the system, the Nos. 6, 16, 16A, 76 and 76A vents are used.

HOT WATER CONTROLLED HEAT EQUIPMENT

The Hoffman Temperature Controller is connected by capillary tubing to the Outdoor Temperature Bulb, and to the Water Temperature Bulb installed in the supply main. Variations in outdoor and circulating water temperatures are instantly transmitted by these two Bulbs to the Temperature Controller which electrically opens or closes the Control Valve.

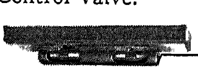
Temperature Controller



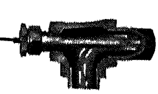
The Hoffman Control Valve. Admission of hot water from the boiler into the circulating system is controlled by this valve. It is opened and closed electrically when actuated by demands for more or less heat from the Hoffman Temperature Controller.

Control Valve

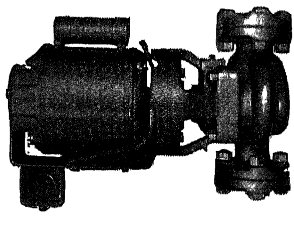
Available in sizes to correspond with Hoffman Circulator.



Outdoor Temperature Bulb ocated on exterior of building



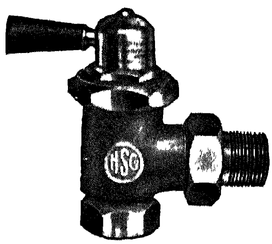
Water Temperature Bulb



Hoffman Circulator

The Hoffman Circulator is a centrifugal pump of large capacity, low power consumption and furnished in all standard sizes. It is installed in the return main and operates continuously except when outdoor temperature rises above 65 deg.

HOFFMAN CONTROLLED HEAT



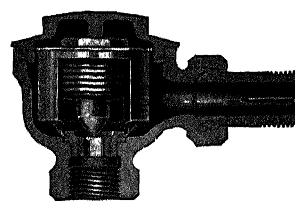
No. 7 Modulating Valve

A Hoffman Controlled Heat System consists of the No. 7 Adjustable Orifice Modulating Valve on the supply end of the radiator, the No. 8-A Thermostatic Trap on the return end and either a Hoffman Differential Loop (for coal-fired installations operating at pressures up to 8 oz), or a Boiler Return Trap where higher pressures are encountered, for returning the condensate to the boiler.

SUPPLY VALVES

Besides the No. 7 Adjustable Orifice Modulating Valve the Nos. 37 and 47 series (not illustrated) represent a complete line of Packless Supply Valves that meet the exacting requirements of architects and engineers.

THERMOSTATIC TRAPS



No. 8A—½ in.

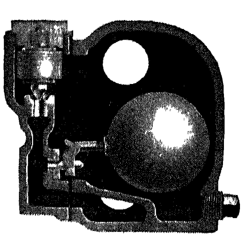
The line of Bellows Type Thermostatic Traps, with hydraulically formed and tested bellows, consists of the Nos. 17-A, 8-A and 9-A, and are principally used for low pressure steam or vapor systems. These traps have nominal capacities from 200 sq ft up to 700 sq ft of radiation. The Nos. 8-A and 9-A have renewable elements, which combine the thermostat, valve pin and renewable seat into a single unit.

The No. 10-A Hoffman Trap, which is equipped with waterhammer proof bellows, 1 in. connection, has a nominal capacity of 2800 sq ft.

The Nos. 8 and 9 Traps have a thermostatic element consisting of three chambers each having a top and bottom diaphragm. These chambers are all joined together and the complete thermostatic member is housed in a cage and is not attached to the valve body or cap. This allows the thermostatic element and valve pin to be easily removed and replaced without adjustment. These traps range in sizes from ⅜ in. to 1 in., are medium pressure traps, and are recommended where pressures up to 50 lb are encountered.

The Nos. 20-H and 21-H High Pressure Traps are equipped with waterhammer proof bellows, for use on pressures up to 125 lb. Available in ½ in. to 1 in. connection.

DRIP AND HEAVY DUTY TRAPS



No. 60 Series Trap

Where large amounts of condensation are encountered, it is recommended to use one of the float and thermostatic traps, which are available with or without the thermostatic element. These traps are available in large capacities and are mainly used for venting and dripping risers, steam mains, unit heaters, blast coils, etc. These traps are made in four different pressure ranges 15 lb, 30 lb, 60 lb, and 125 lb.

VACUUM AND CONDENSATION PUMPS

The Hoffman-Economy line of Vacuum and Condensation Pumps offers a dependable method of economically returning the condensation from larger heating systems to the boiler. These pumps are made in single and duplex units, for varying capacities and pressures.

HOFFMAN SALES AND SERVICE

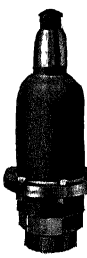
Hoffman Products are sold and stocked by leading wholesalers of heating and plumbing supplies everywhere. Hoffman representatives are available to assist in selection of suitable equipment for various services.

ILLINOIS ENGINEERING COMPANY

General Offices and Factory:
Chicago

Branches and Representatives in Principal Cities

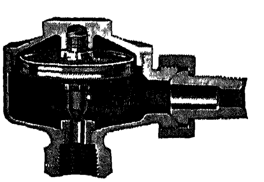
Illinois Selective Pressure Control Systems



Illinois Selective Controller

An entirely new and unique method of Steam Circulation Control . . . Heating Systems that set new standards in comfort, economy, simplicity and convenience of operation. Each system is individually engineered to meet the exact requirements. Recorded fuel savings, without sacrifice of comfort, warrant your investigation. Ask for Bulletin 16.

Illinois Thermo Radiator Traps



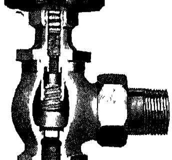
Series G

Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems. Has cone type valve. Flushes thoroughly and seats perfectly at all times. Valve and seat are of Nitralloy. The duplex diaphragm is of special phosphor bronze. Scientific design and rugged construction assure flexibility and long life. These diaphragms have withstood over three million strokes on a breakdown test.

Made in three sizes ½ in., ¾ in. and 1 in. and in a variety of patterns.

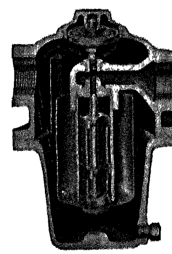
Special thermostatic traps can be furnished for working pressures up to 125 lb.

Illinois Modulating Supply Valve



Quick-opening, packless. Steam tight on 50 lb pressure. Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns.

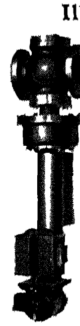
Illinois Combination F & T Traps



Series 7G

Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers—wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold.

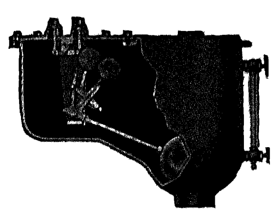
Illinois Flow Control Valves



Type I

In large installations where process steam or zoning requirements do not permit the variable control of combustion, and in housing projects or on Central Station service, Illinois Flow Control Valves are used. They are of the full floating type, giving complete regulation of steam flow. Furnished for manual, pneumatic or electric operation. Ask for Bulletin 517.

Illinois Return Trap



Automatically returns the condensation to the boiler, regardless of pressure on the boiler up to 8 lb, at the same time discharging the air. Insures positive and complete circulation, and prevents cracked boiler sections.

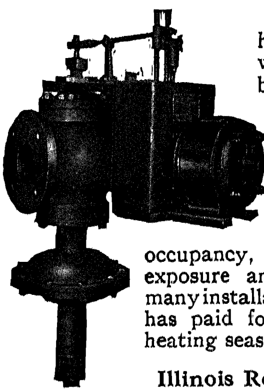
Trap is self-contained, with no external working parts to be misadjusted, tampered with or injured. No stuffing boxes or packed joints, which insures continuous tightness against air or water leakage.

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago

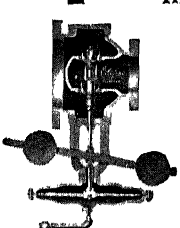
Branches and Representatives in Principal Cities

Illinois Motorized Valves (on and off)



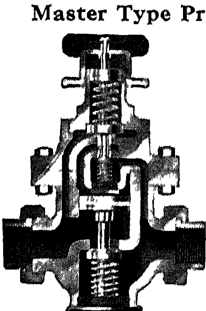
Prevent overheating and fuel waste in large buildings or groups of buildings heated from one central power plant. Buildings may be zoned as to occupancy, time, location, exposure and so on. In many installations this valve has paid for itself in one heating season.

Illinois Reducing Valve



In general use on vacuum or low pressure heating systems. Will reduce to 4 oz pressure from an initial pressure of 150 lb. The large diaphragm insures sensitive operation. Made in both straightway and expanded outlet bodies in sizes from ¾ in. to 12 in.

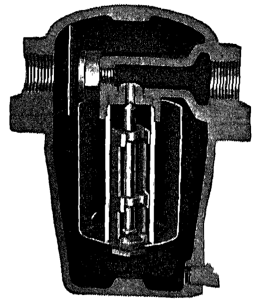
Master Type Pressure Regulator



For exacting requirements, such as tire and rubber vulcanizing, chemical process pressure control, and wherever high pressure steam must be accurately reduced in varying amount to any steady lower pressure not less than 10 lb.

It will reduce initial pressure up to 300 lb down to any lower pressure not less than 10 lb, and does not build up pressure on a closed or dead end line. Made of bronze with Monel metal valves and seats.

Illinois Steam Trap



Series 30

Valve and stem are separate from the bucket and operated only by the bucket at the extreme top and bottom of travel—result—*valve is always either full open or tight closed.* No wire drawing or cutting of valve and seat which are of Monel metal.

Eclipse Spring Controlled Regulating Valve

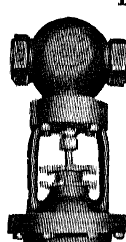
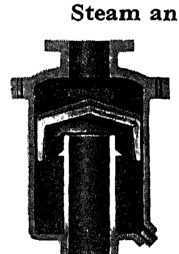


Fig. 121

Furnished in either single seated or double seated type as the service conditions require, for the control of steam, air or gas. Controlling spring is completely enclosed, protecting it from dirt and rust. Valves are furnished with the proper size diaphragm and the proper length spring to give satisfactory service under all operating conditions.

Steam and Oil Separators



Vertical Standard Separators

Eclipse steam separators are made in both horizontal and vertical type, and also the special receiver separators for standard or extra heavy pressures.

Eclipse oil separators are furnished in the horizontal type and have a removable baffle plate to facilitate cleaning of baffle and keeping the separator's efficiency at the highest point.

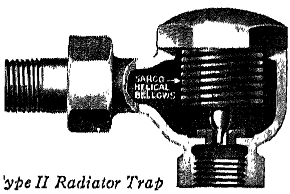
Sarco Company, Inc.

183 Madison Ave., New York, N. Y.

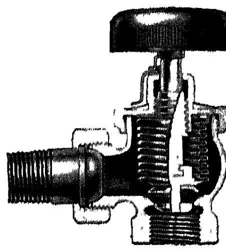
Branches in Principal Cities

SARCO CANADA LIMITED, 85 Richmond St., W., Toronto, Ont.

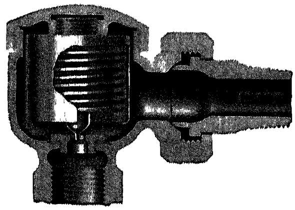
PRODUCTS—A complete line of Specialities for Vapor, Vacuum and Gravity Steam Heating Systems and Control combined with a competent Engineering Service to architects and heating engineers to assist them in providing modern heating.



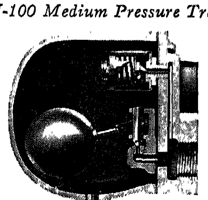
ype II Radiator Trap



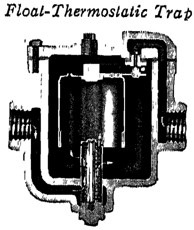
Bellows-Packless Valve



N-100 Medium Pressure Trap



Float-Thermostatic Trap



Inverted Bucket Trap

SARCO RADIATOR TRAPS

Sarco Heating Systems are "prestige Systems." The traps and valves *are* the system as far as maintenance and cost are concerned.

Sarco Type II Traps—Are available in angle, straightway, and corner patterns. The Sarco Thermostatic Bellows—made by special machinery, has not been duplicated or even imitated with success. It works efficiently, repeatedly and persistently. It has worked that way for a quarter of a century. Sizes ½ in. to 1 in. Catalog HV-45.

SARCO RADIATOR VALVES

Sarco Packless Valves Used for one and two pipe heating systems and are truly packless. Steam leaks are impossible. Furnished with round or lever handles or lock shield in angle, straightway, or corner patterns. Sizes ½ in. to 1½ in. Catalog HV-45.

SARCO N-100 TRAP

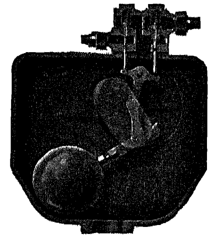
For high pressure radiators and heating coils in stationary and marine service, and for hospital and kitchen equipment. Has full length protecting shield and stainless steel valve head and seat. Sizes ⅜ in. to 1 in. Catalog HV-46.

SARCO FLOAT-THERMOSTATIC TRAPS

For dripping ends of mains and risers, and for stack or blast heaters, large unit heaters and hot water generators. Automatic thermostatic air vents built in. Available in six sizes with connections ¾ in. to 2 in. Catalog HV-38.

SARCO INVERTED BUCKET TRAPS

Are recommended for high pressure unit heaters and sometimes preferred for kitchen and laundry equipment. Strainers are built right into these sturdy traps. Seats and valves are stainless steel and renewable. Thermostatic air vents can be furnished on the larger sizes. Available in sizes ½ in. to 2 in. for pressures up to 900 lb. Catalog HV-165.

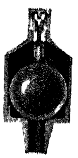

Alternating Receiver

SARCO ALTERNATING RECEIVER

A complete line of boiler return traps for vapor systems.

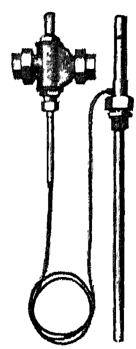
Returns water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions.

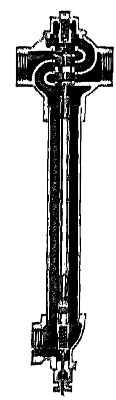
Made in six sizes for from 1,500 to 25,000 sq ft of radiation. Catalog HV-45.



SARCO AIR ELIMINATORS

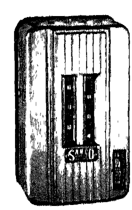
For venting air from vapor systems at one central point in the basement. Available in two sizes: No. 6 for systems up to 3,000 sq ft and No. 12-A for 15,000 sq ft. Both are equipped with float valves to stop water escaping through the vent and with check valves to prevent ingress of air when system is under vacuum. Catalog HV-45.

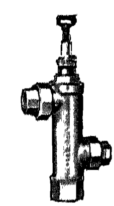

Type TR-21 Standard for hot water storage tanks, fan units, etc.


Water Blender Type MB

SARCO SELF-CONTAINED TEMPERATURE REGULATORS

Sarco Temperature Regulators are simple, self-operated valves—the only self-contained units that use the irresistible force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required; all moving parts are *inside* the equipment. Here again—a type and size for every purpose—for steam, gas, oil, water or brine for temperatures ranging from 0 to 400° F. Catalog HV-52.

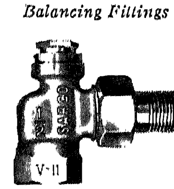

Room Thermostat


Water Blender Type DB

SARCO WATER BLENDERS AND TEMPERING VALVES

For mixing hot and cold water to deliver automatically water at any desired temperature. Two models are available, type MB for showers, wash basins, etc., and type DB, a tempering valve for use with submerged heating coils or tankless heaters. Catalog HV-140.

Balancing Fittings



SARCOFLOW BALANCING FITTINGS

For balancing radiators in hot water heating systems, to secure uniform temperatures in all rooms.

Sarcoflows are placed in the radiator outlets instead of ordinary union ells and can be adjusted to throttle the water flow thru each radiator as needed.

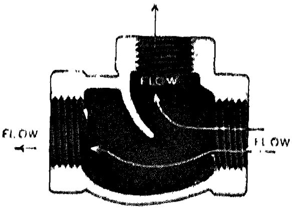
Available in three patterns with either screwed or sweat type connections. Catalog HV-221.

SARCO ELECTRIC CONTROLS

Comprise room thermostats, aquastats, and limit controls for all heating and air conditioning needs; also motor valves for steam, water, brine, or freon.

Sarco has available direct-by-the-weather control systems for steam and hot water heated buildings. Catalogs HV-209 and HV-210.

TRIPLEX DISTRIBUTOR and AIR ELIMINATOR FITTINGS



CUT OPEN VIEW OF TRIPLEX DISTRIBUTOR

COPPER PIPE DISTRIBUTORS

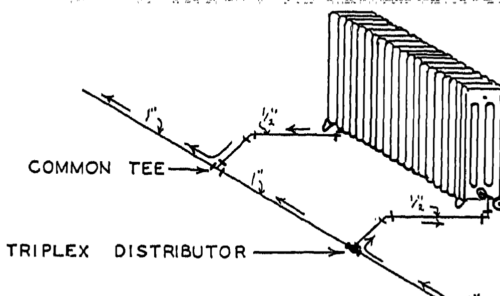
Main Pipe Size	Run	Outlet	Ship. Wt.
3/4"	3/4" x 3/4"	1/2"	1 lb.
1"	1" x 1"	1/2" or 3/4"	1 1/2 lbs.
1 1/4"	1 1/4" x 1 1/4"	1/2" or 3/4"	2 lbs.
1 1/2"	1 1/2" x 1 1/2"	1/2" or 3/4"	3 lbs.

IRON PIPE DISTRIBUTORS

Size	Tapping	No. per Box	Wt. per Box
3/4"	3/4" x 3/4" x 1/2"	6	9
1"	1" x 1" x 1/2" -1" x 1" x 3/4"	6	11
1 1/4"	1 1/4" x 1 1/4" x 1/2" -3/4" -1"	6	13
1 1/2"	1 1/2" x 1 1/2" x 1/2" -3/4" -1"	6	18
2"	2" x 2" x 1/2" -3/4" -1"	6	24

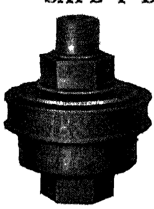
TRIPLEX DISTRIBUTOR

Circulating fittings for one pipe systems. Efficiency comparable to a two pipe system. Only one fitting per radiator, except where basement radiators are installed which require two Distributors. By closing a radiator valve the full area of the main is bypassed through the fitting to the balance of system.



DETAIL OF DISTRIBUTOR INSTALLATION

SAFE-T BOILER PLUG

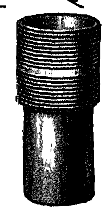

No. 36

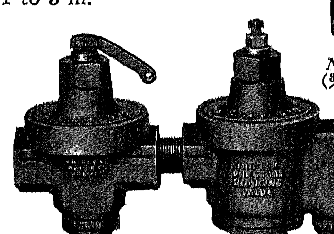
Positive relief from dangerous pressures. Included with Flow Control Systems (complete), Tank-in-Basement Systems, No. 7 and 8 Control Units. Breakages of boiler from excessive pressure eliminated and protection assured.

No.	Break. Pres.	Breaking Bars	Diaphragms	Dimensions Wd.	Ht.	Dia.	Ship. Wt.
36	40 lbs.	Wht. Iron	Lead	3 1/2	4 1/2	3 1/2	3 lbs.
37	75 lbs.	Wht. Iron	Lead	3 1/2	4 1/2	3 1/2	3 lbs.
38	100 lbs.	Wht. Iron	Lead	3 1/2	4 1/2	3 1/2	3 lbs.
39	125 lbs.	Wht. Iron	Lead	3 1/2	4 1/2	3 1/2	3 lbs.

AIR ELIMINATOR

A positive method of permanently eliminating air from concealed copper or cast iron radiator after the system is vented the first time. Sizes 1 to 3 in.




No. 15 (3/4 in.)

TRIPLEX CONTROL UNITS

Easily installed. All water strained as it enters system. Strainer cast integral with Pressure Reducing Valve. Proper amount of water maintained in the system and the city water pressure reduced with the Pressure Reducing Valve. The **fast-filling feature will save the cost of the unit in time saved filling the system.** The Relief Differential Type Valve, seating under water and has lever for testing and flushing. Check is between Relief and Pressure Reducing Valves.

No. 7 CONTROL UNIT is a positive protection against boiler breakages from pressure. It consists of a No. 15 unit and No. 36 (40 lb) Safe-T Boiler Plug.

No.	Tapped	Body	Diaphragms	Springs	Dimensions Length	Ht.	Dia.	Ship. Wt.
15	3/4"	R. P. Iron	Bronze	Bz. & R. P. Stl.	10 3/8"	6 3/4"	3 1/4"	9 1/4 lbs.
7	3/4"	R. P. Iron	Bronze	Bz. & R. P. Stl.	10 3/8"	6 3/4"	3 1/4"	12 1/4 lbs.
14	1/2"	R. P. Iron	Bronze	Bz. & R. P. Stl.	9 3/4"	6 3/8"	3 1/4"	7 3/4 lbs.
14B	1/2"	R. Brass	Bronze	Bz. & R. P. Stl.	9 3/4"	6 3/8"	3 1/4"	7 lbs.
8	1/2"	R. P. Iron	Bronze	Bz. & R. P. Stl.	9 3/4"	6 3/8"	3 1/4"	10 3/4 lbs.
13	1/2"	R. P. Iron	Bronze	Bz. & R. P. Stl.	9 3/4"	6"	3 1/4"	7 1/2 lbs.

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

-since 1888 Webster Systems of Steam Heating

Main Office and Factory:
Camden, New Jersey

Representatives in over 60 cities—
Consult Your Local Phone Directory

Webster Nesbitt UNIT HEATERS

PRODUCTS AND SERVICES

Webster Systems of Steam Heating including Vacuum and Type "R" (vapor).

Webster Central Control Systems including HYLO and MODERATOR.

Modernization of Obsolete and Faulty Heating Systems.

Webster System Equipment including Light-Weight Concealed Radiation (Gravity Convection Heaters), Radiator Supply Valves, Metering Orifices, Thermostatic Traps, Drip Traps, Heavy Duty Traps, Dirt Strainers, Dirt Pockets, Boiler Return Traps, Vent Traps, Damper Regulators, Boiler Protectors, Lift Fittings, Expansion Joints, Separating Tanks, Steam and Oil Separators, Steam Vacuum Pump Governors, Air Separating and Receiving Tanks, Gages, Water Accumulators.

Webster Series "78" and Series "79" Traps for use at process pressures (10 to 150 lb per sq in.)

Webster-Nesbitt Unit Heaters and Residential Conditioners.

WEBSTER SYSTEMS

Webster Systems are low pressure, two-pipe systems of steam circulation *with the addition* of accurately-sized metering orifices at radiator supply connections and, when required, intermediate metering orifices at points in branch mains. Metering orifices effect even distribution of steam to all parts of the heating system and permit

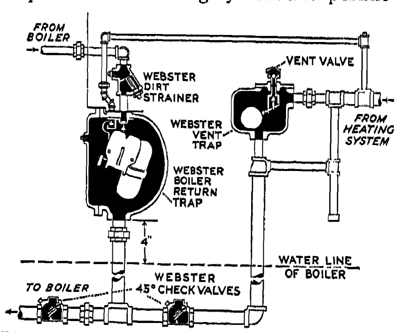


Fig. 1. Conventional arrangement of piping around Webster Basement Equipment for the Webster Type "R" System

the successful application of a centralized control. Webster Valves are used at supply of radiators. Webster Thermostatic Traps prevent flow of steam into return mains when radiators are filled. Webster Drip Traps and Dirt Strainers are used where needed on steam mains. Webster Systems are available for vacuum, open return or "vapor" operation. The Type "R" System corresponds to the so-called Vapor type. Fig. 1 illustrates a typical arrangement of Boiler Return Trap, Vent Trap, etc., when low pressure boiler is the source of steam.

WEBSTER CENTRAL CONTROLS

These are patented systems for varying the amount of steam to all radiators according to outside temperature. They provide *continuous heat delivery* with effective *fractional filling of radiators.* The Hylo Systems may be provided for manual control, or if desired, may be semi-automatic by incorporation of inside thermostat or thermostat and schedule clock. The Moderator Systems employ an automatic Outdoor Thermostat supplemented by a manual Variator.

The latter is used for quick heating-up, night load, and unusual weather or occupancy conditions. Use of Webster Central Control Systems results in (1) increased comfort because over-heating and underheating are minimized and (2) lower fuel or steam costs.

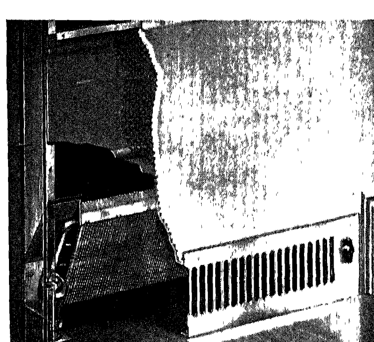


Fig 2. Webster System Radiation

WEBSTER SYSTEM RADIATION

Concealed, non-ferrous type for use exclusively with Improved Webster Systems. Is unique in that it combines *in a single unit,* a light-weight heating element of high efficiency with an orificed radiator supply valve, a radiator trap and supply and return piping connections. Metal enclosures for installation within the wall and exposed metal cabinets are available. Webster System Radiation and enclosures are so designed that the entire heating element can be quickly removed without damage to plaster or paint. Space requirements reduced to a minimum and installation greatly simplified.

RADIATOR SUPPLY VALVES

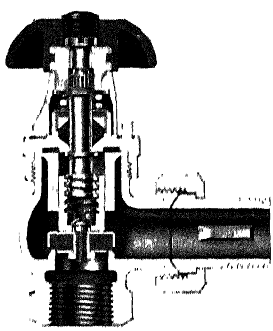


Fig. 3. Webster Type "WB" Valve

The new Webster Series 600 and Series 600S are supply valves of highest quality designed to eliminate the many sources of annoyances caused by valves of inferior design and quality. Webster Supply Valves open quickly and easily in less than a turn of the handle. They have non-rising stems. Steam can actually be shut off from radiators because they seat positively.

Type "WB" Valve (Series 600)—Uses a molded ring packing but may almost be called "permanently packed" as the packing seldom requires renewing. A feature is the spring-retained, metal-to-metal seal giving extra protection against leakage. Standard models have screwed packing gland. Modulation sleeve furnished on special order at slight added cost. This type entirely suitable for hot water heating service; furnished with or without leak hole as desired.

Sylphon Packless Valve (Series 600S)—Same features as Type "WB" except for genuine seamless Sylphon Bellows to completely encase valve stem. Meets fully "bellows packless" specifications. Modulation sleeve is standard equipment for ½, ¾, and 1 inch sizes. Not suitable for hot water heating service.

Bodies and Handles—Angle Body is made in ½, ¾, 1, 1¼ and 1½ inch sizes; right corner, left corner, straightway (both single and double union) bodies in ¾ and 1 inch sizes.

The ¾ in. size is available with ½ in. spud also. Choice of wheel, lever, lockshield, chain wheel, and extended stem handles.

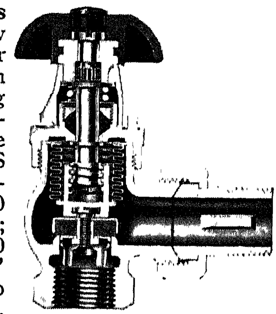


Fig. 4. Webster Sylphon Packless Valve

Pressures—For low pressure vapor and vacuum steam heating service. Maximum pressure for Series 600S Sylphon Packless Valve, 20 lb per sq in.; for Series 600 Type "WB" Valve, 75 lb per sq in. Other Webster Valves are available on order for higher maximum pressures.

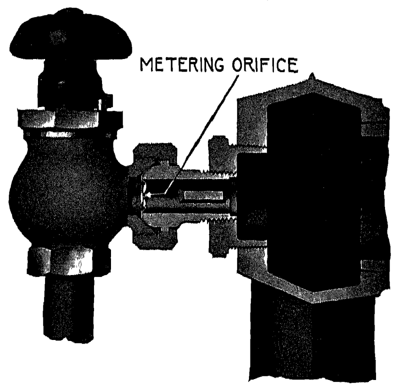


Fig. 5. Metering Orifice Inserted in Union Connection of a Webster Supply Valve. Other types are available.

Metering Orifices—Accurately sized and made of heavy gage Monel Metal to resist erosion and corrosion, amply thick to be free from vibration and shaped for quiet operation.

RETURN TRAPS

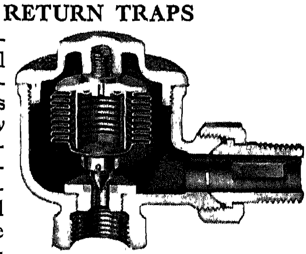


Fig. 6. Webster 502 Sylphon Trap

Sylphon—Perfected thermostatic bellows trap, *fully compensated for pressure.* Stainless steel valve piece and renewable seat. Factory adjusted. Made in angle, right-corner, left-corner, vertical and straightway bodies. Sizes: ½, ¾ and 1 in. For low pressure

vapor and vacuum steam heating service. Maximum pressure 25 lb per sq in.

Series "7M"—Perfected diaphragm-type thermostatic trap, *fully compensated for pressure.* Uses Monel Metal diaphragm, Stainless Steel valve piece and seat insert. Renewable seat. Factory adjusted. Made in angle, right-corner, left-corner, vertical and straightway bodies. Sizes: ½, ¾ and 1 in. For low pressure vapor and vacuum steam heating service. Maximum pressure 25 lb per sq in.

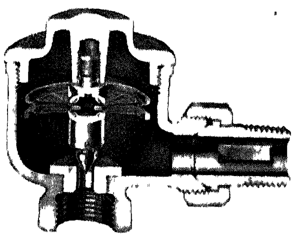


Fig. 7. Webster Size 702-M Trap

Series 7 with phosphor-bronze diaphragm, brass valve piece and seat is also available.

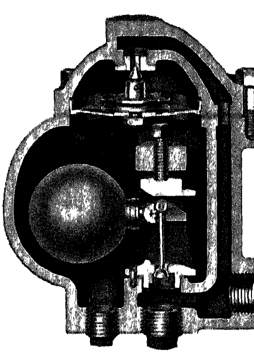


Fig. 8. The Webster Size 0026-T Drip Trap is Rated 700 lb Water per Hour at 2 lb Pressure Difference

Series "26"—A heavy duty trap for drips of mains, blast radiation, unit heaters, hot water generators and similar applications. A rugged float-type trap available with and without thermostatic air vent. Made in six sizes: 200, 700, 1200, 2400, 5000 and 11,700 lb water per hour at 2 lb pressure difference. Maximum working pressure is 15 lb per sq in.

Series "78"—thermostatic trap *built for* process steam pressures (10 to 150 lb per sq in.). Monel Metal diaphragm. Stainless Steel valve piece and seat insert. Angle model only. Sizes: ⅜, ½, ¾ and 1 in. Extensively used with laundry, cooking, sterilizing and other process-steam uses.

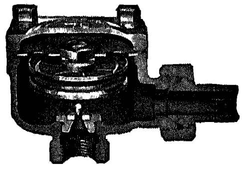


Fig. 9. Webster Size 782 Trap

Series "79"—For use where large volumes of very hot condensate form more quickly than can be discharged by thermostatic traps alone. Float and thermostatic traps designed for normal working pressures between 15 and 150 lb per sq in. Water of condensation is passed through a float-controlled seat opening while air is discharged into the return piping by a thermostatically controlled vent. Compact and light in weight. Can be readily mounted in a pipe line without other support. Available with either ¾ in. or 1 in. inlet and outlet.

Cast iron body, copper asbestos gasket and cover bolted together with steel cap screws. Monel Metal valve piece and stem. Stainless steel seat. Air vent unit is Monel Metal diaphragm with Stainless Steel valve piece and brass seat with Stainless Steel insert.

DIRT STRAINERS AND POCKETS

Placed in return lines of steam heating systems to prevent dirt, rust and scale from impairing tightness of traps.

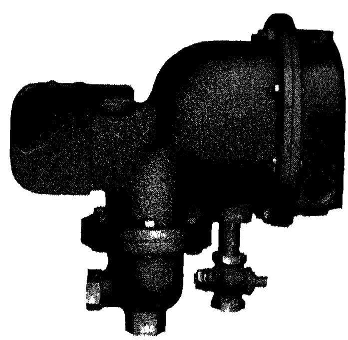


Fig. 10. Size 34C-1 Webster Boiler Protector with Low Water Electrical Cut-out Switch. Size 34 has no Cut-out Switch

BOILER PROTECTOR

Prevents breakage in low pressure heating boilers when water level becomes inadequate. Automatically supplies raw water to boiler when water level drops to 1 in. above bottom of gage glass.

For maximum boiler pressure of 15 lb per sq in. Maximum cold water main pressure should not exceed 150 lb per sq in.; minimum must not be less than 25 lb per sq in.

Made with ¾ in. connections, with or without electrical cut-out switch.

WEBSTER-NESBITT UNIT HEATERS

Are manufactured by John J. Nesbitt, Inc., Holmesburg, Philadelphia, Pa., and are distributed solely through Warren Webster & Company, Camden, New Jersey. Designed to circulate large volumes of air at comparatively low temperatures, assuring quick heating.

Ratings of Webster-Nesbitt Unit Heaters are based on tests made in accordance with standard test code of Industrial Unit Heater Association and A.S.H.V.E.

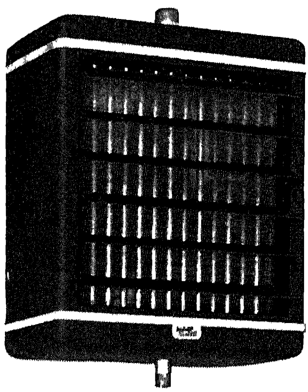


Fig. 11. Standard Propeller-Fan Type

PROPELLER-FAN UNIT HEATERS

Newly designed twenty-one unit models now available giving a wide selective range. Quiet large area blade fans. Rubber-isolated motors; single or multispeed. Compact suspended type. Sturdy casings. Modern styling. Catalog W-N 111.

GIANT UNIT HEATERS

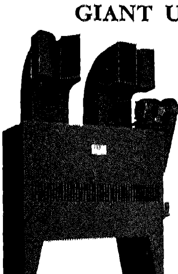


Fig. 12 Blower-Fan Type

Blower-fan type for economical heating of large areas. Floor-mounted, wall-mounted, ceiling-suspended, from 109,000 Btu, 2620 cfm to 1,008,000 Btu, 16,000 cfm with 2 lb steam, 60 degrees entering air. With or without Thermadjust Temperature Control Damper. Catalog W-N 104.

SERIES F UNIT HEATERS

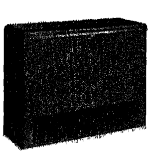


Fig. 13 Series F Unit Heater

Centrifugal fan type for lobbies, showrooms, offices where quietness and appearance count. Four casing sizes with two radiator sizes available for each casing. Floor or ceiling mounting. Publication W-N 105.

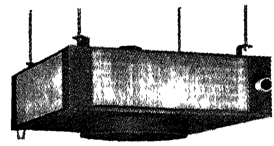


Fig. 14 "Little Giant" Down Blow Type

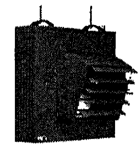


Fig. 15. Horizontal Blow Type

LITTLE GIANT UNIT HEATERS

New, light, compact, draw - through, high-velocity units available in down blow and horizontal blow models. 39,200 Btu, 710 cfm to 625,000 Btu, 9150 cfm.

Down Blow Type—In general, indicated when the presence of cranes and other machinery requires that the unit and piping be located well above the floor level.

Horizontal Blow Type—Application follows principles of heat distribution regularly employed in suspended blower fan type heater. Units are located closer to working zone than in the case of the Down Blow type. See Publication W-N 109.

RESIDENTIAL CONDITIONERS

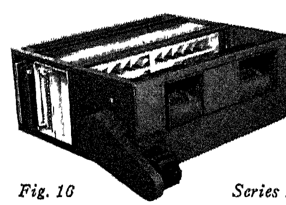


Fig. 16 Series R

Series R—For large and small residences. In two-section combinations for winter heating only with steam or hot water systems; or for heating, air cleaning and humidification (with a trouble-free cascade-type of humidifier) for summer cooling and dehumidification with cold water. Eight basic sizes: 750 to 4000 cfm.

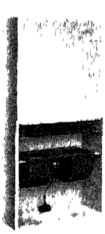


Fig. 17 Series D

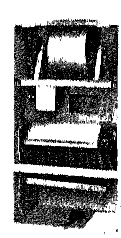


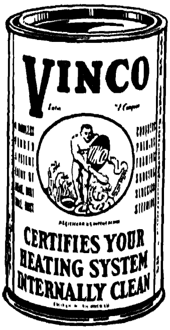
Fig. 18 Series A

Series D—Simplified compact unit heating systems for apartments and multiple residences. Available with steam or hot water heating element. Three sizes: 300, 450, 600 cfm.

Series A—For apartments. All the newly designed features of Series D Units plus (in a little added space) air cleaning; winter humidification, and summer cooling with cold water. Four sizes: 350 to 1000 cfm. Send for Publication W-N 107.

The Vinco Company, Inc.

305 East 45th Street — **New York, N. Y.**



Boiler Cleanser
3, 5 and 10 lb cans

VINCO BOILER CLEANSER

A positively harmless insoluble powder cleaner for new, remodeled and old heating systems. A unique, scientifically processed compound on a special formula not to be confused with other powder boiler cleaners.

What Vinco Boiler Cleanser Does

VINCO removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water *without the labor of blowing boilers over the top.*

By this thorough cleaning Vinco stops foaming, priming, surging, and slow steaming.

How Vinco Boiler Cleanser Works

Each minute grain of VINCO powder adsorbs several times its own weight of oil, rust and dirt. These larger grains of adsorbed impurities then settle and are drained through the bottom according to directions on each can.

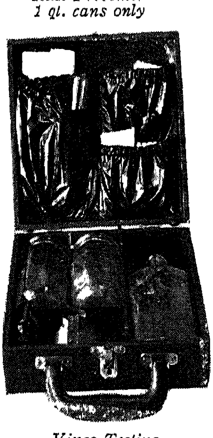
Our Guarantees

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients.
2. Purchase price is refunded if results are not as claimed when VINCO has been used according to directions.

Rust Preventer
1 qt. cans only

VINCO RUST PREVENTER

When used after VINCO Boiler Cleanser has removed oil, grease, rust, scale and dirt, it will keep the rust inhibiting alkalinity at the optimal constant for a year or more. (Testing kit below has complete instructions and chart.)



Vinco Testing
Kit No. 10
(Patent applied for)

VINCO TESTING KIT No. 10

for Testing Heating Boiler Waters

The kit enables the layman to make simple, rapid tests to diagnose and prescribe correct treatment of boiler waters right on the job.

A new time saving method that permits valid conclusions heretofore requiring complicated and often lengthy laboratory analysis and technique.

Each kit has sufficient material for about 100 tests.

Refills cost about 2 cents per test.

Soot-Off—1 lb cans only

VINCO SOOT-OFF

Non-explosive, thoroughly safe. No black smoke—no fire hazards. Destroys soot from coal, oil or gas burning heating equipment. Easier, cleaner, quicker, and more thorough than brushing. Cleans fire pot, flues and chimney in one simple operation.

Liquid Boiler Seal
1 qt. cans only

VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid seal. Unique in that it does not induce priming and foaming. It has no unpleasant smell. Makes speedy and permanent repairs of boiler and heating system leaks. Fine to tighten up new jobs. Directions simple.

Quantities

Steam and Vapor Systems—Use 1 quart VINCO Liquid Boiler Seal to each 6 sq. ft. *grate* area.

Hot Water Systems—Use 2 quarts VINCO Liquid Boiler Seal to each 6 sq. ft. *grate* area.

SPECIFICATIONS FOR COMPLETE VINCO TREATMENT

of New and Remodeled Steam or Vapor Heating Systems

CAUTION

Do not use as a cleansing agent soda or any alkali, vinegar or any acid in this boiler. USE VINCO.

DIRECTIONS FOR CLEANING

After the system is tested and tight, use the proper quantity of VINCO listed on the reverse side of this tag and follow directions given on each can of VINCO. Satisfactory results will be obtained in record time with minimum expense.

QUANTITIES OF VINCO (IN POUNDS) REQUIRED FOR HEATING SYSTEMS

(Note that quantities are based on actual installed radiation, not on boiler capacity.)

	For Steam or Vapor Systems to prevent or cure priming or foaming. Also for Hot Water Heating Systems maintained at [illegible] or above	Annually to remove rust scale [illegible] and for Hot Water Systems below [illegible]
up to 350 sq. ft. of radiation	3	1½
351 " 600 " " " "	5	2½
601 " 1100 " " " "	8	4
1101 " 1400 " " " "	10	5
1401 " 1000 " " " "	13	6½
1801 " 2100 " " " "	15	7½
2101 " 2700 " " " "	18	9
2701 " 3100 " " " "	20	10
3101 " 3700 " " " "	23	11½
3701 " 4200 " " " "	26	13
4201 " 4600 " " " "	28	14
4601 " 5000 " " " "	30	15
5001 " 5300 " " " "	31	15½
5301 " 5600 " " " "	32	16
5601 " 5900 " " " "	33	16½
5901 " 6200 " " " "	34	17
6201 " 6500 " " " "	35	17½
6501 " 6800 " " " "	36	18
6801 " 7100 " " " "	37	18½
7101 " 7400 " " " "	38	19
7401 " 7700 " " " "	39	19½
7701 " 8000 " " " "	40	20
8001 " 8300 " " " "	41	20½
8301 " 8600 " " " "	42	21
8601 " 8900 " " " "	43	21½
8901 " 9200 " " " "	44	22
9201 " 9500 " " " "	45	22½
9501 " 9800 " " " "	46	23
9801 " 10100* " " "	47	23½

*Above 10100 sq. ft. use an additional pound Vinco for each additional 300 sq. ft. of actual installed radiation.

1. These tags used by the leading boiler manufacturers on their boilers give proper directions for cleaning boilers of oil, grease, rust, scale and dirt.
2. After using VINCO Boiler Cleanser, VINCO Testing Kit No. 10 and accompanying chart should be used to determine and apply the proper quantity of VINCO Rust Preventer.

NOTE: VINCO Rust Preventer should be applied annually, but VINCO Boiler Cleanser need only be used when radiation has been added or piping changes have been made or a new boiler installed.

3. Use VINCO Soot-Off several times a year.

HOT WATER SYSTEMS

1. Use VINCO Boiler Cleanser in the quantities listed on above tags.
2. Apply VINCO Rust Preventer in the same quantities as for steam or vapor systems.
3. Use VINCO Soot-Off several times a year.

McDONNELL & MILLER

Manufacturers of McDONNELL *Boiler Water Level* CONTROLS

General Offices: Wrigley Building, **Chicago, Ill.**

"Doing one thing well"

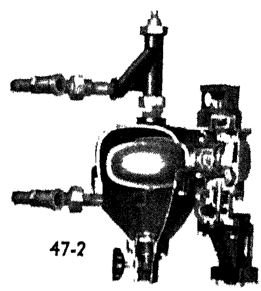
PRODUCTS:

Boiler Water Feeders; Feeder-Cut-Off Combinations; Low Water Cut-Offs; Pump Controls and Low Water Alarms for Steam Boilers. Humidifier Water Level Controls for Warm Air Furnaces. Safety Valves for Hot Water Heating Boilers. High and Low Oil Switches and related equipment.

Boiler Water Feeders — McDonnell boiler feeders protect steam boilers from low water by automatically supplying water to the boiler when it is needed and in exact proportion to the need. They are available in models to suit every make and size of steam boiler with every type of firing and operating at any pressure up to 75 lb. They are divided, broadly, into two classes: (1) Feeders for boilers below 5000 sq ft capacity—largely taken care of by No. 47 or 147; (2) Feeders for boilers above 5000 sq ft, largely taken care of by the No. 51 or 53.

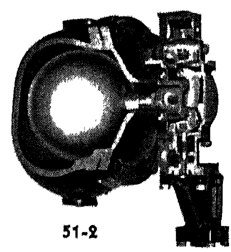
Feeder Cut-Off Combinations—For automatically fired boilers The McDonnell No. 2 Low Water Cut-off Switch is added to form a feeder-cut-off combination, like the 47-2, No. 51-2, etc. In such a combination the feeder takes care of the normal water requirements. In case of an emergency, however, such as excessive priming and foaming, the low water cut-off switch stops the burner. When the water level has been restored to ¾ in. in the water glass, the circuit is re-established so the burner can start again. The same feeder-cut-off effect can be obtained by using the No. 101 Electric Water Feeder with either the No. 67 Low Water Cut-off or with McDonnell Built-in cut-offs. The latter arrangement is the only practical means of making a neat, unobtrusive feeder-cut-off installation on a dressed-up jacketed boiler with recessed water glass.

Low Water Cut-Offs—If the feeder-cut-off combination is not desired, the No. 67 alone can be installed to dependably stop the burner when low water threatens. For high pressure jobs the No. 150 will serve not only as a low water cut-off but as a boiler feeder and low water alarm as well—for pressures as high as 150 lbs.



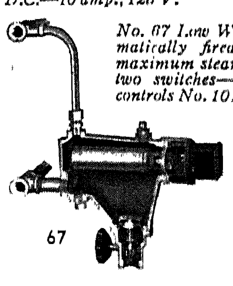
47-2

No. 47-2 Feeder-Cut-Off Combination is for automatically fired boilers below 5000 sq ft capacity—maximum steam pressure 25 lbs. No. 47 is for hand fired boiler—same service range, but without No. 2 Switch. For process boilers below 5000 sq ft capacity or for boilers of this size with pressure up to 35 lbs, use the No. 147.



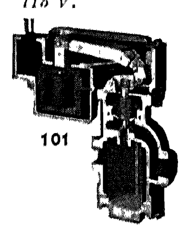
51-2

No. 51-2 Feeder-Cut-Off Combination is for automatically fired boilers above 5000 sq ft, maximum steam pressure, 35 lbs. Use No. 51 (without No. 2 Cut-off switch) for hand fired boilers. For pressures from 35 to 75 lbs use the No. 53. Electrical ratings of No. 2 Cut-off Switch: A.C.—¾ Hp., 110-220 V.; D.C.—10 amp., 125 V.



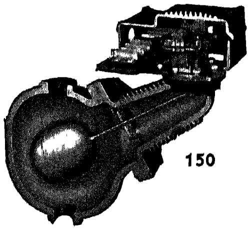
67

No. 67 Low Water Cut-Off is for automatically fired boilers of any size; maximum steam pressure, 25 lbs. Has two switches—one operates alarm or controls No. 101 Electric Feeder (below), other acts as low water cut-off. Rating (each switch): A.C.—¾ Hp., 115-230 V., D.C.—¼ Hp., 115 V.



101

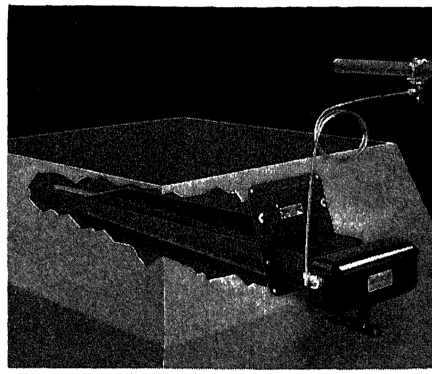
No. 101 Electric Water Feeder for use with No. 67 or Built-in Cut-offs (see next page) to provide Feeder-Cut-off Combinations.



150

No. 150 Combination Pump Control Low Water Cut-off and Alarm, for steam pressures up to 150 lbs. Has two switches; one controls pump—other stops burner and completes alarm circuit when water level falls to danger zone.

Advanced features—A notable advance in the No. 47 and No. 67 is the deep sediment chamber with McDonnell-built, self-closing, large-capacity, straight-through blow-off valve which complies with the A.S.M.E. Boiler Code. This permits draining the float chamber faster than water can flow into it so that the insurance inspector or owner can check operation. Other features of McDonnell feeders and cut-offs include: patented Quick-Hook-Up, patented cool feed valves, finer stainless steel valves, large area (14 sq in.) built-in strainers, and packless (sylphon) construction.



Phantom view showing how the McDonnell No. 217 Humidifier valve is installed on the humidifying pans of warm air furnaces. Its snap action prevents the clogging and caking that soon stops the ordinary humidifying valve.

Built-in Low Water Cut-Offs

McDonnell "Built-in" Low Water Cut-offs have been chosen as standard equipment on modern jacketed boilers and boiler manufacturers have provided a tapping in which they can be quickly and neatly installed. A recent development is the self-cleaning feature which insures dependable operation even when mud and silt are present in the boiler water. McDonnell "Built-ins" should be specified *with* the boiler to insure the proper type for a particular make.

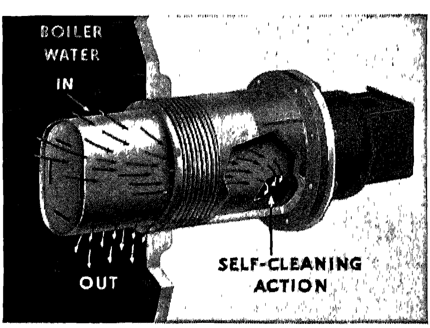
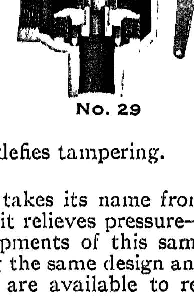


Illustration shows how special float shield design directs boiler water circulation through the shield, constantly flushing space around float and bellows.

McDonnell No. 217 Humidifier Water Control for automatically maintaining proper water level in evaporation pans of warm air furnaces—A new snap action eliminates the tendency of former controls to become stopped up by foreign matter or to stick and become inoperative due to the presence of lime and scale. This valve has only two positions —tight closed and *wide* open. When water falls ¼ in. in pan it *snaps* wide open feeding a full stream that flushes away all foreign matter, lime and scale. It stays wide open until level is restored, then *snaps* to a tight leak-proof closure against any water pressure up to 150 lbs.

The McDonnell No. 217 Humidifier Water Control can be adapted to fit all furnaces. Available as float and valve unit only; with sturdy float chamber; with fittings and tubing for tapping into water supply pipe; or as complete humidifying unit with evaporating pans of various sizes.

McDonnell No. 29 Safety Valve for Hot Water Heating Boilers—Tests (available to you) prove that this is the first safety valve that will positively prevent pressure rising above its set relief point of 29 lbs. It does not simply crack open when the relief point is reached as former types do; it snaps *wide open* and quickly relieves the pressure; then snaps to a leak-proof closure. The No. 29 complies with the A.S.M.E. code. It is factory set and defies tampering.



No. 29

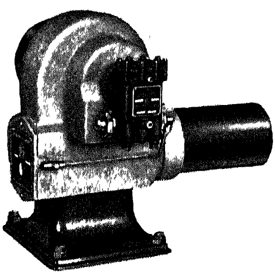
While the No. 29 takes its name from the setting at which it relieves pressure—29 lbs—other developments of this same safety valve, utilizing the same design and operating principles, are available to relieve pressure at various higher settings.

AMERICAN RADIATOR & Standard Sanitary

New York CORPORATION Pittsburgh

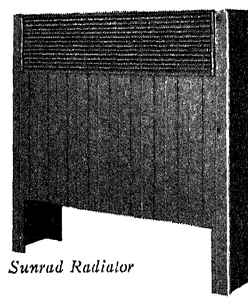
YOU CAN FILL EVERY HEATING NEED FROM THIS COMPLETE LINE

American Heating Equipment includes Boilers of all types, in all sizes for Coal—stoker or hand-fired, Oil or Gas--and a full line of Radiators, Convectors, Oil Burners, Domestic Water Heaters and Accessories, all backed by the undivided responsibility of the world's largest Heating and Plumbing organization.



ARCOFLAME OIL BURNERS

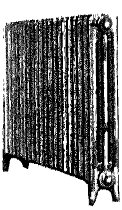
The Model "C" Arcoflame has a capacity of up to 3 gallons per hour. The Model "L" (not shown) from 3 to 7 gallons per hour. Both embody unusual and highly efficient features.



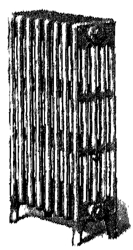
Sunrad Radiator

ARCO RADIATOR

Slim, space-saving and highly efficient, the Arco Radiator comes in three narrow widths and in four heights.



Arco Radiator



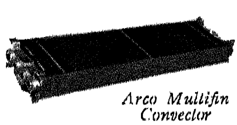
Corto Radiator

CORTO RADIATOR

The Corto is the original thin tube radiator. It is available in seven heights and five widths.

SUNRAD RADIATOR

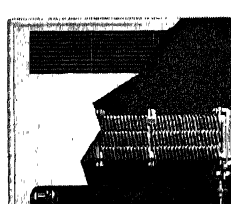
Streamlined and modern, the Sunrad is one integral unit and supplies both radiant and convected heat.



Arco Multifin Convector

ARCO MULTIFIN CONVECTOR

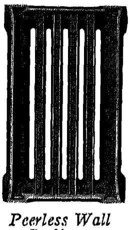
Non-ferrous. Highly efficient. For all systems except one-pipe steam. Available in five widths.



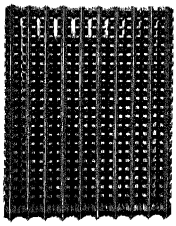
Arco Convector

ARCO CONVECTOR

For concealed radiation at its best. Available in four widths and in virtually any desired length.

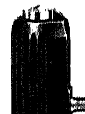


Peerless Wall Radiator

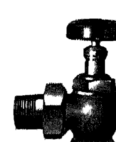


Vento (for fan and blower work)

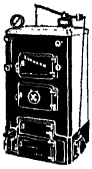
No. 861 Arco Detroit Hurivent Valve (for main)



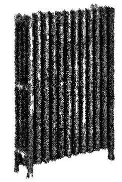
No. 900 Arco-Detroit Multi-port Valve (for radiators)



No. 999 Arco Packless Steam Radiator Valve



Burnham Boiler Corporation



Irvington-on-Hudson, New York

Zanesville, Ohio

There's a Burnham for Every Purpose

Catalogs Sent on Request

1—**Welded Steel Boilers Also Three Purpose Welded Steel Boilers.**
For heating, hot water supply and incineration. Coal or oil. Completely welded for 15 lb working pressure. Multiple shaking grates. Sizes for commercial or domestic uses. Special folder sent on request.

2—**Water Tube Boilers for Steam and Hot Water Heating.**
17, 21, 27, and 36 in. Double shaking grates, long fire travel. Steam rating to 4,920 sq ft; for water, 7,880 sq ft.

3—**Water Tube Boilers Jacketed in Color.**
17, 21, and 27 in. Steel Jacket and 4 ply air cell asbestos insulation. Enameled rich red and black. Jacket goes on after all other set up work. Rating to 2,500 sq ft for steam and 4,000 sq ft for water.

4—**Burnham Oil-Burning Boilers.**
A specific-sized boiler for each specific heat job for use with any standard oil burner. Round Sectional Burnhams in 4 series and 8 sizes. Square Burnhams in 5 series and 41 sizes. For steam, vapor, or water.

5—**Big Twin Sectional Boilers.**
50 in. Grate divided for easy shaking. Twin sections, divided down the middle. Ratings to 14,600 sq ft for steam, 23,360 sq ft for water.

6—**Tube Type Smokeless Boilers.**
Burn soft coal efficiently, without smoke. Meet smoke ordinances everywhere. Similar to (2) above with addition of smokeless feature.

7—**Round Sectional Boilers.**
This boiler made the long fire travel famous. Corrugated crown sheet. Very large steam dome. Ratings up to 830 sq ft for steam, 1,330 for water.

8—**High Pressure Hot Water Supply Boilers.**
Sectional construction. Guaranteed to 120 lb working pressure. Supplies up to 6,400 gallons.

9—**Dome Top Hot Water Supply Boilers.**
Will keep 50 to 1800 gal tank always full of hot water. Guaranteed to 120 lb working pressure.

10—**Burnham-Taco Tanks.**
Combining water heater and storage tank in one unit for summer-winter use. Removable copper heating element. Tanks may be galvanized, Everdur or copper.

11—**Burnham Slenderized Radiators.**
Cast iron radiators that occupy 40 per cent less space than ordinary tube type of same rating. Shorter. Lower. Narrower. 3-tube type $3\frac{1}{4}$ in. wide. 4-tube type $4\frac{7}{16}$ in. wide. 5-tube type $5\frac{11}{16}$ in. wide. 6-tube type $6\frac{15}{16}$ in. wide. Can be recessed.

12—**Burnham Radiant Radiators.**
A self-contained cabinet type of combined radiator and grille.

13—**Burnham Air Conditioning Units.**
Do double duty of both heating and winter air conditioning. Units placed in the room. Have no basement equipment. Take up no more room space than usual grille-enclosed radiator. Entirely automatically controlled.

14—**Burnham Air and Vacuum Valves.**
Full line for radiators, risers and mains.

15—**Burnham Radiator Valves.**
Complete line of heating accessories. Including steel tanks of all kinds.

16—**Burnham Flexible Headers.**

17—**Burnham Unit Heaters.**
Complete line in modern designs.

18—**Burnham Fan Cooler.**
Portable unit for offices, hotels, restaurants and home rooms. Also complete equipment package for attic cooling of residences.

Burnham Boiler Corporation

Manufacturers of Cast Iron and Welded Steel Boilers, Cast Iron Radiators and Heating Accessories

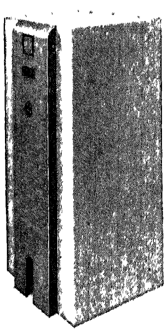
Irvington-on-Hudson, N. Y. — Zanesville, Ohio

Branch Offices

Lancaster, Pa. — Elizabeth, N. J. — Boston, Mass. — Baltimore, Md.
Pittsburgh, Pa. — Geneva, N. Y. — Philadelphia, Pa. — Springfield, Mass.
Zanesville, Ohio — Queens Village, N. Y. — Chicago, Ill.

Plants

Irvington, N. Y. — Elizabeth, N. J. — Lancaster, Pa. — Zanesville, Ohio — Geneva, N. Y.



Yello-Jacket Boiler (for coal or oil) For large and small homes

BOILERS

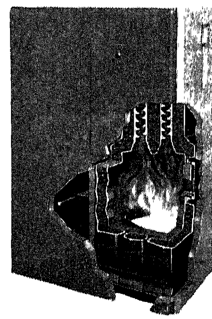
Jacketed—Unjacketed
Cast Iron—Steel
Hand-fired—Oil-fired
Stoker-fired—Gas-fired

Round or Square Sectional boilers for steam, vapor, or water.

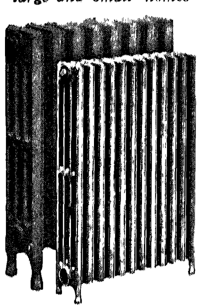
Water Tube boilers for steam and hot water heating.

Tube-type Smokeless boilers for burning soft coal.

Welded Steel boilers for both heavy duty and home heating.



Cast Iron DeLuxe Oil Burning Boiler



Slenderized Radiator—40% smaller than tube type

RADIATORS

Exposed and Concealed

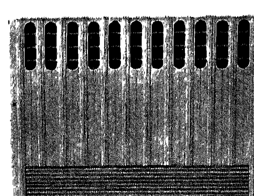
Cast iron radiators that occupy 40 per cent less space than ordinary radiators of same rating.

Shorter, lower, narrower.

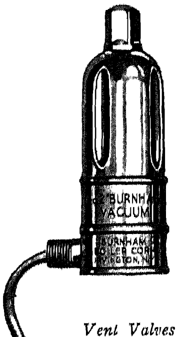
Can be recessed if desired.

In 3-tube, 4, 5, and 6-tube types.

Burnham cabinet type Radiant radiator.



Radiant Radiator with underneath concealed valve—Lower grille removable



Vent Valves for radiators

ACCESSORIES

Air and Vacuum valves for mains, risers and radiators.

Radiator valves in many designs.

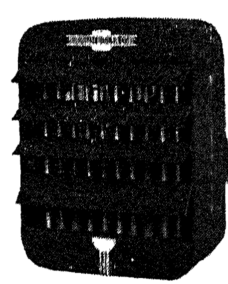
Unit Heaters—a complete line in modern designs.

Burnham-Taco tanks for hot water supply.

Flexible Headers for all purposes.

Unit Air Conditioners—for heating and winter air conditioning.

Attic Fans—complete units for cooling of residences.



Unit Heater

THE BABCOCK & WILCOX COMPANY

85 Liberty Street — New York, N. Y.

Manufacturers of

Water-Tube Boilers — Chain-Grate Stokers

Oil Burners — Seamless Steel Tubing and Pipe

Branch Offices and Representatives in all Principal Cities

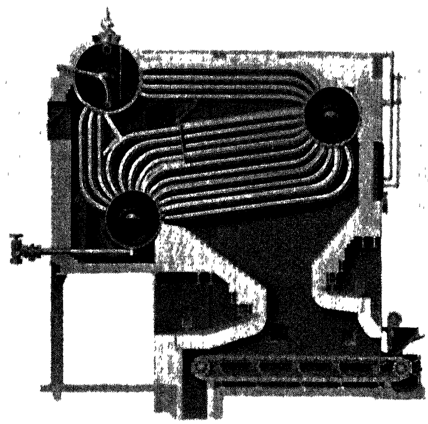
Type H Stirling Boiler

The Babcock & Wilcox Type H Stirling Boiler is a highly efficient unit built for moderate pressures at moderate prices.... and is designed to occupy minimum floor space and head room for the heating surface required.

This boiler is built in four classes and 36 sizes ranging from 691 to 6225 sq ft of heating surface, and can be designed for operation with any fuel and every method of firing.

The moderate price is due only to the simplicity of design, efficient production methods and superior shop equipment.

Class	Heating Surface	Depth of Setting, Ft, In.	Width of Setting: Single Boiler, Ft, In.	Width of Setting: Two Boilers in Battery Ft, In.	Floor to Center of Mud Drum, Ft, In.	Floor to Face of Steam Outlet, Ft, In.	Floor to Top of Boiler, Ft, In.	Size of Steam Outlet, In.
	691	15-2	6-0	11-0	5-2	14-5½	13-3⅛	5
	921	"	7-0	13-0	"	"	"	"
	1152	"	8-0	15-0	"	"	"	"
H-1	1382	"	9-0	17-0	"	"	"	"
	1612	"	10-0	19-0	"	"	"	"
	1843	"	11-0	21-0	"	"	"	"
	2073	"	12-0	23-0	"	"	"	"
	2304	"	13-0	25-0	"	"	"	"
	877	17-8	6-0	11-0	4-9¾	14-5½	13-3⅛	5
	1169	"	7-0	13-0	"	"	"	"
	1462	"	8-0	15-0	"	"	"	"
H-2	1754	"	9-0	17-0	"	"	"	"
	2046	"	10-0	19-0	"	"	"	"
	2339	"	11-0	21-0	"	"	"	"
	2631	"	12-0	23-0	"	"	"	"
	2924	"	13-0	25-0	"	"	"	"
	1063	20-2	6-0	11-0	4-5½	14-5½	13-3⅛	5
	1417	"	7-0	13-0	"	"	"	"
	1772	"	8-0	15-0	"	"	"	"
	2126	"	9-0	17-0	"	"	"	"
H-3	2480	"	10-0	19-0	"	"	"	"
	2835	"	11-0	21-0	"	"	"	"
	3189	"	12-0	23-0	"	"	"	"
	3544	"	13-0	25-0	"	"	"	"
	3898	"	14-0	27-0	"	"	"	6
	4252	"	15-0	29-0	"	"	"	6
	1245	22-8	6-0	11-0	4-1¼	14-11⅜	13-3⅛	5
	1660	"	7-0	13-0	"	"	"	"
	2075	"	8-0	15-0	"	"	"	"
	2490	"	9-0	17-0	"	"	"	"
H-4	2905	"	10-0	19-0	"	"	"	"
	3320	"	11-0	21-0	"	"	"	"
	3735	"	12-0	23-0	"	"	"	"
	4150	"	13-0	25-0	"	"	"	"
	4565	"	14-0	27-0	"	"	"	6
	4980	"	15-0	29-0	"	"	"	6



Type H Stirling Boiler with Babcock & Wilcox Chain-Grate Stoker

The advantages of the Babcock & Wilcox Type H Stirling Boiler may be summarized as follows:

Unusual steaming capacity for the floor space and head-room required.

Boilers may be set singly or in battery.

Setting heights can be varied to suit any condition of firing.

The choice of three locations for gas exit reduces cost of flues and breeching.

Distribution baffles make effective all of the heating surface.

Tube renewal is facilitated by correct tube spacing, and a tube removal door.

Soot blowers can be readily installed to simplify thorough cleaning of all tubes.

A superheater can be furnished without any change in the standard design or construction.

The boiler is supported by a structural-steel framework entirely independent of the brickwork.

Ample provision is made for free movement of parts due to expansion and contraction.

A complete table of sizes and dimensions, together with pertinent installation data, is contained in a new bulletin which will be sent upon request. Simply ask for Bulletin G-8-C.

Farrar & Trefts

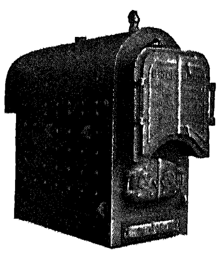
Incorporated

Buffalo, N. Y.

HEATING AND POWER BOILERS
- Bison Compact Boilers
- Bisonette Compact Boilers
- Firebox Return Tubular Boilers
- Firebox Locomotive Type Boilers
- Scotch Marine Type Boilers
- Vertical Boilers
- Horizontal Return Tubular Boilers
- Bison Two-Pass Return Tubular Boilers

Established 1864

STEEL PLATE CONSTRUCTION
- Class I Fusion Welding with X-raying and Stress Relieving
- Storage and Pressure Tanks
- Receivers, Steel Pipe, Buoys
- Bubble Towers, Condensers, Kettles
- Smokestacks and Breechings
- Special Work in Stainless Steel, Everdur, Nickel, Aluminum or Monel Metal

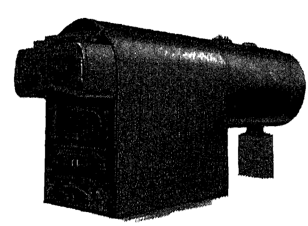


The Bison Compact

The **F&T Bison Compact Welded Heating Boiler** is more than just another boiler. It has been designed carefully so as to have a large furnace volume, the proper volume of water, just the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result is a remarkably steady water line—**A Balanced Boiler.**

This boiler requires a minimum amount of floor space and is easy and inexpensive to install. It is reasonable as to first cost and economical in operation. Construction is in accordance with the *A.S.M.E.* Code for 15 lb working pressure and boilers are designed for hand firing with anthracite or bituminous coal or for mechanical firing with oil, gas or stoker. There are various sizes available from 1800 to 35,000 sq ft of steam radiation, all ratings as required by the *Steel Heating Boiler Institute.*

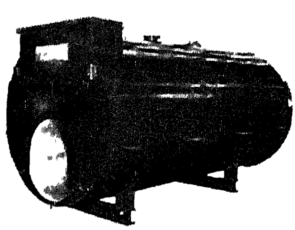
The **Bisonette Compact Boiler** has the same characteristics as the larger Bison Compact Boiler. It has been designed for installation in large residences and small business establishments where the advantages inherent in a **Steel** boiler are desired.



Firebox Return Tubular Boiler

Firebox Return Tubular Heating Boilers are **Quality Boilers.** They are constructed to measure up to the high standards set by Heating Engineers and will give unfailing service under all conditions. Being economical to install and operate, they are highly favored by Architects and Engineers for heating Schools, Hospitals, etc.

There are two types of Firebox Boilers, the Up-Draft Type and the Down-Draft Type. Both types are made of welded or riveted construction for heating purposes at 15 lb working pressure and riveted, or, Class 1 fusion welded x-rayed and stress-relieved for power purposes at 100, 125 and 150 lb working pressure in accordance with the *A.S.M.E.* Code. Sizes from 1800 to 35,000 sq ft of steam radiation, as rated by the *Steel Heating Boiler Institute*, are designed for hand firing with coal or for mechanical firing with oil, gas or stoker.



The **Bison Low Pressure Scotch Wet-Back Top Boilers** are *carefully proportioned* and *balanced.* They are designed for hand, oil, gas or stoker firing, for ratings from 15 to 250 hp. These boilers operate efficiently and carry sustained overloads. The *Front Smokebox Door Open Sideways* giving easy access to the tubes.

The *Wet-Back Top* increases the heating surface and steam disengaging area, thus adding to the capacity of these boilers. **F & T** boilers are designed so that the round furnace is always longer than the tube length which increases the furnace volume. This gives a large combustion volume in proportion to horsepower rating which makes the boilers very *economical to operate* and exceedingly "*Quick Steamers.*"

Crane Co.

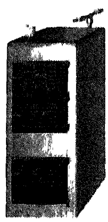
BOILERS, RADIATORS, VALVES, FITTINGS, PIPE, STEAM SPECIALTIES, PLUMBING AND HEATING MATERIALS

General Offices: 836 South Michigan Avenue, **Chicago, Illinois**

Nation-Wide Service Through Branches, Wholesalers, Plumbing and Heating Contractors

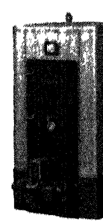
A complete line of heating equipment boilers and furnaces for coal, coke, oil, or gas burning for steam, hot water, or warm air systems. Full descriptions and specifications are given in your Crane Catalog or supplied on request.

BOILERS FOR SMALL HOMES



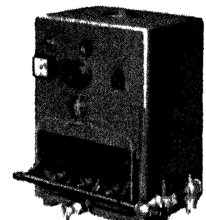
SERIES FOURTEEN

Wet base; low return inlet. Patented controlled water travel. Large ceiling heating surface. Internal heater and jacket optional. For steam or hot water. Capacities: manual firing, up to 90,000 Btu., oil or stoker up to 119,000 Btu. (IBR).



CONSERVOIL UNIT

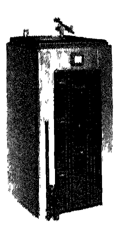
Low-priced boiler-burner unit in 4 sizes up to 131,000 Btu. (IBR) Controlled water travel, large ceiling surface, and flue inserts assure fuel economy. Includes burner, draft regulator and 3 controls. For steam or hot water.



No. 2WG BASMOR GAS BOILER

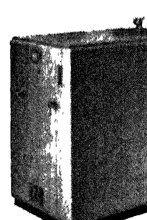
New hot water boiler for smallest homes. Sections are cast-iron with water-jacketed combustion chamber. Fully automatic. Shipped completely assembled; housing, controls in position. Up to 110,800 Btu. net capacity.

BOILERS FOR AVERAGE-SIZE HOMES



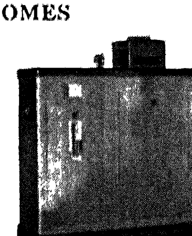
No. 10 ALL-FUEL BOILER

Can be installed for manual firing—easily converted for stoker, oil or gas firing. High base and removable grate lugs give ample space for stoker or oil burner. Provision for internal heater. For steam or hot water. Net capacity up to 207,000 Btu. (IBR).



No. 16 SUSTAINED HEAT BOILER-BURNER UNIT

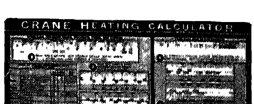
Application of Crane sustained heat principle extracts more heat from fuel. Down-draft flue construction prevents escape of combustion gases before heat has been absorbed. Net capacity up to 216,000 Btu. (IBR). Steam or water.



No. 25 BASMOR GAS BOILER

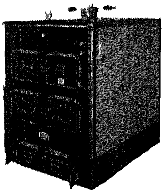
Unusual efficiency obtained with staggered fin construction and improved Bunsen-type burners. Safe, can't back-fire. Simple controls. Many sizes; for manufactured and natural gas. Net capacity to 177,400 Btu. Steam or hot water.

CRANE HEATING CALCULATOR FREE



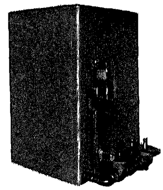
With this accurate calculator, employing the A.S.H.&V.E. method of determining heat losses, you can quickly select the right boiler and radiator requirements for any job. Easy to use—slide rule type. Free on request. Please write on your letterhead to address at top of this page.

BOILERS FOR LARGER BUILDINGS



No. 4 SECTIONAL BOILER

For manual, oil, and stoker firing. Up to 1,756,800 Btu. net capacity- steam or hot water.

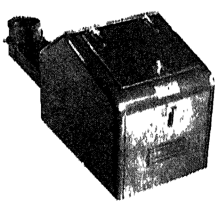


SERIES 60 BASMOR GAS BOILER

Has built-in steam header with Hartford return loop. Up to 2,865,600 Btu. net capacity—steam or hot water.

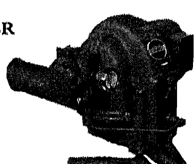
AUTOMATIC HEAT-CONVERSION UNITS

AUTOCOAL STOKER



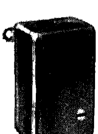
For even, controlled room temperature with minimum attention. Hopper models: 20 to 350 lb. per hour capacity. 35 and 50 lb. bin-feed models.

CONSERVOIL BURNER

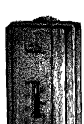


Will burn lower grades of fuel oil. Only one moving part. Quiet; cannot foul. Models up to 35 gal. per hour capacity.

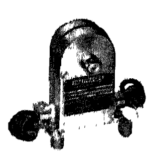
CONTROLS



Low Voltage Relay-Transformer



Room Thermostat



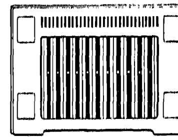
Draft Tender

A full line of precision-built Crane controls including room thermostats, night set-back clocks, oil and stoker controls, limit switches for steam, hot water, and furnace systems.

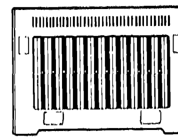
The Crane line includes valves, fittings, and pipe for all boiler and radiator systems; a selection of furnaces for coal, oil, and gas; also split-system equipment and well-water cooling for year 'round air conditioning.

HEATING ELEMENTS—ALL TYPES

COMPAC SLIM-TUBE RADIATORS

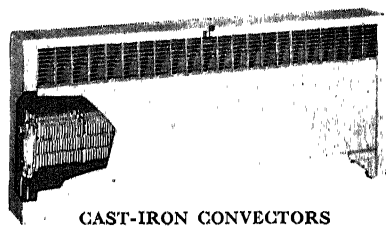


End Connection



Bottom Connection

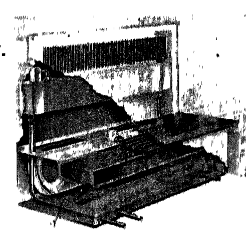
Cast-iron; space-saving. Modern slender design. For free-standing or recessed installation—with or without attractive front panel. Maximum delivery of radiant, infra-red ray heat. Further space-saving with bottom connections.



CAST-IRON CONVECTORS

Enclosures of heavy steel, smartly styled. Models for fully or partially recessed, free-standing, wall-hung, and plaster-front installations. Convectors of sturdy cast-iron with large integral fins designed to stimulate air flow. For all systems.

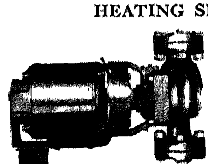
DUCTLESS WINTER AIR-CONDITIONING UNIT



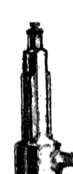
Recessed in wall and floor; no sheet metal work. Provides heat, humidification, filtering and circulation.

FOR LARGE SPACE HEATING REQUIREMENTS, SPECIFY CRANE SPEED HEATERS. A COMPLETE LINE

HEATING SPECIALTIES



Circulating Pump



Steam Venting Valves

Crane supplies a complete range of hot water specialties including circulators, flow controls, monoflo fittings, pressure tank systems, indirect heaters. Also, air valves, traps, condensation and vacuum pumps, low-water cut-offs, and other steam specialties.

Fitzgibbons Boiler Company, Inc.

Established 1886

General Offices: Architects Bldg., 101 Park Avenue

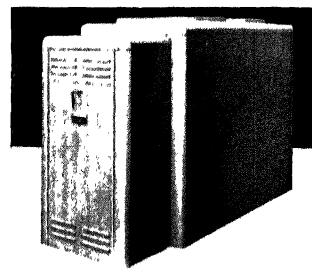
New York, N. Y.

Works: OSWEGO, N.Y.

Branches and Representatives in Principal Cities

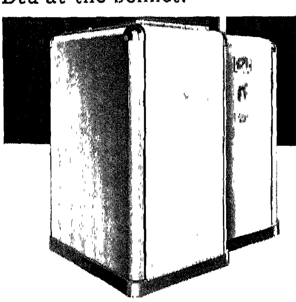
PRODUCTS—STEEL HEATING and POWER BOILERS for all fuels and all heating systems. Capacities to meet requirements of any building. Built and rated according to *S. H. B. I.* Code. —AIR CONDITIONERS for "Split-Systems" and for Direct-Fired installations in residences of all sizes.

Direct-Fired Air Conditioners



The DIRECTAIRE—The conditioner that has broken the shackles of traditional "hot air furnace" design, providing far greater Efficiency, Ruggedness, Quietness, Fuel Economy, Cleanability. Streamlined jacket in two types. Nine sizes 65,000 to 600,000 Btu at the bonnet.

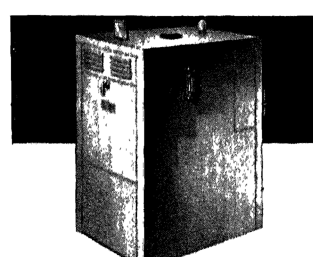
The SPECIAL 80-120 DIRECTAIRE—Designed and priced, without sacrifice in quality, for the small home budget. Already in successful use in scores of housing developments. Two sizes 80,000 to 120,000 Btu at the bonnet.



"Split-System" Air Conditioners

The FITZGIBBONSAIRE

combines with Fitzgibbons Steel Boilers for automatic firing with oil, gas or stoker, to provide: (1) CONDITIONED AIR (cleaned, humidified, tempered, circulated) to all rooms where desired; (2) RADIATOR HEAT to kitchen, baths, garage and other parts through which recirculation is undesirable; (3) YEAR-'ROUND DOMESTIC HOT WATER, without a storage tank.



Steel Heating Boilers

The 400 SERIES—A boiler for small home heating that brings steel boiler economy and comfort within the building budget of the low-cost house. Built of welded copper-steel throughout, crack-proof, leak-proof, corrosion-resistant, quick-heating. Cooperates with any good oil burner, stoker, gas burner. Gives year-'round domestic hot water, with or without tank. Heavy insulated jacket. Ratings, Steam—3 sizes—400 to 680 sq ft.

The OIL-EIGHTY AUTOMATIC*—An outstanding residential steel boiler for oil firing. Teams up with any good rotary or gun type burner to form a highly efficient unit. Provides room for burner inside the jacket. Year-'round tankless domestic hot water optional. Ratings, Steam—12 sizes—425 to 2680 sq ft.

The GAS-EIGHTY—For gas. Jacketed. Ratings, Steam—12 sizes—425 to 2680 sq ft.

The STOKER-EIGHTY—For anthracite and bituminous stokers. Jacketed. Stoker may be installed at either side if desired, to allow free access for inspection through door in front. Supplies year-'round hot water with or without tank as desired. Approved by *Anthracite Industries, Inc.* Ratings, Steam—6 sizes—485 to 2000 sq ft.

*Reg. U.S. Pat. Office.

FITZGIBBONS R-Z-U JUNIOR

Multi-Service Steel Boiler

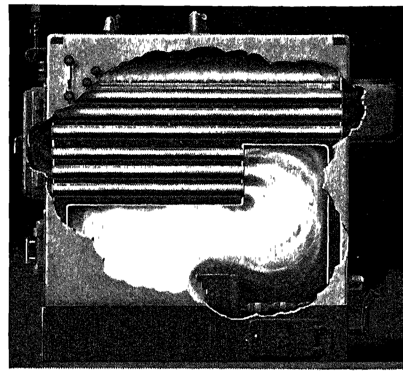
RATINGS, STEAM

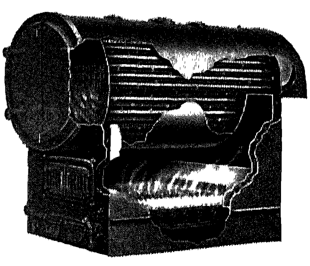
Coal Burning Type........ 900 to 3200 sq ft
Oil Firing Type...............1100 to 3900 sq ft
Stoker Firing Type.........1100 to 3900 sq ft

Outstanding Features

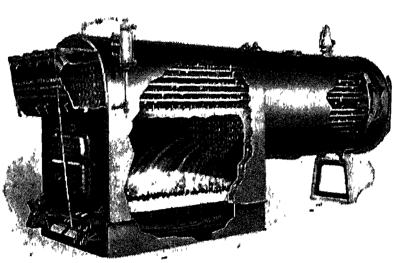
Tanksaver (optional) supplies year-'round hot water without a separate storage tank. **Tankheater** (optional) a more efficient indirect water heater. **Auxiliary Grate** (optional), for refuse disposal and stand-by heating duty in oil fired installations. **Compact,** largest size will pass thru a 31 in. doorway. **Low Water Line,** eliminates need for a pit. **Jacket** (optional), on all types.

Descriptive Bulletin on Request

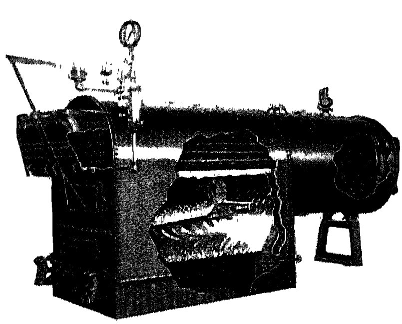




R-Z-U



500 Series



600 and 800 Series

FITZGIBBONS Z-U
Steel Firebox Boilers

Built for 15 lb w.s.p.—*A.S.M.E.* Code.
Up-Draft Type......1800 to 35,000 sq ft steam

FITZGIBBONS R-Z-U
Steel Firebox Boilers

The Z-U arranged for rear smoke outlet.
Built for 15 lb w.s.p.—*A.S.M.E.* Code.
Up-Draft Type......1800 to 35,000 sq ft steam
Smokeless Type......1800 to 35,000 sq ft steam
Oil, Gas, Stoker....2190 to 42,500 sq ft steam

FITZGIBBONS "F" SERIES
Portable Riveted Firebox Boilers

Built for 100 lb w.s.p.—*A.S.M.E.* Code.
Ratings, steam—1800 to 15,000 sq ft

FITZGIBBONS 500 SERIES
Portable Welded Firebox Boilers—Return Tubular

Built for 15 lb w.s.p.—*A.S.M.E.* Code
Ratings, steam—3500 to 35,000 sq ft

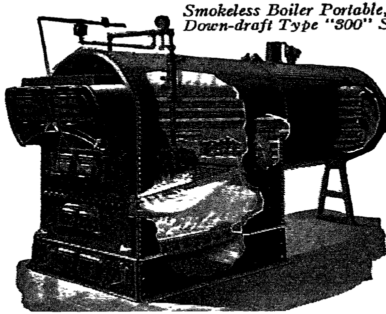
FITZGIBBONS 700 AND "P" SERIES
Portable Riveted Firebox Boilers

700 Series for 15 lb w.s.p.—*A.S.M.E.* Code.
Ratings, steam—3500 to 35,000 sq ft
"P" Series for 100 lb w.s.p.—*A.S.M.E.* Code.
Ratings, horsepower—25 to 250.

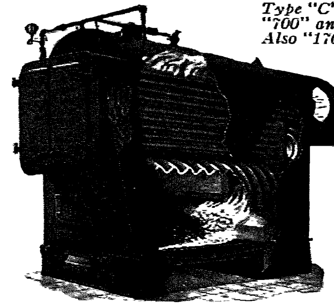
FITZGIBBONS 600 AND 800 SERIES
Smokeless Down-Draft Riveted Firebox Boilers

Built for 15 to 100 lb w.s.p.—*A.S.M.E.* Code.
Ratings, steam—3500 to 35,000 sq ft

Descriptive Bulletins on any or all of above boilers will be mailed on request.

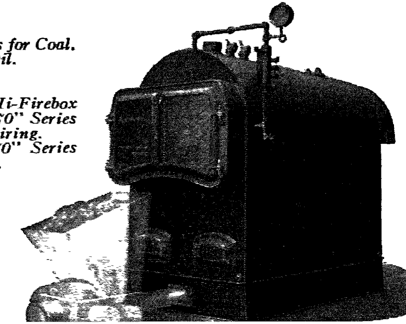


Smokeless Boiler Portable, Down-draft Type "300" Series



Type "C" Boiler "700" and "2700" Series for Coal. Also "1700" Series for Oil.

At Right: Type "C" Hi-Firebox Boiler, "7L70" Series for Stoker Firing. Also "27L70" Series Hand Fired.



SPECIFICATIONS—SMOKELESS DOWN-DRAFT BOILER

Boiler No.	376	377	378	379	380	381	382	383	384	385	386	387	388	389	390
Rated Steam Capacity:															
Coal ... Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker ... Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length ... In. x Ft In.	42x8-3	42x9-3	48x8-7½	48x9-5	54x11-2½	54x12-11	60x12-8½	60x14-9	66x14-11	66x17-4	72x16-5	78x17-0½	78x20-7½	84x19-11	84x22-8
Overall Height Shell ... In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line ... In.	70	70	73	73	78	78	85	85	88½	88½	94½	95	95	107½	107½
Approximate Weight: Coal ... Lb	6800	7500	8100	8800	10000	11100	13600	15300	18000	20500	22900	25100	29500	33600	37500
Oil ... Lb	6200	6900	7400	8000	9100	10100	12500	14100	16500	18900	21200	23200	27500	31500	35200

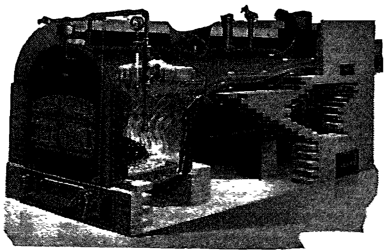
SPECIFICATIONS—TYPE "C" WELDED BOILER

*Boiler No.	2773	774 2774	775 2775	776 2776	777 2777	778 2778	779 2779	780 2780	781 2781	782 2782	783 2783	784 2784	785 2785	786 2786	787 2787	788 2788	789 2789	790 2790
Rated Steam Capacity:																		
Coal ... Sq Ft	2200	2600	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker . Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width & Length, In.xFt in.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height Shell ... In.	77½	77½	77½	77½	83½	83½	83½	86½	86½	99	99	108½	112	118	118	122	135	135
Height of Water Line .. In.	69	69	69	69	72	72	72	73	73	85	85	94	95	101	101	103	114	114
Approximate Weight:																		
700 Series, Coal ..... Lb	....	3900	4400	5000	5500	6000	6500	7500	8400	9700	11000	12900	14900	16600	18400	22000	25200	28400
2700 Series, Coal .... Lb	3400	3800	4300	4800	5300	5800	6300	7200	8100	9400	10600	12500	14400	16100	17900	21200	24400	27500
Oil ..... Lb	2900	3300	3700	4100	4600	5000	5400	6100	6900	8000	9100	10700	12300	13800	15400	18100	20900	23400

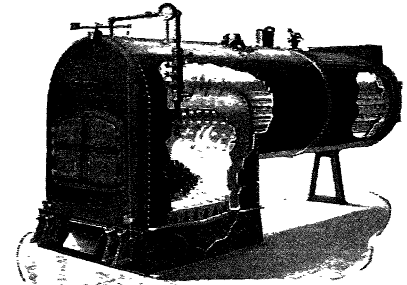
SPECIFICATIONS—TYPE "C" HI-FIREBOX WELDED BOILER

*Boiler Series 1773-1790 for Oil, Gas, Stoker; Series 2773-2790 for Anthracite.

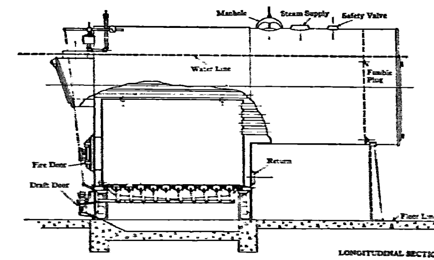
Boiler No.	7L73	7L74	7L75	7L76	7L77	7L78	7L79	7L80	7L81	7L82	7L83	7L84	7L85	7L86	7L87	7L88	7L89	7L90
Rated steam capacity,																		
Stoker ... Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width x Length, In.xFt In.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height ... In.	82½	82½	82½	82½	88½	88½	88½	94½	94½	108	108	119½	123	131	134	140	157	161
Height of Water Line .. In.	74	74	74	74	77	77	77	81	81	94	94	105	106	114	117	121	136	140
Approximate Weight .. Lb	3100	3500	3900	4400	4900	5400	5800	6700	7600	8800	10000	11800	13600	15300	17000	20100	23200	26100



Brickset Boiler Up-draft Type



Type "K" Up-draft Boiler



Weld + Rivet Firebox Boiler "5000" Series Up-draft. Also "6000" Series Smokeless Down-draft

SPECIFICATIONS—BRICK-SET AND TYPE "K" PORTABLE UP-DRAFT BOILERS

Boiler No.	3 3K	4 4K	5 5K	6 6K	8K	8 9K	9 10K	10 11K	11 12K	12 13K	13 14K	14 15K	15 16K	17 18K	18 20K	19	20
Rated Steam Capacity:																	
Coal....Sq Ft	1240	1380	1800	2200	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000
Oil, Gas or Stoker....Sq Ft	1770	2020	2190	2680	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290
Width and Length ...In. x Ft In.	30x9-10	36x8-10	36x10-4	36x11-10	42x11-4	42x12-10	48x11-10	48x13-4	48x14-10	54x15-3	54x17-9	60x17-0	60x19-6	66x19-6	72x19-6	72x16-0	72x18-0
Overall Height Shell....In.	59	65	65	65	71	71	77	77	77	83	83	93	93	99	105	107	107
Height of Water Line....In.	52	55	55	55	58½	58½	61	61	61	66	66	75	75	80	85½	85½	85½
Approximate Weight: Coal....Lb	3500	3900	4200	4700	5700	6400	7000	7600	8200	9500	10700	12600	14500	17600	20700	19400	21900
Oil....Lb	3130	3550	3800	4200	5100	5800	6400	6900	7400	8600	9700	11500	13300	16100	19100	17700	20000

SPECIFICATIONS—WELD + RIVET FIREBOX BOILER "5000" SERIES

Boiler No.	5076	5077	5078	5079	5080	5081	5082	5083	5084	5085	5086	5087	5088	5089	5090
Rated Steam Cap: Coal....Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, gas or stoker....Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Diam. x Length....In. x Ft In.	42x7-3	42x8-0	42x8-11	42x9-10	48x9-9	48x11-1¼	54x11-1	54x12-9	60x13-2½	60x15-3	66x14-5½	66x16-0½	72x16-8½	78x17-5½	78x19-8½
Overall Height....In.	81	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line....In.	70	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Approx. Weight: Coal....Lb	5100	5600	6100	6600	7500	8400	9600	10800	12500	14400	16100	17900	21200	24400	27500
Oil....Lb	4500	5000	5400	5800	6600	7400	8500	9600	11100	12800	14400	16000	19200	22300	25200

SPECIFICATIONS—WELD + RIVET SMOKELESS BOILER—"6000" SERIES

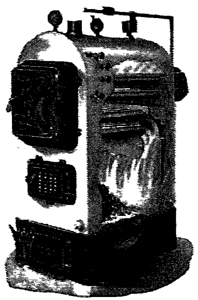
Boiler No.	6077	6078	6079	6080	6081	6082	6083	6084	6085	6086	6087	6088	6089	6090
Rated Steam Capacity: Coal....Sq Ft	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker....Sq Ft	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Diameter x Length....In. x Ft In.	42x8-2	42x8-8	42x9-6	48x9-7	48x11-0	54x10-8½	54x12-4½	60x12-9½	60x14-9½	66x14-0½	66x15-7½	72x16-4½	78x17-1	78x19-4
Overall Height....In.	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line....In.	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Approximate Weight: Coal....Lb	5800	6300	6800	7700	8600	9900	11200	13100	14900	16600	18300	21500	24500	27300
Oil....Lb	5100	5600	6100	7000	7800	8900	10100	11900	13500	15100	16700	19800	22700	25300

KEWANEE TYPE "R" RESIDENCE BOILERS

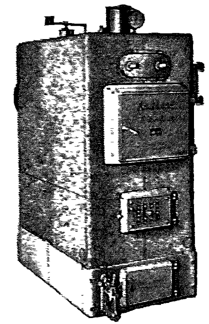
Kewanee Type "R" Boilers are especially designed and constructed to meet all heating and hot water requirements for homes and small buildings. Every kind of solid fuel, coke, all grades of hard or soft coals and their briquette or treated forms are burned with excellent results. Also, any liquid fuel, oil, and natural or commercial gas can be used with high efficiency.

Standard snug fitting jackets, or Regal style for completely enclosing burners are available for Round "R" and 83R.

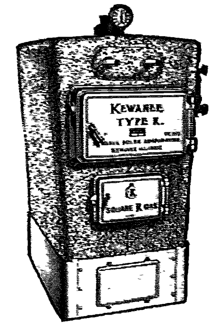
Hot Water Copper Coil. Capacities up to 720 may be ordered for Square and 83R Boilers.



Square Type "R" Residence Boiler



Square Type "R" Jacketed Boiler



83R Oil or Gas Jacketed Boiler

SPECIFICATIONS—RESIDENCE SQUARE TYPE "R" BOILER

*Boiler No.	742	743	745	746	747	748
Rated Steam Capacity: Coal ... Sq Ft	790	1000	1350	1600	1780	1960
Oil, Gas or Stoker ... Sq Ft	840	1120	1470	1900	2160	2380
Width and Length ... In. x In.	32x39½	32x45½	32x45½	32x51½	32x57½	32x63½
Overall Height Shell Top ... In.	59½	59½	70¼	70¼	70¼	70¼
Height of Water Line ... In.	48	48	58½	58½	58½	58½
Approximate Weight: Coal ... Lb	2150	2360	2800	3050	3300	3550
Oil ... Lb	1900	2060	2500	2730	2920	3125
Standard Jacket, Crated ... Lb	205	225	175	190	200	225

Boiler No., Square "R" Oil or Gas	83R1	83R2	83R3	83R4	83R6	83R7	83R8	83R9
Rated Steam Capacity ... Sq Ft	901	1105	1326	1513	2091	2363	2652	2924
Approximate Weight with Jacket ... Lb	1600	1800	2000	2200	2800	2900	3100	3300

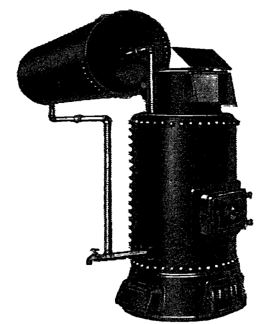
*Boiler Series 1742-1748 for Oil, Gas or Stoker; 2742-2748 for Anthracite. Kewanee Indirect Hot Water Heating Coils for Type C and Square "R" Boilers; 55 sizes, 90 to 1520 Gal.

KEWANEE WATER HEATERS AND TANKS

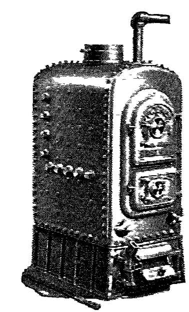
Tabasco Water Heaters, steel riveted, 17 sizes, heat 130-700 gal 50 deg per hour. Standard 60 lb: Extra Heavy 100 lb W. P.

Water Heating Garbage Burners. 3 types, 16 sizes, heat 200-2600 gal 50 deg per hour. Working Pressure up to 125 lbs.

Water Storage Tanks, Riveted and Welded, 48 sizes, 66-1700 gal. Also Pressure and Pneumatic Tanks, Air Receivers, Gravel Basins.



Tabasco Water Heater, Magazine Feed



Water Heating Garbage Burners Type D

The International Boiler Works Company

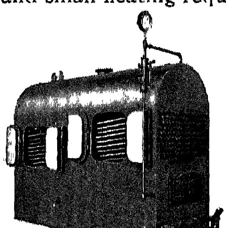
East Stroudsburg, Pa.

"Fuel Saver" Water Tube Steel Heating Boilers

SALES OFFICES

Alaska
Albany, N. Y.
Albuquerque, N. M.
Boston, Mass.
Buffalo, N. Y.
Cincinnati, O.
Detroit, Mich.
E. Stroudsburg, Pa.
Harrisburg, Pa.
Jackson, Miss.
Lexington, Ky.
Los Angeles, Calif.
Louisville, Ky.
Missoula, Mont.
Newark, N. J.
New Haven, Conn.
New York City, N. Y.
North Olmsted, O.
Philadelphia, Pa.
Pittsburgh, Pa.
Pittsfield, Mass.
Poughkeepsie, N. Y.
Providence, R. I.
Rochester, N. Y.
St. Paul, Minn.
Salt Lake City, Utah
Scranton, Pa.
Springfield, Mass.
Syracuse, N. Y.
Tampa, Fla.
Toronto, Ont., Canada
Utica, N. Y.
Washington, D. C.
White Plains, N. Y.
Wilmington, Del.

International "FUEL SAVER" Water Tube Steel Heating Boilers offer the same quick steaming and economy that have long been accepted as most efficient in marine and industrial service. "FUEL SAVER" Water Tube Boilers are available for large and small heating requirements in a wide range of types and capacities.



TYPE C "FUEL-SAVER" WATER TUBE STEEL HEATING BOILERS

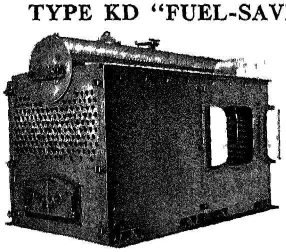
For Office and Apartment Buildings, Schools, Hotels, Theaters, Institutions and Industrial Plants

Built in a complete range of standardized sizes and provide highly efficient performance for heating large buildings.

Up-to-date water tube design permits absorbing the intense heat released by modern methods of firing and they will operate efficiently under loads considerably in excess of ratings.

18 sizes { from 2080 to 42,500 sq ft mechanically fired rating.
from 2200 to 35,000 sq ft hand fired rating.

TYPE KD "FUEL-SAVER" WATER TUBE STEEL HEATING BOILERS



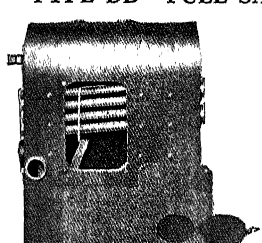
For Replacement Installations in Large Buildings Eliminates Costly Cutting and Patching

Especially designed for renovation and replacement work. Shipped knocked down in standardized parts that can be taken through existing doors or openings to basement and boiler room.

International erects or assumes full responsibility for erection work of knocked down boilers.

15 sizes { 5850 to 56,470 sq ft mechanically fired rating.
4810 to 46,510 sq ft hand fired rating.

TYPE DD "FUEL-SAVER" WATER TUBE STEEL HEATING BOILERS

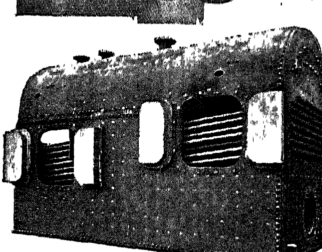


For Residences, Small Apartments and Other Buildings

Highly economical. Superior performance of water tube design usually found only in larger commercial and industrial installations. Fully utilize intense heat generated by modern automatic firing devices. Stoker-fired coal and oil most commonly used, with savings often exceeding 20 per cent over previous installations. Stoker-fired boiler tested and approved by Anthracite Industries Laboratory.

A copper coil submerged above the crown furnishes ample hot water for domestic needs. Insulated steel jackets in two tones of gray enamel are included.

10 sizes { 510 to 2550 sq ft net steam rating.
816 to 4080 sq ft net hot water rating



TYPE CR "FUEL-SAVER" WATER TUBE STEEL POWER BOILERS

For High Pressure Steam Service

Designed for pressure of 100, 125 and 150 lbs, the Type CR is especially suitable for hospitals, hotels, laundries, dairies, institutions and manufacturing plants requiring process steam.

Sizes range from 5 to 300 hp.

Data and catalogs on "Fuel-Saver" Boilers will be furnished on request.

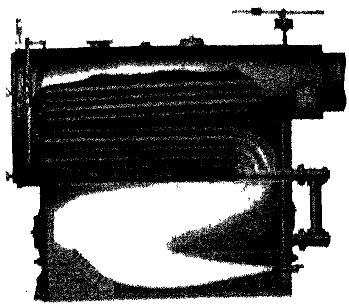
Pacific Steel Boiler Division

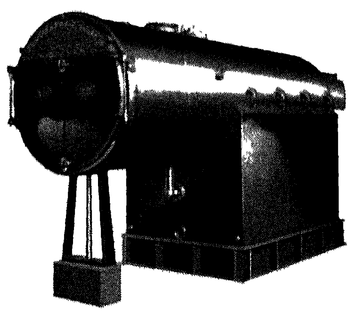
United States Radiator Corporation

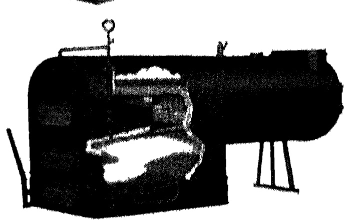
General Offices: **Detroit, Michigan**

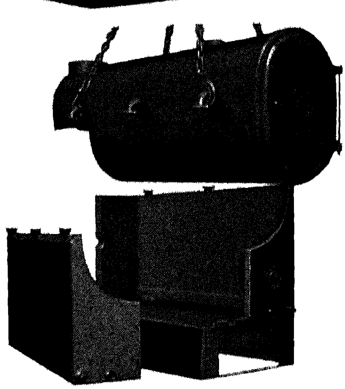
Sales Offices in Principal Cities

A Complete Line of Low Pressure Steel Heating Boilers









All Pacific Boilers are built using the *A.S.M.E.* Boiler Code Standards as minimums.

LOW WATER LINE SERIES

Built in the following capacities for steam:

Coal Burning Sizes—1800 to 35,000 sq ft.

Mechanically Fired Sizes—2680 to 42,500 sq ft.

High Fire Box for Stoker Firing—Sizes—2680 to 42,500 sq ft.

All Pacific Boilers are built, inspected, and tested under the supervision of the Hartford Steam Boiler Inspection and Insurance Company.

TWO-PASS FRONT SMOKE OUTLET

Built in the following capacities for steam:

Coal Burning Sizes—4000 to 30,000 sq ft.

Mechanically Fired Sizes—4860 to 42,500 sq ft.

All Pacific Boilers are made of steel with each joint and seam electrically arc-welded —built to last a life-time.

SINGLE-PASS REAR SMOKE OUTLET

Built in the following capacities for steam:

Coal Burning Sizes—1800 to 6000 sq ft.

Mechanically Fired Sizes—2190 to 7290 sq ft.

PACIFIC THREE-PIECE CONSTRUCTION

Made up of three parts, shell, firebox and base, Pacific Boilers are particularly adaptable to replacement work. Where necessary Pacific fireboxes can be split (as illustrated) allowing the boiler to be taken into the building in four pieces and erected without welding on the job.

Descriptive Bulletins on Pacific Steel Boilers will be mailed on request.

Spencer Heater Division

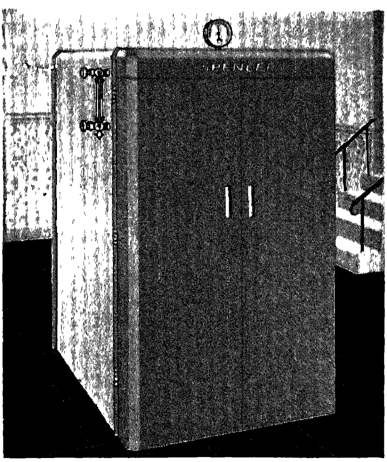
Aviation Manufacturing Corporation

Williamsport, Pa.

Sales Representatives in Principal Cities

Spencer Automatic Magazine Feed Heaters are furnished in cast iron sectional types —and steel tubular types for larger buildings—for steam, vapor and hot water heating. There is a size and capacity for every type of building, to provide economical and convenient heat—safe, dependable, sure.

COMFORTABLE HEAT AT LOW COST



Spencer Jacketed Heater L-1 Series

Why Spencer Heaters perform so satisfactorily can best be explained by an inspection of their design and construction. The Spencer principle, illustrated in the cross-sectional view, is simple:

Once a day fuel (No. 1 Buckwheat Anthracite or small size by-product coke) is put into the magazine. It fills the sloping grate to the level of the magazine mouth. The fire bed always stays at the proper level, for as fast as fuel burns to ash, it shrinks and settles on the sloping grate; and more fuel rolls down automatically over the top of the fire bed. Fuel feed is by gravity alone, in just the right amount to keep the fire always burning at its most efficient combustion point.

This explains why a Spencer Automatic Magazine Feed Heater always gives the same uniform, satisfying heat, and burns less fuel. These exclusive Spencer advantages are available in all types of the magazine feed heaters and boilers.

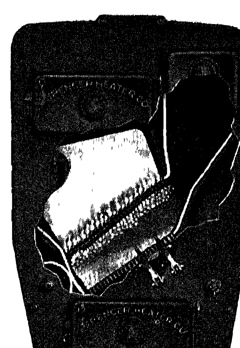
Coal—Coke—Gas—Oil—Spencer J and L series heaters and M series boilers are primarily designed to burn low cost No. 1 Buckwheat Anthracite or small size coke.

If at any time a property owner desires to burn more expensive fuels—oil or gas—his Spencer Heater can be readily converted and will show a high efficiency.

Thermostats Thermostats and electric damper motors are furnished as optional equipment.

Jacketed Covering Attractive metallic jackets of the deluxe enclosing type, as illustrated, are available for Spencer Cast Iron Heaters, either with or without the enclosing jacket doors.

Spencer Heavy Duty Tank Heaters—With the automatic magazine feed construction, they provide ample domestic hot water at lowest cost, and with a minimum of tank heater attention.



Cutaway sectional view Spencer Cast Iron Heater

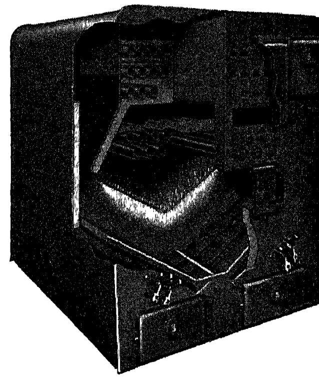
SPENCER ALL YEAR SYSTEM

In addition to the excellent heating facilities afforded by Spencer Magazine Feed Heaters, Year Round Domestic Hot Water Service can also be provided and assures at all times an ample supply of domestic hot water at lowest cost. Complete data for installation and operation upon request.

SPENCER STEEL TUBULAR MAGAZINE FEED BOILERS

For large buildings we recommend Spencer Steel Tubular Magazine Feed Boilers, burning low cost No. 1 Buckwheat Anthracite or coke.

In the cross-section diagram, part of the fire bed is cut away to show the sloping grates and the two magazines filled with fresh coal, ready to feed down automatically by gravity to the fire. These boilers are built in two vertical sections for ease in handling and installation—a great advantage on replacement jobs, eliminating the necessity of costly tearing out of walls or partitions. Combination water and fire tube construction; built to *A.S.M.E.* standards.



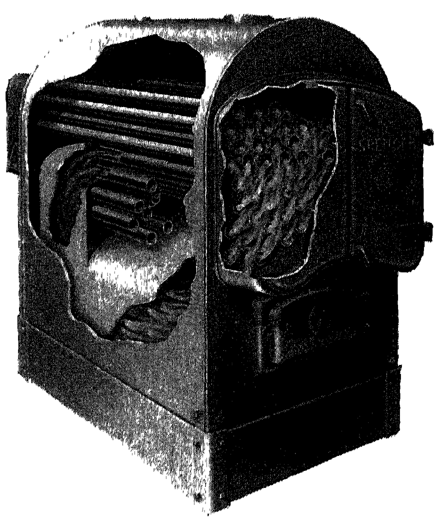
Steel Tubular Magazine Feed Boiler

SPENCER STEEL TUBULAR BOILERS
For Oil, Stoker, Gas or Hand-Firing

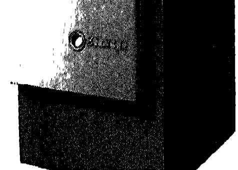
For more than 40 years, Spencer has been building, in the opinion of experts, one of the most efficient, economical and dependable automatic coal burning boilers on the market. With this background of experience, Spencer Engineers developed the Spencer Steel Tubular Boiler for oil, gas, stoker and hand-firing—the "K" and "C" series for residential use, and the Type "A" for larger buildings. It is a better boiler both for the property owner and for the architect or engineer who specifies it.

The high sustained efficiency of these boilers means adequate heat for a lower fuel cost. Design is of the three pass type. Combustion chamber is amply large. Built of best quality open hearth steel boiler plate, and steel tubes. Can be furnished with domestic hot water heating coils, storage tank or instantaneous type.

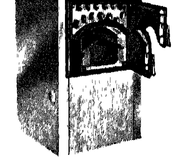
They are better boilers both for the property owners and for the architects or engineers who specify them, and provide a complete range of sizes from 400 sq ft SHBI net steam rating up. They meet or exceed in every particular the requirements of the A.S.M.E. and S. H. B. I. Codes.



Type "A" Steel Boiler



"C" Series Steel Boiler



"K" Series

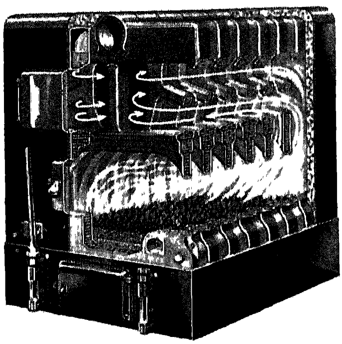
Every Spencer Boiler is guaranteed to carry more than its full rated load giving the installer a definite factor of safety.

These boilers have all the advantages of the Spencer exclusive design. High sustained efficiency—low fuel cost.

UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • Branches and Sales Offices in Principal Cities

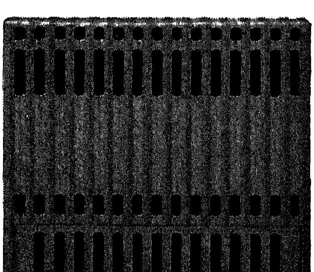
Detroit, Michigan



CAPITOL RED TOP BOILERS FOR ALL FUELS

	Capacity in Sq Ft	Direct Cast Iron Radiator Loads—Sq Ft
"A" Series	Steam— 575-1450 Water— 975-2475	240- 740 385-1185
"B" Series	Steam—1200-3600 Water—1980-5940	550-2030 910-3350
"C" Series	Steam—4700-10,500 Water—7760-17,325	1865-5805 3080-9580

Illustrated above is a Capitol Red Top Series "C" Boiler. Capitol Red Top Boilers can be furnished with extra high steel bases to provide extra setting height or desired additional furnace volume for stoker firing.



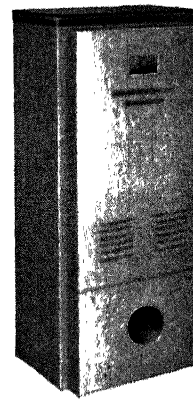
U. S. SUNRAY RADIATOR

Space Saving—Can be fully or partially recessed—Also well adapted for free standing installation.

Self-contained Cabinet Radiator—designed to form its own enclosure.

U. S. SUNRAY BOILER No 1. Series

Trade Mark U. S. Reg. Pat. Off.

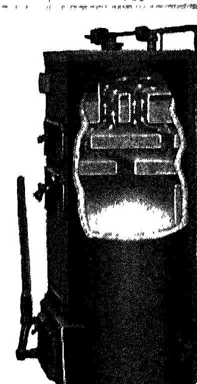


The "No. 1" Series Capitol Sunray Boiler is furnished in Coal-Burning and Oil-Burning types.

The unusually generous sized combustion chamber is scientifically proportioned so that it will operate at highest efficiency.

Boiler No.	STEAM	WATER
	Oil	Oil
1-03	260	415
1-04	381	610
1-05	503	805
1-06	625	1000
	Coal	Coal
1-04	300	480
1-5	450	720
1-6	600	960

CAPITOL RED CAP BOILERS
For All Fuels



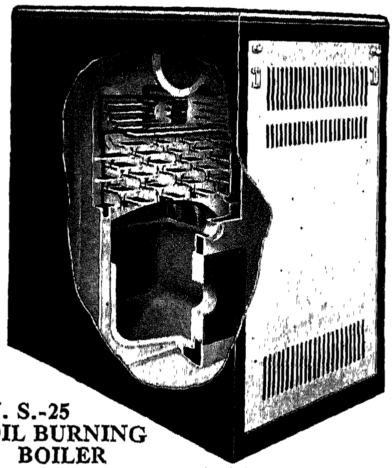
Boiler No.	Direct Cast Iron Radiator Loads Sq Ft	
	Steam	Water
19-4	300	495
19-5	350	580
20-4	400	660
20-5	450	745
22-4	500	825
22-5	550	910
25-4	625	1030
25-5	675	1115

Capitol Red Cap Boiler design brings the advantages of (1) long fire travel, (2) flue passages that force the hot gases to circulate through every section, extracting the maximum heat from the fuel consumed. (3) Deep firepot that provides the extra space needed for better combustion, and smooth, tapered firepot walls to assure a clean surface for better heat absorption.

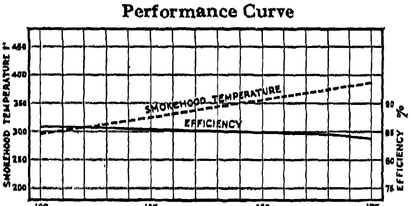
UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • Branches and Sales Offices in Principal Cities

Detroit, Michigan

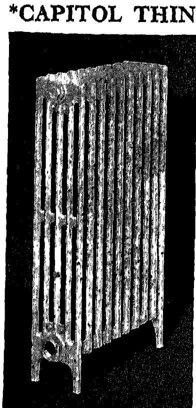


U. S.-25 OIL BURNING BOILER
Front Extended Jacket
Performance Curve



Output—% of Direct Standing Radiator Load Flue Gas Analysis CO_2—12.5%; O_2—4.1%; CO—0.0%.

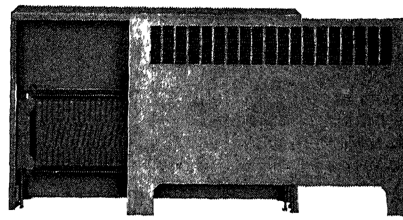
*CAPITOL THINTUBE RADIATORS



Heights	Per Section Heating Surface
3-Tube	
25″	1.6 Sq Ft
4-Tube	
19″	1.6 Sq Ft
22″	1.8 Sq Ft
25″	2.0 Sq Ft
5-Tube	
22″	2.1 Sq Ft
25″	2.4 Sq Ft
6-Tube	
19″	2.3 Sq Ft
25″	3.0 Sq Ft
32″	3.7 Sq Ft

*1¾ in. Centers.

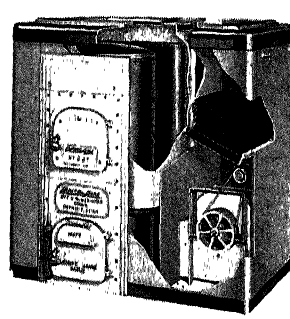
40 per cent less space needed for these graceful, efficient Capitol ThinTube Radiators.



CAPITOL CAST IRON CONVECTOR WITH ENCLOSURE

Capitol Cast Iron Convectors are made entirely of cast iron, without joints, and cast in one piece. A large variety of lengths and widths insures exact size to meet each need. These radiators are tapped top, bottom and ends.

A complete choice of enclosures. These units can be completely or partially recessed, or free-standing, adding to the attractiveness of any room when finished to harmonize with modern interiors.



THRIFT SERIES CAPITOLAIRE CONDITIONING UNIT

An air conditioning unit especially designed for all-fuel firing. Streamline flue construction in preheating and primeheating sections insures high efficiencies. The handsome enamelled casing houses the heating element, blower and filter.

The blower and filter compartment may be placed at either side of the heating element. The "Luxury" series units have blower section at rear only.

Other gravity and forced air furnaces available for gas, oil, stoker or hand coal firing.

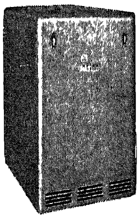
Weil-McLain Company

Manufacturing Division: **Michigan City, Ind. and Erie, Pa.**

General Offices: **641 W. Lake Street, Chicago**

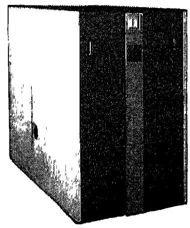
NEW YORK OFFICES: **501 Fifth Avenue**

Prompt Weil-McLain Boiler and Radiator service is made conveniently available through local stocks carried by Weil-McLain Distributors in most of the important distributing centers.



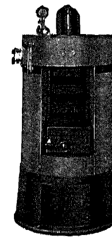
No. 68 Boiler for Automatic Firing

Boiler is completely jacketed and insulated. Has an integral front burner extension. Connected Load Ratings: Steam 380 to 650 sq ft, Water 610 to 1,040 sq ft.



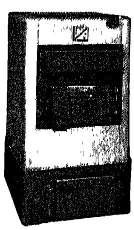
No. 78 Boiler for Automatic Firing

Boiler has insulated enameled de luxe jacket. Front or rear jacket extension available. Connected Load Ratings: Steam 400 to 1,030 sq ft, Water 640 to 1,650 sq ft.



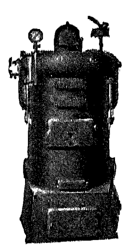
"RO Series" Boiler for Automatic Firing

Jacketed and insulated round boiler for small homes. Connected Load Ratings: Steam 420 and 520 sq ft, Water 630 and 790 sq ft.



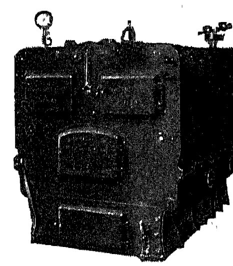
New All-Fuel Boilers No. 67 and No. 77

Conversion type boilers with insulated enameled jacket. For hand or automatic firing. Connected Load Ratings: Steam 350 to 1,030 sq ft, Water 560 to 1,650 sq ft.



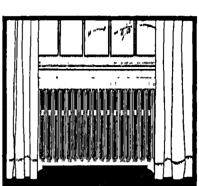
Round-Type Boiler

Unjacketed Round Boiler with corrugated heating surfaces for economical home heating. Connected Load Ratings: Steam 210 to 1,000 sq ft, Water 335 to 1,600 sq ft.



Square-Type Boilers

Sectional boilers for large installations. Complete range of sizes. Connected Load Ratings: Steam 1,020 to 11,300 sq ft, Water 1,630 to 17,900 sq ft.



Raydiant "Concealed"

A Raydiant convector type all cast-iron Radiator. Also made in "Concealed," Partially Recessed, Cabinet and Humidifying types.

Solray Radiator

Free standing Cabinet type Radiator in a lower price range than Raydiant Cabinet Radiators. Available in three depths in 21, 24 and 27 in. heights.

Junior Radiator

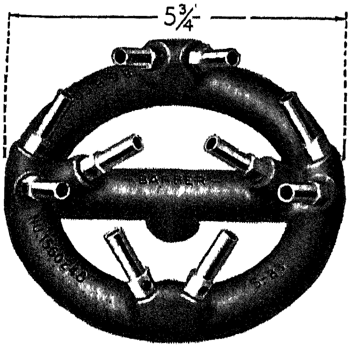
Smaller Tubular type Radiation which conserves space. Available in 1¾ in. centers in 3, 4 and 6 tube widths and 14 to 33 in. heights.

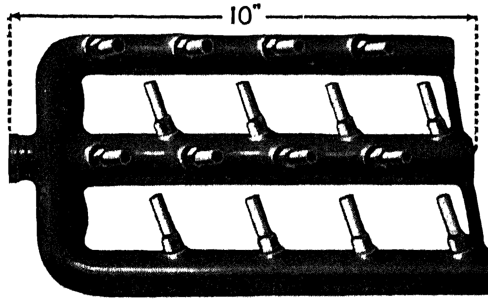
The Barber Gas Burner Company

3704 Superior Ave., **Cleveland, Ohio**

Address Michigan Inquiries to The Barber Gas Burner Co., of Mich., 4475 Cass Ave., Detroit, Mich.

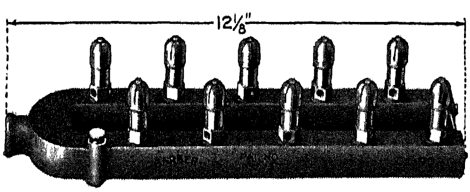
Barber Automatic Jet Gas Conversion Burners, for heating and air conditioning equipment, have a record of high efficiency, for a period of over 20 years, giving continuous satisfaction in many thousands of homes and other buildings in United States and Canada. The exclusive Barber Jet principle of combustion, attaining 1900 deg flame temperature on atmospheric pressure, and other basic advantages of design, have given Barber a permanent place in modern heating and air conditioning practice. Barber Burners for gas burning appliances have been adopted as standards by many appliance manufacturers. Shown here are only a few items from Barber's complete line. Illustrated No. 38A Catalog and Price List furnished on request.



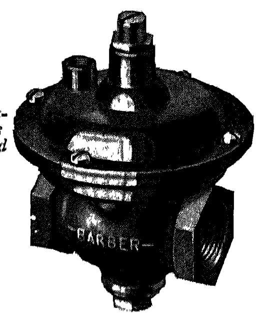


No. U-16 Burner Unit

No. S-85 Burner Unit

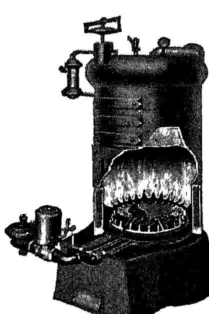


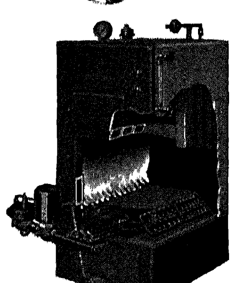
No. C. L-90 Burner Unit



Barber Gas Pressure Regulators A.G.A. Approved

Made in sizes ¼" and up.





Conversion Burners for Furnaces or Boilers

Burners are adjustable as to diameter, on the job, to fit practically all round grate sizes. Also to fit grates of oblong furnaces and boilers. Listed in the *A.G.A.* Directory of Approved Appliances. Equipped with automatic controls with motor gas valve, with magnetic gas valve control, with quick acting gas valve control (for buildings equipped with automatic heat control), or in "M" series with manual control.

Barber Burners and Regulators are Adaptable to: Air Conditioning Equipment, High Pressure Boilers (Tubular and Tubeless), Bakery Ovens, Garage Heaters, Coffee Urns, Hair Dryers, Space Heaters, Floor Furnaces, Clothes Dryers, Water Heaters, Confectioners' Stoves, Vulcanizing Machines, Pressing Machine Boilers, Japanning Ovens, Core Ovens, Banana Room Heaters, Other Appliances.

Gas Burner Specialists offering Engineering Department and Laboratory facilities for Gas Burner problems. Consultation invited.

The Webster Engineering Co.

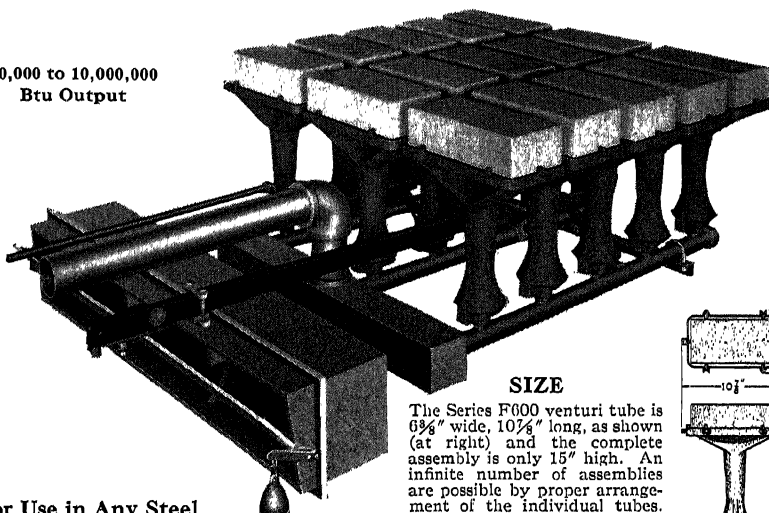
419 West 2nd St., **Tulsa, Oklahoma**

Division of

SURFACE COMBUSTION CORP., TOLEDO, OHIO

WECO-N.G.E. SERIES F600 GAS BURNERS

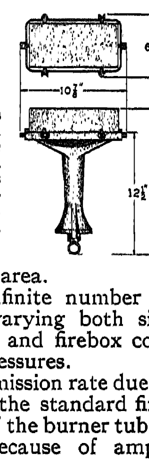
50,000 to 10,000,000 Btu Output



For Use in Any Steel Firebox or Sectional Boiler

SIZE

The Series F600 venturi tube is 6⅝″ wide, 10⅞″ long, as shown (at right) and the complete assembly is only 15″ high. An infinite number of assemblies are possible by proper arrangement of the individual tubes. For complete sizing information see Bulletin F600H.



Improved venturi and greater port area insure much higher capacities at lower pressures.

Unique baffles at the outlet of the mixing tube make possible perfectly even distribution of flame completely around the baffle brick. As a result the maximum flame length is greatly reduced.

Interchangeable grills with multiple ports can be varied to suit the combustion characteristics of various gases. The proper sizing of these grills prevents any possibility of flash back.

In addition to the above major improvements the F600 possesses the same desirable features that made the 600 so popular.

1. Simple installation requiring no expensive insulated combustion chamber and having no furnace radiation loss.
2. Extreme quietness due to low rate of combustion over a large area.
3. Flexibility from infinite number of possible combinations varying both size and shape to meet load and firebox conditions at various gas pressures.
4. High radiant transmission rate due to radiant temperature of the standard firebrick baffles on the top of the burner tubes.
5. Low draft loss because of ample secondary air openings.
6. Plain gas pilots of heat resistant material and of a design that will not allow flame to pull off.
7. Safety pilot applied in a cool zone in a manner that insures perfect direct ignition of the burner yet allowing the the thermal element to cool quickly upon flame failure.
8. Guaranteed vibrationless under all conditions.

CAPACITY OF SINGLE F600 VENTURI TUBE—No. 16 M. T. D. ORIFICE

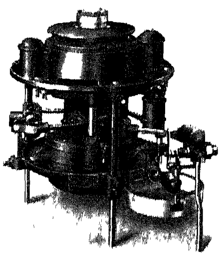
Manifold Gas Pressure	0.5″ W.C.	1.0″ W.C.	2.0″ W.C.	3.0″ W.C.	4.0″ W.C.	5.0″ W.C.	6.0″ W.C.	4 oz	6 oz	8 oz
Input—Cu Ft, 1 hr...	26.5	40.5	60.5	75.0	87.0	98.0	108.0	116.0	143.5	166.5
Output—Sq Ft, St. Rad	81	124	185	229	267	301	331	355	440	510
Output—Boiler H.P...	0.58	0.89	1.33	1.65	1.91	2.15	2.37	2.54	3.14	3.66

REG. U. S. PAT. OFF.

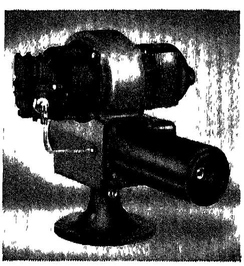
Automatic Burner Corp.

New York, N. Y. **Chicago, Ill.**

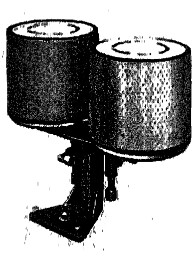
Domestic Pressure and Rotary Oil Burners
Range Burners **Boiler Units**
Furnace Units **Water Heaters**



20 Years of Manufacturing Rotary Type Oil Burners



Highly Efficient, Quiet Operating Pressure Burners



Correctly designed ABC All Steel Range Burner Units

ABC Oil Burners have been manufactured by the AUTOMATIC BURNER CORP. since 1920 and well over 100,000 are in use. There are five sizes of rotary type burners and seven sizes of the pressure type burners. All models are listed as standard by the Underwriters' Laboratories.

SPECIFICATIONS

Pressure Burners / Rotary Burners

Model	Max. Cap. Gal Oil per Hour	Max. Capacity Sq Ft Net Standing		Model	Gal Oil per Hour	Max. Capacity Sq Ft Net Standing	
		Steam	Water			Steam	Water
W2	3.5	900	1440	H1	2.0	500	800
P7#2	1.3	300	480	E3	3.0	800	1280
P7#3	2.5	600	960	E4	7.5	2000	3200
P7#4	4.0	1000	1600	E5	15.0	4300	6880
P7#6	6.0	1600	2560				
P11	11.0	3200	5120				
P27	27.0	8000	12,800				

ABC Oil Burning Boiler Burner Units

Boiler Size	15	16	18	23	26
Rating SHBI Steam	325	420	500	675	1150
Rating SHBI Water	520	675	800	1080	1840
Overall—Height—Inches	49	49	49	55	58
Overall—Width—Inches	22	22	22	23	27
Overall—Length—Inches	37	37	37	38½	46
Approx. Shipping Weight—Lb	750	850	950	1100	1500

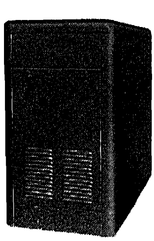
ABC Oil Burning Air Conditioning Units

Model Number	Btu per Hour Output at Registers	Approx. Shipping Weight—Lb
100D	80—100,000	790
150D	100—150,000	975
200D	150—200,000	1050

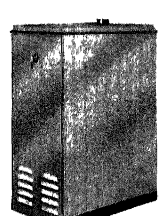
ABC Oil Burning Domestic Water Heaters

Heater Number	65WH	125WH	175WH	300WH	500WH
Capacity—Gal per Hour—90° Rise	65	125	175	300	500
Water Storage Capacity—Gal	18	27	30	36	42
Max. Aux. Water Storage Tank Capacity—Gal	300	400	600	1200	1800
Oil Firing Rates Recommended—gph	.8	1.0	1.3	2.0	3.0
Height—Overall—Inches	49	61	61	61	71

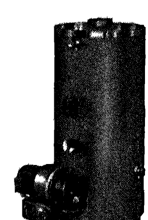
Write for Descriptive Literature



ABC MATCHED Air Conditioning and Oil Burner Unit



ABC MATCHED Boiler and Oil Burner Unit



ABC Matched Domestic Hot-Water Heater and Oil Burner Unit

The Brownell Company

ESTABLISHED 1855

Dayton, Ohio

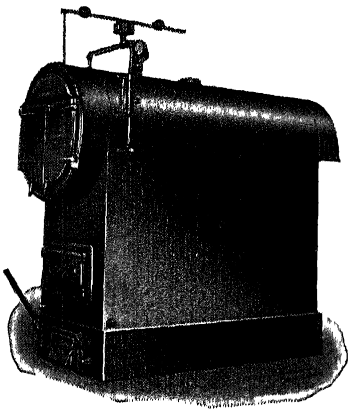
Manufacturers of

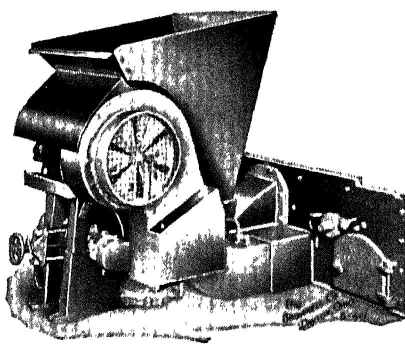
BROWNELL BOILERS AND STOKERS

Representatives in All Principal Cities

FIRE TUBE BOILERS of various types. HEATING BOILERS riveted and welded. UNDERFEED STOKERS from 5 Horse Power upwards STEEL STACKS, TANKS AND SPECIAL PLATE WORK.

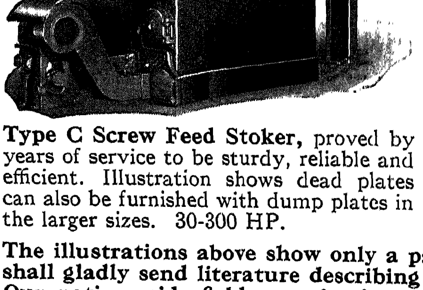
Welded Triple Pass Heating Boilers built in either high leg or low water line types. Hand fired ratings 500 to 35,500 sq. ft. steam, 800 to 56,800 sq. ft. water radiation. Stoker, Oil or Gas fired up to 43,100 sq. ft. steam or 69,000 sq. ft. water radiation. A.S.M.E. Code construction.



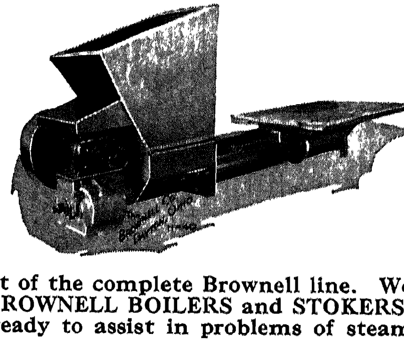


Type L R (Low Set) Underfeed Ram Type Stoker with automatic air volume control. Can be furnished with Brownell exclusive, fully automatic coal feed control. Sizes up to 300 horse power. An ideal stoker for firebox boiler or other installations where height of setting is limited.

High or Low Pressure Double Pass Boiler with Type L R Stoker. Designed and manufactured as a matched unit steam generating plant. Furnished in working pressures from 15 to 150 pounds and sizes up to 300 horse power. For power, heating and process steam. Steam ratings 3,600 to 42,500 sq. ft. Water rating 5,800 to 68,000 sq. ft. when used with stoker, oil or gas. A.S.M.E. Code construction.



Type C Screw Feed Stoker, proved by years of service to be sturdy, reliable and efficient. Illustration shows dead plates can also be furnished with dump plates in the larger sizes. 30-300 HP.



The illustrations above show only a part of the complete Brownell line. We shall gladly send literature describing BROWNELL BOILERS and STOKERS. Our nation wide field organization is ready to assist in problems of steam generation.

Combustion Engineering Company, Inc.

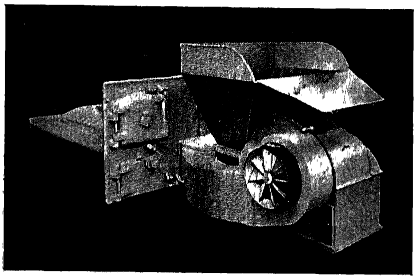
All Types of Fire Tube and Water Tube Boilers
Mechanical Stokers

Complete Steam Generating Units
Pulverized Fuel Systems

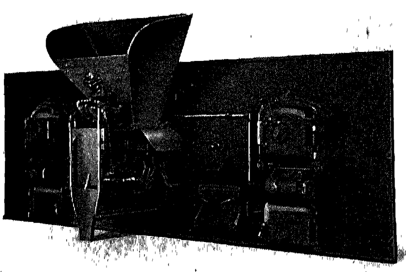
200 Madison Avenue, **New York, N. Y.**

Offices in all principal cities of the United States and Canada

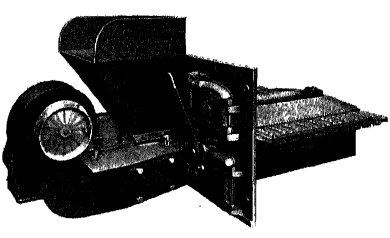
More than 16,000 C-E Stokers installed to date



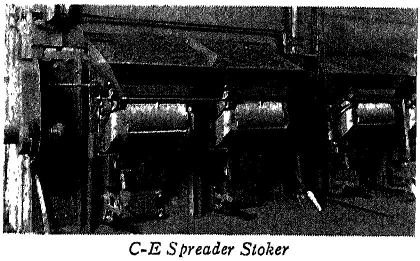
C-E Skelly Stoker Unit



Type E Stoker



C-E Low Ram Stoker



C-E Spreader Stoker

C-E Skelly Stoker Unit—A compact, self-contained unit with integral forced-draft fan and adapted to burn either anthracite or bituminous coal. Alternate arrangement of fixed and moving grate bars assures lateral distribution of fuel. Automatic control is standard equipment. Approximate application range—20 to 200 rated boiler hp.

Type E Stoker—A single-retort, underfeed stoker with an established reputation of many years' standing for dependable service. Designed to burn a variety of bituminous coals under boilers up to about 600 rated hp. Available with steam, electric or hydraulic drive.

C-E Low Ram Stoker—A single-retort, stationary-grate underfeed stoker for burning bituminous coals under boilers in the upper size range of the C-E Skelly Stoker Unit.

C-E Spreader Stoker—A simple, rugged overfeed stoker designed to burn a wide variety of coals. Fines are burned in suspension and the coarser coal on a grate which may be of either stationary or dumping type. Rate of coal feed and air supply may be regulated over a wide range and are readily adaptable to automatic control. Applicable to boilers from about 100 boiler hp up.

C-E Multiple Retort Stoker—For burning bituminous and semi-bituminous coals under boilers up to the largest sizes.

C-E Traveling Grate Stokers—Including both Coxe and Green types. Available with grate surfaces suitable for anthracite, coke breeze, lignite or bituminous coal, as required. Chain grate types are built for either forced- or natural-draft application.

C-E Boilers—All fire tube and water tube types in sizes ranging from 25 hp up to the largest. Standard and special designs to suit all conditions of fuel, load and space. Included are all types formerly known by the trade names "Heine," "Walsh & Weidner," "Casey-Hedges," "Ladd" and "Nuway".

Separate Catalogs describing each of these stokers are available. A-531

Detroit Stoker Company

Sales and Engineering Offices
General Motors Bldg., Detroit, Mich.

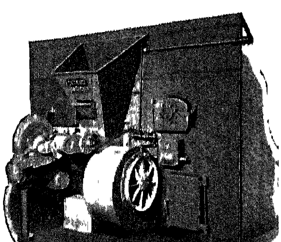
Main Offices and Works at
Monroe, Mich.

Since 1898

District Offices in
Principal Cities

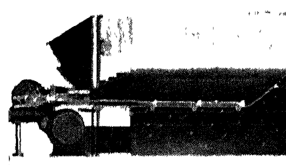
Built in
Canada at London, Ont.

Detroit Stokers are unsurpassed for economy and dependability. The complete line includes both Underfeed and Overfeed Stokers of many sizes and capacities for all types of boilers, 30 Horse Power and upwards. All grades of Bituminous Coal are burned successfully. Operating costs are low. Detroit Stokers are of substantial heavy duty design, representing over 40 years' experience in Stoker manufacture exclusively. Materials of the highest quality are used. All Detroit Stokers erected and tested at the works and installed under the direction of experienced erection superintendents. Catalogs of various types will be furnished on request. Write Detroit Stoker Company, Detroit, Michigan, or district offices in principal cities.

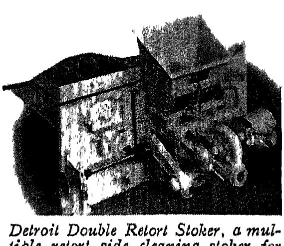


Detroit UniStoker with Detroit Adjustable Feed provides a wide range of coal feed control.

Detroit UniStoker with Detroit Adjustable Feed (Coal Feed Control) insures accurate fuel and air supply for best economy. Single Retort, Side Cleaning, for boilers approximately 125 to 250 horsepower.

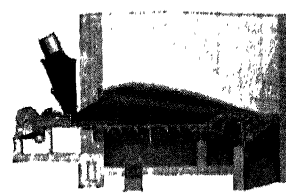


Each Detroit UniStoker with its motor or steam-turbine driven full-housed blower is an independent unit, self-contained.

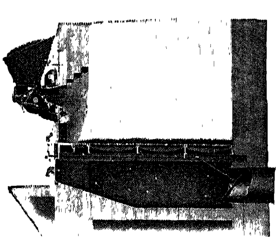


Detroit Double Retort Stoker, a multiple retort side cleaning stoker for medium size boilers.

Detroit Double Retort Stoker, a Multiple Retort Stoker having two retorts with the side cleaning feature. For medium sized boilers having wide furnaces. Used to advantage where limited space conditions prevent the use of the rear cleaning Multiple Retort Stoker.

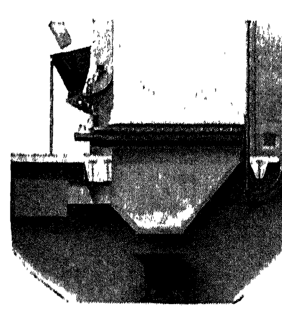


Detroit Multiple Retort Stoker for large boilers and high capacities. An inclined fuel bed Stoker, possessing all outstanding modern features



Detroit RotoStoker, (Stationary Grate Type). Ash removed through doors at grate level successfully burns a wide range of fuels.

Detroit Roto-Stokers are Overfeed Spreader Type Stokers, having an *Overthrow Rotor* action, which insures uniform fuel distribution over the entire area. Offers advantages over other firing methods for burning inferior fuels and efficiently handling extremely fluctuating loads.



Detroit RotoStoker (Dumping Grate Type) (Either Power or Hand Operated) for large boilers. Particularly suited to fluctuating loads.

DETROIT LOSTOKER

Detroit LoStoker is a complete mechanical firing unit in many grate area sizes and capacities for application to all types of boilers from approximately 30 to 150 hp. Burns various grades of Bituminous Coal with high efficiency. Fuel is fed only when needed—none wasted. Single Retort, Side Cleaning, Adjustable Plunger Feed Type, mechanically driven from electric motor, requires little power for operation. Automatically controlled from steam pressure water temperature room thermostat, compact, easily installed, responsive and automatic. A great coal saver.

DETROIT LOSTOKER ADVANTAGES:

Continuous Adjustable Plunger Feed with control of the quantity of coal fed and its distribution.

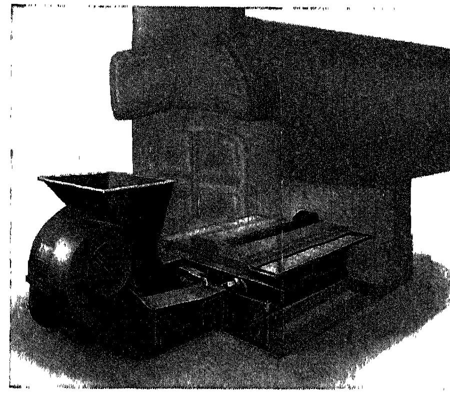
Heavy Mechanical Drive of simple design, requires little power.

Side Cleaning with dumping grates, ashes removed through doors provided in the Stoker front. No hand cleaning.

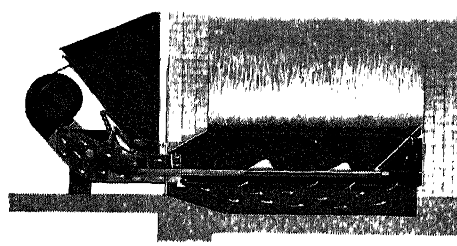
Agitator in coal hopper for continuous coal feed, cannot stick or jam with wet coal.

Automatically Controlled. Motor or steam turbine driven, controlled from steam pressure, water temperature or thermostat.

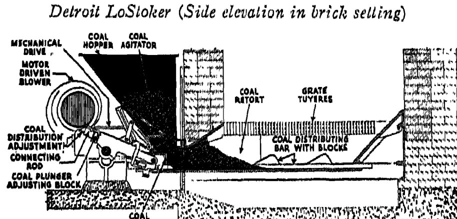
Many grate area sizes and capacities to fit the furnace and provide the proper grate area to readily handle heavy loads and also to operate efficiently under light load conditions.



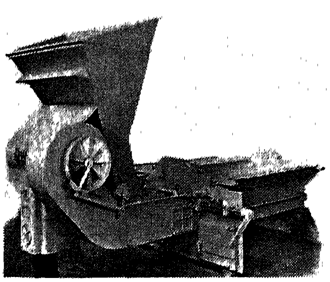
Detroit LoStoker readily applied to Firebox Boilers—built to fit the Furnace or Firebox. Coal Hopper with Agitator designed to clear Boiler Doors. Plunger Feed-side cleaning feature eliminates arduous hand cleaning of fires and corresponding losses.



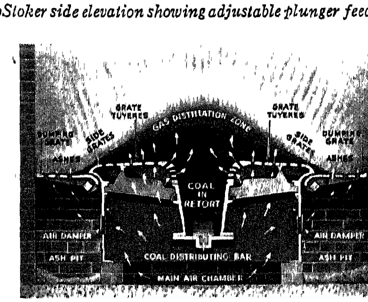
Detroit LoStoker (Side elevation in brick setting)



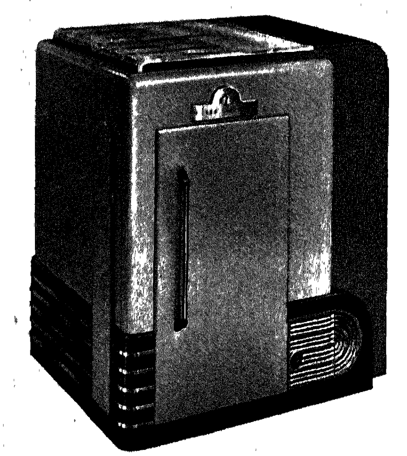
Detroit LoStoker side elevation showing adjustable plunger feed.



Detroit LoStoker with hydraulic drive may be either brickset or firebox type as above. Drive has many advanced features, that provide a wide range of coal feeds and requires little power for operation.

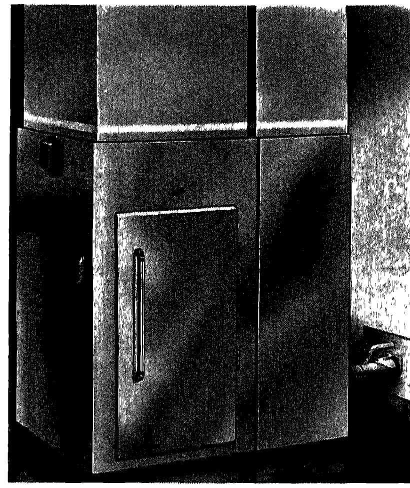


Front Elevation of Detroit LoStoker (brickset type) built to fit the furnace. For use with horizontal tubular, firebox boilers on brick foundations or water tube boilers. Arrows indicate flow of air to all parts of the fuel bed.



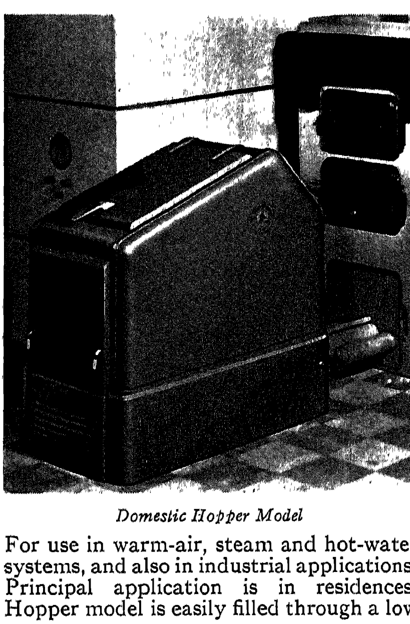
Unit Heatmaker Room Furnace

The Unit Heatmaker is a combination automatic coal stoker, room-furnace, humidifier, and forced warm-air circulator. It is made in two sizes; each size available for either bituminous or anthracite coal. This unit is particularly adaptable for small commercial and industrial heating. Forced circulation improves the distribution of heat. The Iron Fireman Unit Heatmaker is clean and quiet in operation.



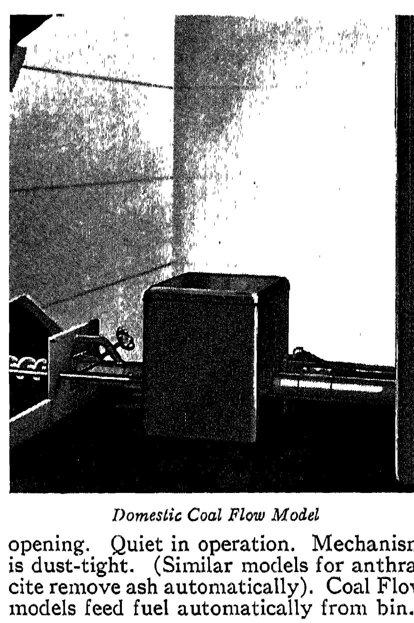
Self-Firing Winter Air-conditioner

This Iron Fireman innovation produces superior winter air conditioning at extra low first cost and comparably low operating cost . . . together with the convenience of coal feeding automatically from the bin. This unit contains stoker, steel furnace, humidifier, filters and blower. May be easily adapted to existing conditions and requires a minimum amount of space. Bituminous and anthracite models.



Domestic Hopper Model

For use in warm-air, steam and hot-water systems, and also in industrial applications. Principal application is in residences. Hopper model is easily filled through a low opening. Quiet in operation. Mechanism is dust-tight. (Similar models for anthracite remove ash automatically). Coal Flow models feed fuel automatically from bin.



Domestic Coal Flow Model

Motorstokor Division
Hershey Machine & Foundry Co.

Factory and Home Office

Manheim, Pa.

MOTOR STOKOR

Installation and Service by factory-trained dealers in all anthracite burning areas.

DEFINITION

A complete stoker, burner, and optional ash-removal system for automatic combustion of buckwheat or rice anthracite. Applicable to coal, gas or oil furnaces or boilers for providing warm air, hot water, or steam. Especially designed for automatically heating buildings and providing year-round hot water.

RANGE OF TYPES

Standard installations in all sizes include direct-from-bin feed with ash removal, direct-from-bin feed with pit collection, hopper feed with ash removal, and hopper feed with pit collection of ashes.

RANGE OF SIZES

Available in 12 models, providing a complete range. The smallest is the new domestic MOTORSTOKOR No. 10 capable of heating average small homes. It feeds up to 40 lb of anthracite per hour, is rated at 1200 sq ft of steam and 1920 sq ft of water radiation. The largest are MOTORSTOKORS No. 2 and No. 3, feeding up to 100 lb of coal per hour and rated at 2850 sq ft of steam or 4560 sq ft of water radiation.

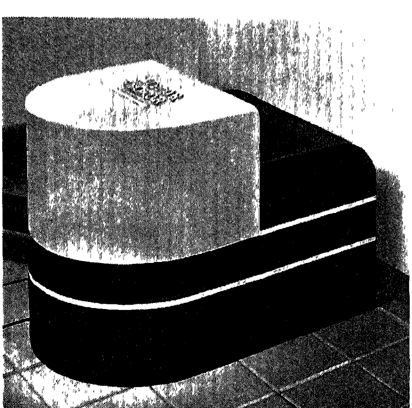
ADVANTAGES

Simplicity: Entire mechanism functions intermittently, including coal feed, controlled draft, and ash removal. A portion of the combustion air is fed with the coal, preventing dust and back draft. Worm feed. Concentric dustless ring burner needs no cleaning. Revolving bar breaks all clinkers.

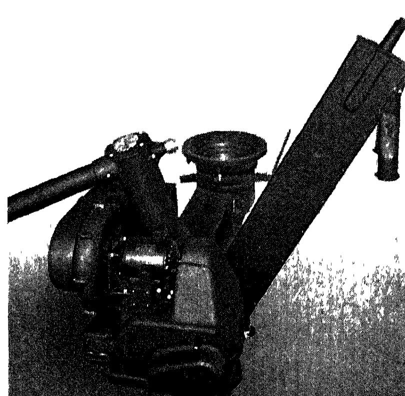
Efficiency: Fixed air mixture for uniform combustion. Floating worm assures uniform, trouble-free coal feed. Flexibly mounted motor operating intermittently and using little current. Quiet, direct-mounted, self-compensating fan.

Ruggedness: Heavy cast parts, with lavish use of chrome-moly, monel, nickel, and special alloys. Oil-submerged gears reduce all operations to very slow wear-free motions. Extraordinary structural and metallurgical protections against corrosion.

Safety: Floating coal worm minimizes stoppage or jams. Air feed through coal prevents back draft and escaping gas. Automatic release-clutch cuts off current when over-loaded. Minneapolis-Honeywell controls.



The new MOTORSTOKOR No. 10 marks its manufacturer's new low in the first cost of completely automatic anthracite equipment.



MOTORSTOKOR 2AF for heavy-duty service in apartments, office buildings, etc. Bin-feed pipe at left. Ash removal system at right.

Buffalo Pumps, Inc.

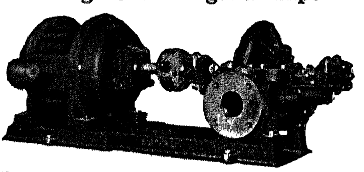
450 Broadway, **Buffalo, N. Y.**

Branch Offices

Albany, N. Y., 1305 Standard Bldg., H. S. Johnson
Atlanta, Ga., 305 Techwood Drive
Baltimore, Md., 508 St. Paul St., E. E. Thompson
Boston, Mass., 486 Main St., Melrose Station, E. D. Johnson
Chicago, Ill., 20 N. Wacker Drive, L. D. Emmert
Cincinnati, Ohio, Building Industries Bldg., F. W. Twombly
Cleveland, Ohio, 418 Rockefeller Bldg., T. A. Weager
Dallas, Texas, 1801 Tower Petroleum Bldg., T. H. Anspacher
Davenport Iowa, 305 Security Bldg., D. C. Murphy Co., Inc.
Denver, Colo., 1718 California St., Stearns Roger Mfg. Co.
Des Moines, Iowa, 214 Old Colony Bldg., D. C. Murphy Co., Inc.
Detroit, Mich., 2051 W. Lafayette Blvd., Coon-De Visser Co., T. E. Coon
Greenville, S. C., 21 Blue Bldg.
Kansas City, Mo., 424 Dwight Bldg., A. E. Williams
Los Angeles, Calif., 708 Pershing Sq. Bldg., P. R. Adrianse
Minneapolis, Minn., 2102 Foshay Tower, E. F. Bell
New Orleans, La., Devlin Bros., 1003 Maritime Bldg.
New York, N. Y., 39 Cortlandt St., W. S. Koithan
Philadelphia, Pa., 703 Cunard Bldg., Davidson & Hunger
Pittsburgh, Pa., Power Equipment Co..........Oliver Bldg.
Richmond, Va., Williamson & Wilmer, Inc.....Mutual Bldg.
Seattle, Wash., 500 First Ave., So., A. T. Forsyth
St. Louis, Mo., 1598 Arcade Bldg., J. W. Cooper
Toledo, Ohio, 1922 Linwood Ave., C. M. Eyster
Washington, D. C. 640 Woodward Bldg., G. S. Frankel
Complete line manufactured in Canada by Canada Pumps, Ltd., Kitchener, Ont.

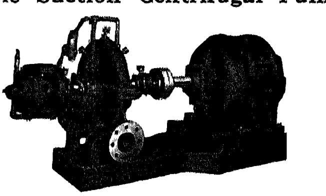
PRODUCTS—A complete line of Single and Multi-stage Centrifugal Pumps and Special Pumps for use in all types of heating and air conditioning installations.

Buffalo Double Suction Single Stage Centrifugal Pumps



For general service where clear water is handled you will get top performance with these pumps. They embody all of the accepted modern features of centrifugal pump design. Capacities range from 10 to 50 thousand U.S. gallons per minute.

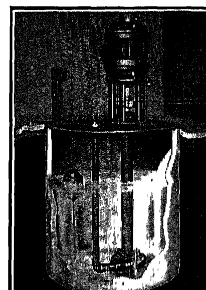
Buffalo Self-Priming Single and Double Suction Centrifugal Pumps



Now available with positive self-priming device built with the pump. This primer is built under license from the Nash Engineering Company and is fully covered by patent.

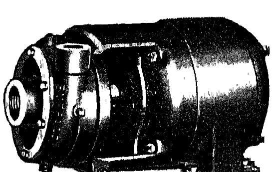
Buffalo Self-Priming Pumps offer these advantages: (1) All working parts are above the liquid to be pumped. (2) There is complete access to all parts of installation. (3) Rotors are balanced—vibrationless. (4) Buffalo Self-Priming Pumps are very quiet—no long shafts to vibrate and fewer bearings. (5) Constant positive prime obtained without foot valves.

Buffalo Automatic Sump Pumps



Buffalo Sump Pumps are self-contained and have unusually high efficiencies thus permitting the use of small motors. Ball bearing thrust and enclosed shaft especially adapt these pumps for their service.

Buffalo Single Suction Closed-Coupled Pumps



This pump is close-coupled to electric motor, eliminating the necessity for bearings. The impeller is overhung on the motor shaft, providing a compact, easily-serviced unit. Permanent alignment is assured and the pump mounted in this manner requires very little space.

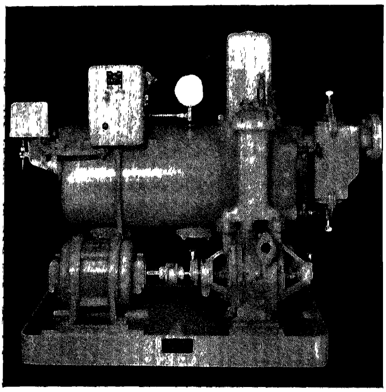
Buffalo Close-Coupled Pumps are suitable for handling hot water with low submergence on suction, or for operating with suction lift as high as 25 ft.

These pumps are also available in special alloys.

The Nash Engineering Company

234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

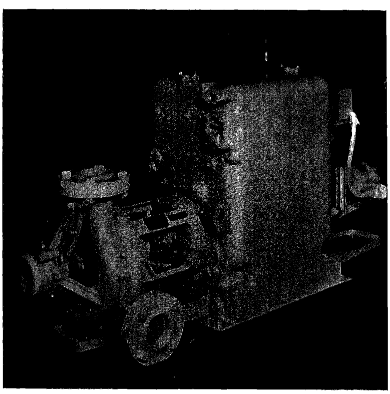


Return Line Vacuum Heating Pump

Standard with the heating industry for over seventeen years. Removes air and condensation from return lines of vacuum steam heating systems, discharging air to atmosphere and returning water to the boiler.

Two independent units are combined in a single casing—an air unit and a water unit. Impellers of both are mounted on the same shaft. Pump is bronze fitted throughout.

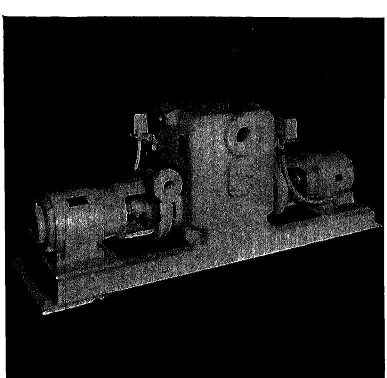
Supplied direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation. Standard in capacities up to 300,000 sq ft E.D.R. Larger units special. Bulletins Nos. 307, 308, 309, and 310 on request.



Vapor Turbine Vacuum Heating Pump

Jennings Vapor Turbine Heating Pumps combine all advantages of the standard return line heating pump with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss.

This pump affords the safety and economy which goes with continuous condensation return and steady vacuum, and at no cost for electric current. Furnished standard in capacities up to 65,000 sq ft E.D.R. Larger units special. Bulletin No. 290 on request.



Condensation Pump and Receiver

Removes the condensation from radiators in return line steam heating systems, particularly radiators set below the boiler water line level, and pumps the condensation back to the boiler. Pump is bronze fitted with enclosed centrifugal impeller of improved design. By making the pump casing a part of the return tank, and bolting the motor base to the tank, floor space is conserved. The rectangular construction permits installation in a corner against the wall.

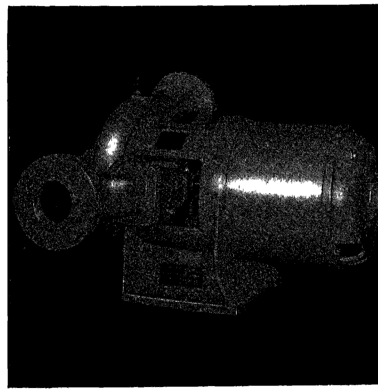
These pumps are furnished in standard sizes with capacities ranging from 1½ to 225 gpm of water. For serving up to 150,000 sq ft of equivalent direct radiation. Bulletin No. 319 on request.

The Nash Engineering Company

234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

Centrifugal Pump



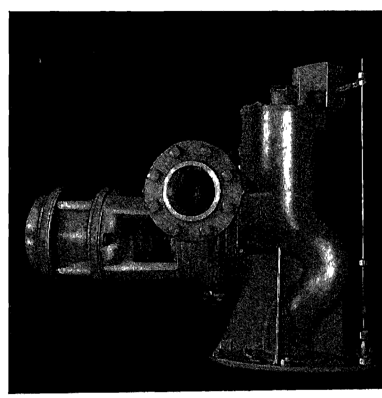
Made in standard and suction (self-priming) types. For circulating hot and cold water; boosting city water pressure; handling water in air washing and conditioning; handling ash sluicing water, etc.

Compact—motor armature and pump impeller are mounted on the same shaft. Simplified—no bearings in pump casing, one stuffing box. Accessible—impeller removable without disturbing piping or shaft alignment.

Self-priming types will handle air or gas continuously with liquid being pumped, and can be operated intermittently without foot valve.

Supplied in 1, 1¼, 1½, 2, 3, 4, 6, and 8 in. sizes, with capacity up to 2000 gpm. Heads up to 300 ft. Bulletin No. 322 on request.

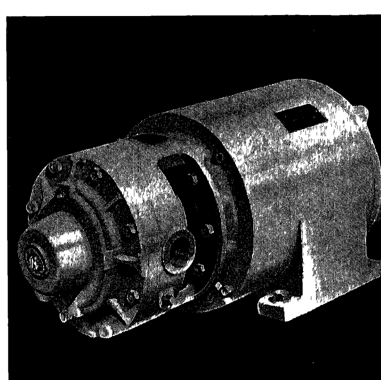
Suction Sump and Sewage Pumps



Jennings Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. Sewage Pumps are equipped with non-clog type impeller for liquids containing solids. Suction piping only is submerged. Centrifugal impeller and vacuum priming rotor are mounted on same shaft that carries rotor of the driving motor, forming a single moving element, rotating without metallic contact.

Will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit, affording perfect accessibility for inspection or cleaning. Capacities to meet all requirements. Bulletins Nos. 159, 161, and 327 on request.

Air Compressor and Vacuum Pump



Nash Air Compressors operate on a unique and different principle. The one moving part rotates in casing without metallic contact. There is nothing to wear, and no internal lubrication.

Nash Compressors deliver absolutely clean air; ideal for agitation of liquids, pressure displacement, and handling gases. Vacuum pumps ideal for priming pumps, blood sucking pumps in hospitals, and wherever non-pulsating vacuum is required.

Pressure 75 lb or vacuum 27 in. of mercury. Furnished for any capacity; special for higher vacuums and pressures. Bulletins Nos. 252, 282, 325 and 331 on request.

Chicago Pump Company

2330 Wolfram Street   **BRUnswick 4110**   **Chicago**

PRODUCTS—Return Line Vacuum Heating and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic and Tankless Water Supply Systems and Automatic Alternator for Duplex Sets of Pumps.

"CONDO-VAC"

Return Line Vacuum Heating and Boiler Feed Pump

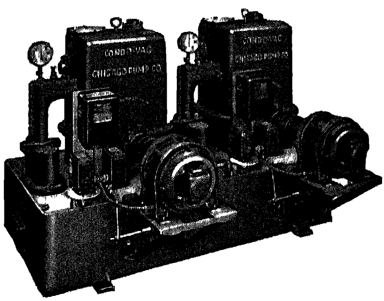


Fig. 2102—Duplex "Condo-Vacs" with Duplex Double Automatic Control

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a "Condo-Vac." "Condo-Vac" reduces corrosion in piping and boiler to minimum—because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. "Condo-Vac" is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain. *Ask for bulletin 270.*

Close-Coupled Pumps

Boiler Feed, Circulating, Tank Filling, Water Supply

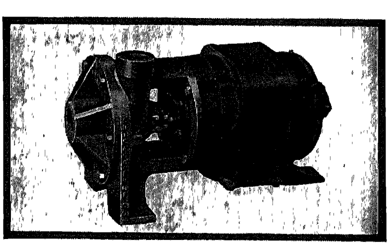


Fig. 2130—Close-Coupled, side suction pump. Capacities range from 3 to 600 Gpm against heads up to 189 ft. Motors from 1/6 to 20 Hp. Discharge 1 to 3 in. Closed and open type impellers. Bulletin 108.

"Sure-Return" Condensation Pump

for Low and Medium Pressure, and Systems up to 35,000 Sq Ft Radiation

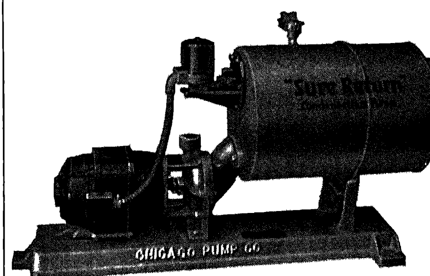


Fig. 1040

"Sure Return" Condensation Pumps and Receivers are built for systems up to 35,000 sq ft of direct radiation and for low and medium pressures. Built in either single or duplex units. Duplex units are alternated in their operation by the Automatic Alternator. *Complete data in Bulletin 250.*

Vertical Condensation Pumps

for Low and Medium Pressure for Systems from 500 to 100,000 Sq Ft Radiation

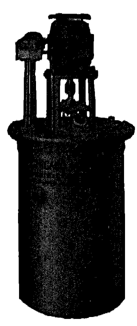


Fig. 1940 Vertical Condensation Pump

The vertical condensation pump is designed to receive returns from lowest radiation. The receiver is placed underground—an ordinary hole sufficing if necessary — and requires very little floor space. Unit is shipped complete, easy to install, assembled so as to prevent steam leaks. Special bearings will stand up under hot water for several years. A special float mechanism is guaranteed not to leak or stick in stuffing box. *Complete data and description in Bulletins 245 and 255.*

ADSCO

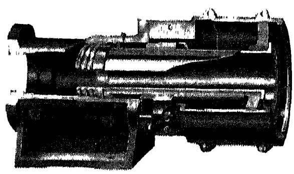
PRODUCTS
for STEAM
SERVICE

AMERICAN DISTRICT STEAM COMPANY

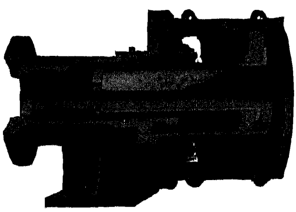
NORTH TONAWANDA, N.Y.

IN BUSINESS OVER SIXTY YEARS

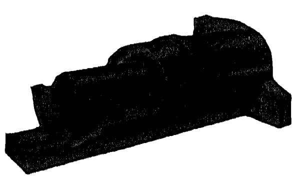
Branches and Agents in Principal Cities



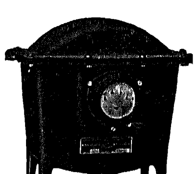
Piston-Ring Type Joint



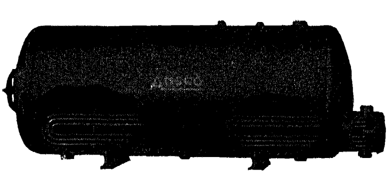
Internally-Externally Guided Joint



Tile Conduit with "Fiberglas" Insulation



Rotary Condensation Meter



Storage Water Heater

PISTON-RING EXPANSION JOINT

Piston rings in the guide ring attached to the inner end of the slip hold the line pressure, enabling the joint to be unpacked and repacked under full operating pressure without interruption to service. A fully guided joint for pressures to 400 lb and temperatures to 750 F. Slip cannot pull out of body. Write for Bulletin No. 35-15G.

INTERNALLY-EXTERNALLY GUIDED SLIP EXPANSION JOINT

A slip type joint with internal guide on inner end of slip and external guide in two-piece, removable hood. Slip cannot pull out of body. Available in semi-steel, cast or wrought steel bodies in sizes from 1¼-20 in. for pressures to 300 lb and temperatures to 750 F. Fully illustrated and described in Bulletin No. 35-20G.

ADSCO-BANNON TILE CONDUIT

Vitrified, tile conduit for underground steam or hot water lines with or without base drain in sizes from 4-24 in. incl. Pipe supports fit reinforcing ribs within the conduit without piercing conduit wall. In combination with "Fiberglas" Filler Insulation, it provides a permanent, efficient installation at reasonable cost. Write for Bulletin No. 35-67G.

ROTARY CONDENSATION METER

Measures steam consumption by metering condensate from heating systems or industrial equipment. Accurate within 1 per cent and factory tested to 150 per cent of rated capacity. Compact, easily cleaned, tamper-proof and equipped with non-fogging counter mechanism. Counter reads directly in pounds. Suitable for vacuum or gravity service. Available in 7 sizes from 250-12,000 lb per hour capacity. Write for Bulletin No. 35-80AG.

ADSCO WATER HEATERS

Made in Storage and Instantaneous Types with copper U-tube or straight tube heating elements in any size or capacity to meet specific operating conditions. Also manufacturers of Heat Economizers, Swimming Pool Heaters, Fuel Oil Heaters and other heat exchanger units. Write for Bulletins 35-75G and 35-76G.

E. B. Badger & Sons Co.

General Office: 75 Pitts Street, **Boston, Mass.**

Representatives

Atlanta, Ga.	140 Edgewood Ave.	London, England	153, Moorgate
Birmingham, Ala.	444 Brown-Marx Bldg.	Los Angeles, Calif.	600 S. Anderson St.
Buffalo, N. Y.	361 Delaware Ave.	Minneapolis, Minn.	732 Builders Exchange
Charlotte, N. C.	1408 Independence Bldg.	Montreal, Quebec	1411 Crescent St.
Chicago, Ill.	1307 S. Michigan Ave.	New Orleans, La.	628 South Peters St.
Cincinnati, Ohio	831 Temple Bar Bldg.	New York, N. Y.	271 Madison Ave.
Cleveland, Ohio	Guardian Bldg.	Philadelphia, Pa.	1500 Walnut St.
Denver, Colo.	725 Denver National Bldg.	Pittsburgh, Pa.	409 Magee Bldg.
Des Moines, Iowa	414 Twelfth St.	Salt Lake City, Utah	Kearns Bldg.
Detroit, Mich.	424 Book Bldg.	San Francisco, Calif.	Sharon Bldg.
Houston, Texas	5646 Navigation Blvd.	Seattle, Wash.	Smith Tower
Indianapolis, Ind.	825 Occidental Bldg.	St. Louis, Mo.	4060 West Pine Blvd.
Kansas City, Mo.	1332 Oak St.	Vancouver, B. C.	1396 Richards St.

ENGINEERS AND MANUFACTURERS

Manufacturers of Copper and Stainless Steel Badger Corrugated Expansion Joints; Engineers and Manufacturers of Chemical Apparatus; Engineers on Process Work; Designers of Complete Plants.

More than forty years' experience in design, manufacture and application are back of BADGER EXPANSION JOINTS. Most recent developments emphasize the constant study Badger engineers are giving to expansion joint development:

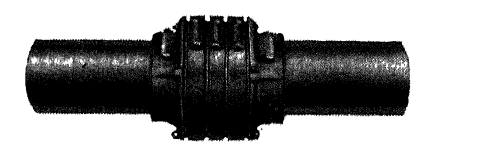
1 . . . **Application of Heat Treatment** . . . scientific heat treatment is applied throughout the fabrication of Badger Expansion Joints with the result that the buyer gets all the benefits of this important metallurgical step.

2 . . . **Directed Flexing** . . . involving a new design corrugation and equalizing ring, resulting in much longer joint life. The all-curve Directed Flexing corrugation distributes flexing stresses which, with straight-sided corrugations, tend to localize.

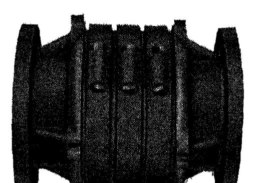
3 . . . **Stainless Steel Joints** . . . perfected after years of study and testing with this useful metal . . . now practicable to use the packless type of joint for high temperatures and high pressure conditions.

The BADGER Expansion Joint is the packless type. Requires no servicing throughout its long life. Ideal particularly for underground use or in cramped quarters. Wide range of traverse.

BADGER Self-Equalizing, Directed Flexing, Expansion Joint



Designed for traverses ranging from fractions to 6 inches single and 12 inches double; for pressures ranging from high vacuum to 200 pounds (copper) and 300 pounds (stainless steel); and for temperatures ranging from sub-zero to 500 F (copper) and 900 F (stainless steel). List prices, installation and other data in **Bulletin 100.**



Welding End and Flanged End, Directed Flexing, Self-Equalizing Expansion Joints.

BADGER Non-Equalizing Expansion Joint

Designed principally for traverses up to ½ inch and for pressures up to 25 pounds; also good as the connecting element between adjacent equipment to absorb vibrations or limited lateral displacements; standard shapes: round, oval, square or rectangular; special shapes to order. **Bulletin No. 200.**

BADGER Flexible Pipe Line Seal

Designed to be used on pipe passing through walls, foundations or bulkheads, the purpose being to allow expansion and contraction but to seal the opening against seepage of ground or other waters.

Bulletin No. 300.

Cochrane Corporation

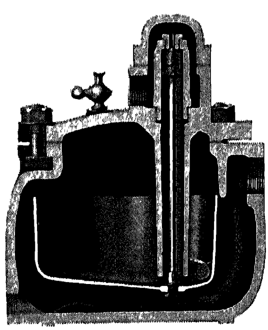
3130 North 17th Street, **Philadelphia, Pa.**

Branch Offices in 40 Principal Cities

COCHRANE HEAVY-DUTY STEAM TRAPS

A high pressure unit for condensate drainage of steam lines, separators, coils, evaporators, etc., and for conditions involving relatively high drainage rates. Recommended for pressures up to 400 lb. Simple construction. No levers, constricted passages or stuffing boxes to become clogged with sediment or scale. All parts are readily accessible. Action is quick and positive, avoiding wire drawing and erosion.

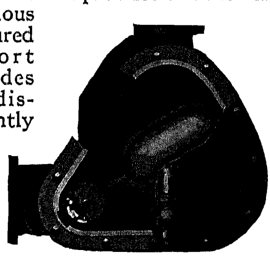
Write for publication No. 2850.



Bucket Trap

MULTIPORT DRAINERS

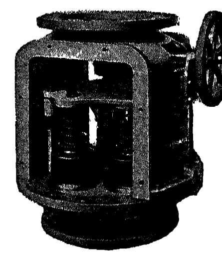
Of the multiport type, they afford unusual capacity for removing condensate or drips from purifiers, separators, jackets, radiators, pressure heating or drying coils, etc. Eliminating condensate delivers maximum heat from steam production at lower cost. Tremendous capacity assured by large port areas. Provides continuous discharge. Instantly responsive. Compact and light in weight. For pressures up to 150 lb.



Multiport Drainer

COCHRANE MULTIPORT RELIEF VALVES

For back pressure, atmospheric relief, flow or check valve service on air, gas, steam or water lines to give positive protection against explosions arising from stuck, jammed or overweighted valves. Differ in the usual construction in that a number of small disks are used instead of one large disk. For full description write for publication No. 2870.

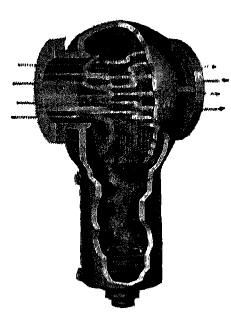


Multiport Back Pressure Valve

ALL-SERVICE SEPARATORS

Steam is rarely or never generated dry or clean and, in that state, corrodes turbine blades, engine and pump cylinders, valves, pistons, etc. Exhaust steam contains oil and entrained solids which should be removed if used for heating purposes or otherwise.

Cochrane Separators purify steam by separating out oil, slugs of water and condensate. Complete removal of entrainment is accomplished by vertical baffle ribs which guide it into a direct unrestricted fall, and a baffle area which extends far beyond the flow from the inlet pipe. Ports at the sides of the baffle prevent the purified steam from passing over the drip area and coming into contact with the entrainment. The steam flow is uninterrupted and pressure loss is minimized.



All Service Separator

For information on other Steam Specialties write for individual publications.

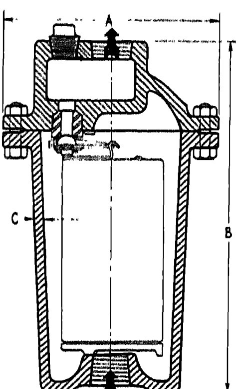
MATERIALS USED IN ARMSTRONG TRAPS

Part No.	Name	Material
1 & 2	Cap and Body	Cast Semi-Steel, 33,000 lb tensile strength
3	*Seat	Chrome Steel
4	Guide Plate Pins	Stainless Steel 18-8
5	*Valve	Chrome Steel
6	Lever	Stainless Steel 18-8
7	Bucket (drawn in one piece for No. 800, 801, 211 and 212)	Stainless Steel
8	Retainer	Stainless Steel 18-8
9	Gasket	Compressed Asbestos 1/32"
10	Bolts and Nuts	Steel 90,000 lb minimum tensile
11	Thermic Vent	Stainless Bi-metal and Stainless Steel
12	Test Plug	Steel

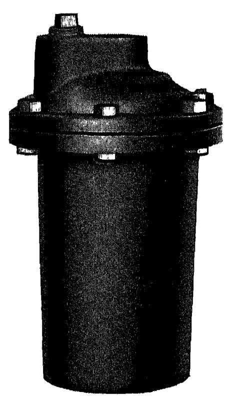
*Valve and Seat are heat treated after machining.

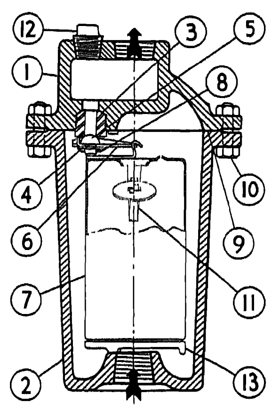
4. The wearing parts in all Armstrong Traps are identical in design, material, and precision workmanship with parts used in Armstrong Forged Steel Traps for pressures up to 1500 lb gage and total temperatures of 850 F.

Armstrong Steam Trap Book. This 36 page book gives complete information on all sizes and types of Armstrong Traps. It also contains 17 pages of data on the subject of trap selection, installation, and maintenance. A free copy will be mailed on request.



No. 211-216, Standard Type





No. 211-216, Blast Type

Traps No. 211-216

Trap Size		No. 211	No. 212	No. 213	No. 214	No. 215	No. 216
Pipe Connections		1/2"	1/2" or 3/4"	1/2" or 3/4"	1"	1" or 1 1/4"	1 1/2" or 2"
List Price (Regular)		$ 9.25	$15.00	$20.75	$29.00	$38.00	$55.00
List Price (Blast Trap)		$10.75	$17.00	$22.75	$31.50	$40.50	$60.00
Telegraph Code (Regular)		Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)		Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height........Dimension B		6 3/8"	8"	10 1/4"	12 1/2"	14"	16 3/4"
Diameter...... " A		4 1/8"	5"	6 3/8"	7 1/2"	8 1/2"	10 5/16"
Wall Thickness " C		3/16"	1/4"	9/32"	3/8"	3/8"	7/16"
Diameter of Bolts		1/4"	1/4"	3/8"	3/8"	1/2"	1/2"
Number of Bolts		6	8	6	8	8	12
Weight		5 1/2 lb	10 1/2 lb	19 lb	32 lb	47 lb	80 lb
Maximum Pressure		250	250	250	250	250	250
Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information, see the Capacity Chart in the Armstrong Steam Trap Book.	Lb Pressure 5	830	1600	2900	4800	7600	14500
	10	950	1900	3500	5800	9000	17300
	15	1060	2100	3900	6500	10000	19200
	20	880	1800	3500	6000	8500	18500
	30	1000	2050	4000	6800	9800	18000
	50	840	1900	4100	6300	9000	18200
	70	950	2200	3800	6000	9200	18300
	100	860	1800	3600	6200	10400	18000
	125	950	2000	3900	6700	10900	20000
	150	810	1500	3500	5700	9500	18500
	200	860	1600	3200	5300	9200	17500
	250	900	1680	3500	5700	7000	19000

GRINNELL COMPANY, INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence, R. I.**

National Distributors of Thermoflex Traps and Heating Specialties

For data on other Grinnell Products, see pages 972-974

Thermoflex Specialties

The heart of all Thermoflex Traps is the Hydron Bellows.

The Hydron Bellows is formed under hydraulic pressure. This powerful internal pressure locates any weakness of any nature in the tubing. Such hydraulic pressure is many times more severe than any pressure the Trap will ever be called upon to control. Every Thermoflex Trap, therefore, is practically indestructible.

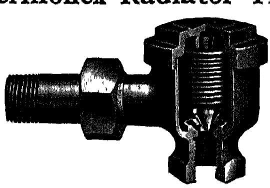
Thermoflex Traps have an exceptionally large orifice. This large orifice combined with high lift, insures fast action and freedom from clogging.

We supply Thermoflex Traps guaranteed for steam pressures of 25 lb, to 50 lb and to 125 lb. Complete information and details of typical installations will be gladly sent on your request. Ask for Catalogue on Thermoflex Heating Specialties.

Valves, Traps, Gauges, Etc.

The Thermoflex line includes: Radiator Traps, Offset Traps, Blast Traps, Drip Traps, High Pressure Traps, Vent Traps, High-grade Packless Inlet Valves, and the Thermoflex Alternator, Thermoflex Compound Gauge, Thermoflex Damper Regulator.

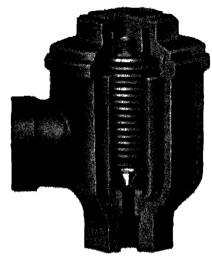
No. 12 Thermoflex Radiator Trap



The full eight-fold Thermoflex-Hydron Bellows is guaranteed because of the Hydron-forming process. Body is heavy bronze construction throughout, with renewable seat.

Fully nickel-plated with highly polished trimmings. The No. 12 is made in angle and in corner patterns, with ½ in. inlet and ½ in. outlet tappings. The inlet neck is double thick to allow for expansion strains. Guaranteed for steam pressures up to 25 lb.

Thermoflex High Pressure Traps



The No. 100A Thermoflex Trap is guaranteed for steam pressures from 50-125 lb. Must not be used where the steam temperature exceeds 400 F.

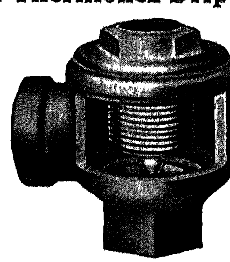
For use with all types of process work, Laundry Machinery, Kitchen Equipment, Hospital Sterilizers, Vulcanizers, Dry Kilns, Unit Heaters, Street Steam Service, etc., in fact any place that a trap is desired for service at the above pressures.

Small, compact and inexpensive.

Extra heavy body. Renewable nickel steel seat and disc. Bellows made from special bronze tubing and encased in brass sleeve to prevent distortion due to pressure.

Regularly furnished without unions, plain nickel finish. Can be furnished with unions, polished nickel or chromium plated at extra cost.

No. 4 Thermoflex Drip Traps



Used for dripping mains, risers, coils and unit heaters. Semi-steel body, bronze cap and inserted renewable bronze seat, angle pattern only, without unions. Can be used for any general purpose where a finished, nickel-plated trap is not necessary, and at a lower cost. Guaranteed for steam pressures up to 25 lb.

Kieley & Mueller, Inc.

Established 1879

Engineering Specialties for Pressure and Flow Control

40 West 13th Street, **New York, N. Y.**

Factory: NEWARK, N. J.

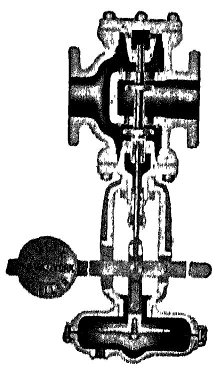
Agents in All Principal Cities

PRODUCTS—Valves: Altitude, Stop and Check, Pressure Regulating, Float, Pilot Reducing, Back Pressure, Tank Control.

Liquid Level Controllers, Water Feeders, Pump Governors, Steam Traps, Y-Type Strainers.

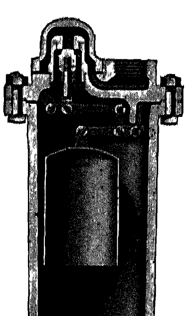
Also Damper Regulators, Hot Water Temperature Controllers, Oil Separators, Steam Separators, Return Traps, Water Columns, etc.

Catalogs sent upon request



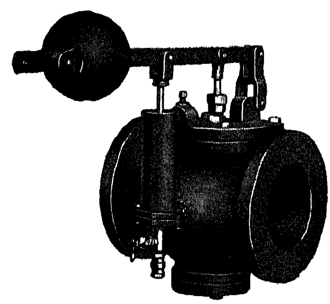
Pressure Regulating Valve

Spring and lever weighted valves for all services and for initial pressures up to 250 lb and reduced pressures from 0 to three-quarters of the initial pressure. Single or double seated in sizes $\frac{3}{8}$ to 16 in. Suitable for steam, water, air, oil and gas. Controlled by a small feeler pipe connected from diaphragm to low pressure side.



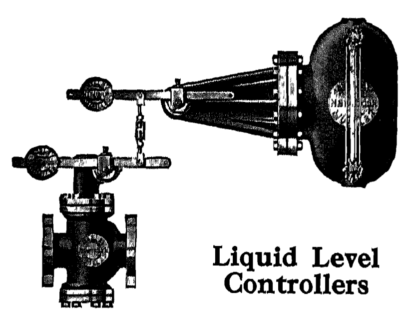
Steam Traps

Large capacity, small sized inverted bucket traps; quick-acting, self-cleaning and non-air binding. Sizes $\frac{3}{8}$ to 2 in. Pressures up to 250 lb. Body and cover, semi - steel. Valve and seat, stainless steel. Removable cap allows inside inspection or replacement of valve parts without disturbing pipe connections. (All parts are interchangeable).



Back Pressure and Atmospheric Relief Valve

For use where plant is operated either condensing or non-condensing. Outside air dash pot insures noiseless operation. Maintains exhaust line back pressure from 0 lb to 25 lb. Made horizontal or vertical lever and weight or spring operated.



Liquid Level Controllers

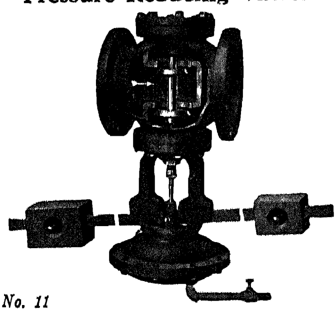
For the accurate control of liquids in tanks or other vessels; suitable for use in industrial plants, gasoline plants, refineries, etc. Direct connected or remote control; ball bearing spindle and easy-to-pack stuffing box; rotary or sliding valve. Write for special bulletin C-3.

Mueller Steam Specialty Co., Inc.

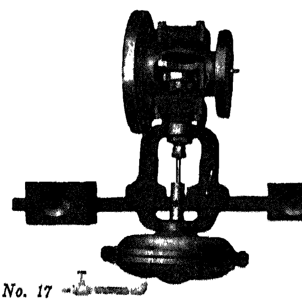
40-20 22nd Street, **Long Island City, N. Y.**

Steam, Water, Air, Oil and Gas Specialties for Heating and Power Plants

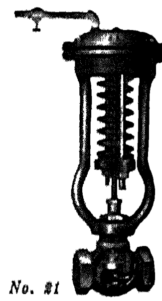
Pressure Reducing Valves—Straight Pattern and With Increased Outlet



No. 11



No. 17



No. 21

No. 11—For Vacuum, Vapor and Low Pressure Heating Systems. Initial Pressures, up to 200 lb; Reduced Pressures, 0 to 10 lb.

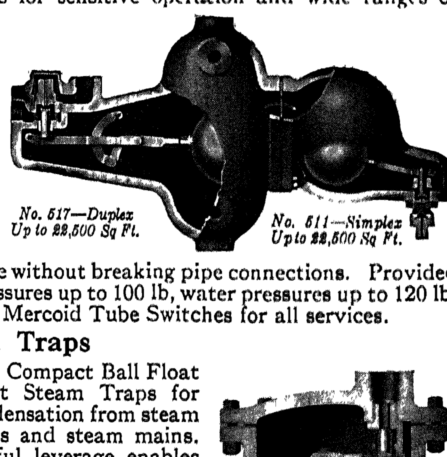
No. 17 and 21—For automatic control of reduced pressures on dead-end service, requiring a tight closing valve, such as tank heaters, kitchen utensils, sterilizing apparatus, laundry equipment, kettles, cookers, driers, etc. Initial Pressures up to 200 lb. Reduced Pressures 0 to 150 lb.

Constructed with full globe bodies. Center guide eliminates the wings on discs, and increases efficiency, assures minimum noise and prolongs the life of the seats and discs. Lever and weight operates on a steel roller bolt, assuring a most sensitive valve. Spring type furnished with special long springs for sensitive operation and wide ranges of reduced pressures.

Automatic Water Feeders

With a powerful leverage to control the water line in steam boilers, etc. They supply make-up water to compensate for evaporation, leaks, steam utilized in process work and condensation wasted. Where condensation held up in the system eventually returns in large quantities, our Duplex type protects the boiler against flooding. All working parts of non-corrosive metal, are accessible without breaking pipe connections. Provided with an integral strainer. For steam pressures up to 100 lb, water pressures up to 120 lb.

Equipped with low water and pressure Mercoid Tube Switches for all services.



No. 517—Duplex Up to 22,500 Sq Ft.

No. 511—Simplex Up to 22,500 Sq Ft.

Steam Traps

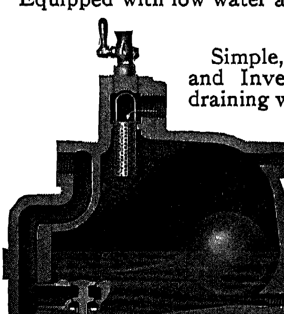
Simple, Sturdy and Compact Ball Float and Inverted Bucket Steam Traps for draining water of condensation from steam apparatus and steam mains.

Powerful leverage enables them to take care of large quantities of condensation.

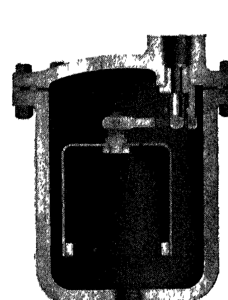
Ball Float Steam Traps equipped with integral strainer; water gages, air cocks, blow-off and integral by-pass valve, when desired.

All working parts are accessible without disturbing any pipes.

Valves are sealed with several inches of water, making the escape of steam impossible.



Ball Float
No. 219—Up to 30 lb.
No. 221—Up to 150 lb.
Sizes ½ to 3 in.



Inverted Bucket
No. 211—For Pressures Up to 250 lb.
Sizes ½ to 2 in.

Catalogue and Bulletins covering our Complete Line gladly furnished on application.

Wright-Austin Co.

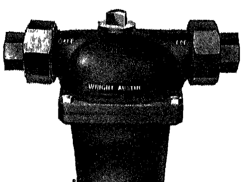
317 West Woodbridge St., **Detroit, Mich.**

PRODUCTS—Steam Traps, Strainers, Air Traps, Steam and Oil Separators, Compressed Air Purifiers, Exhaust Heads, Boiler Feeders and Controllers, Alarm Water Columns, Water Gauges, Trycocks.

"Airxpel" Bucket Type Steam Traps

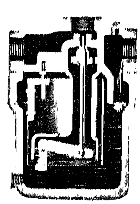
Are "double duty" traps, because they automatically discharge both air and condensate.

Union connections make them easy to connect up. Also, furnished with screw connections when desired. They save money for fittings and installation labor, by having straight through horizontal pipe connections.



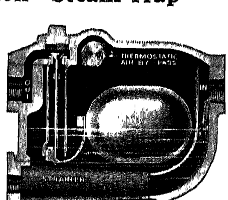
The Cub sizes are made in ½ in., ¾ in., 1 in. Especially suitable for individual unit drainage on heating and process equipment.

Also three "Master" sizes ½ in. to 2 in., for general service.

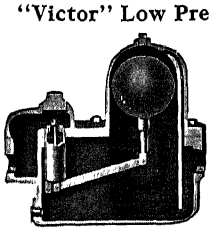


"Combination" Steam Trap

Float Type with internal thermostatic air bypass and strainer for pressures 0 to 40 lb. A modernly designed and very successful trap for vacuum and pressure heating.



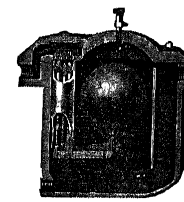
"Victor" Low Pressure Steam Trap

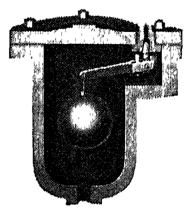


A heavy duty trap for large volumes of condensation at low pressures.

"Emergency" Float Type Steam Trap

Three valve trap with large capacity at high pressures. An exceptionally reliable trap for use in inaccessible places.



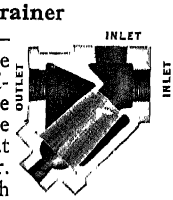


Air Relief Trap

For relieving air from forced circulation hot water heating systems, water supply lines, closed tanks, receivers, pumps, etc.

"Tuway" Strainer

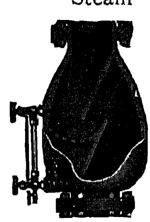
May be used two ways—as a straight-way or angle strainer, in either horizontal or vertical pipe line, because it has the choice of two inlets at right angles to one another.



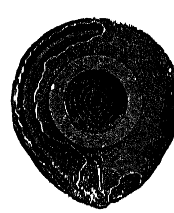
For cleaning, flush through blow-off connection, or remove screen by unscrewing bottom plug.

Separators—Steam and Oil

Type "A" Vertical Steam

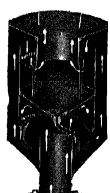


Type "S" Horizontal Oil



We make separators of every type and all sizes for all pressures.

Exhaust Head



Designed to eliminate noise and spray. Three types to select from—the "Cyclone" Heavy Duty, and Standard Galvanized Steel—also, the cast iron type, to remedy all conditions. Sizes 1 in. to 48 in.

Send for descriptive Bulletins on any of the items listed on this page.

Yarnall-Waring Company

Manufacturers of YARWAY **Steam Specialties**

7600 Queen Street, **Philadelphia, Pa.**

YARWAY IMPULSE STEAM TRAPS

Construction—The Yarway Impulse Steam Trap is unique in that there is only one moving part, the simple valve F. This trap is made of bar stock throughout, no castings used. Body and bonnet of cold rolled steel, cadmium plated; cap of tobin bronze, valve and seat of heat treated stainless steel. For pressures 400 to 600 lb, bonnet and cap are stainless steel.

Operation—Movement of the valve is governed by changes in pressure in control chamber (K). When handling ordinary condensate, tiny control flow bypassing through orifice in center of valve reduces chamber pressure below inlet pressure and valve opens, allowing free discharge through seat. As condensate approaches steam temperature, low chamber pressure causes flashing, flow through center orifice is choked and pressure builds up in control chamber closing valve (F).

Advantages

Light Weight—Yarway traps need no support—½ in. trap weighs only 1⅜ lb. 2 in. trap weighs 8⅝ lb.

Small Size—They practically eliminate radiation losses—can be installed in cramped quarters—½ in. trap measures 2¼ in. long—2 in. trap, 4¾ in. long.

Will not air bind.

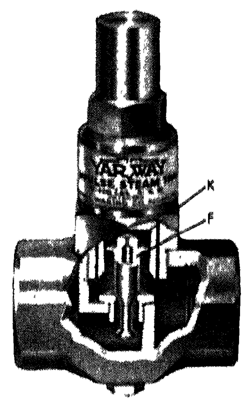
Require no priming.

Insure quick heating.

Operate on exclusive Impulse principle (U. S. Patents No. 2,051,732 and 2,127,649.)

Low Price—Often cheaper than repairing old traps.

Factory set to operate at all pressures up to 400 lb (or 600 lb) without change of valve seat.



List Prices, Weights and Dimensions

No. 60 Series—up to 400 lbs. and

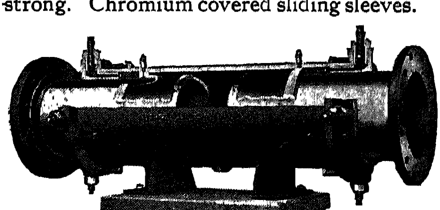
No. 70 Series—up to 600 lbs.

Size	Trap Complete	Weight Pounds	Length Inches
½″ Nos. 60 or 70	$15.00	1¼	2⅞
¾″ Nos. 61 or 71	22.00	2	3
1″ Nos. 63 or 73	31.00	2½	3⅜
1¼″ Nos. 64 or 74	48.00	4	3¾
1½″ Nos. 66 or 76	68.00	5¾	4¼
2″ Nos. 67 or 77	90.00	8½	4¾

For further information send for descriptive bulletin T-1735.

YARWAY GUN-PAKT EXPANSION JOINTS

All-steel welded construction; light but strong. Chromium covered sliding sleeves.

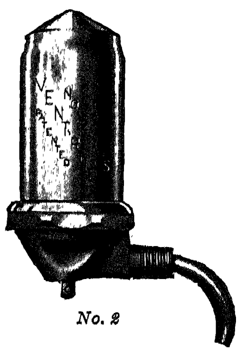


Cylinder guide and stuffing box integral, assuring perfect alignment. Internal limit stops. Gun-pakt and Gland-pakt types; Gun-pakt (illustrated) fitted with screw guns which permit insertion of plastic packing while joint is under pressure. Sizes 2 in. to 24 in., single end or double end, flanged or welding ends; 150, 300 and 400 lb pressures. For additional details send for bulletin EJ-1907.

Anderson Products, Incorporated

Cambridge, Massachusetts

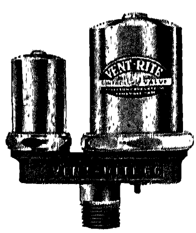
Vent-Rite Controlled Venting Radiator Valves . . . Vent-Rite No. 66 Control Valves . . . Vent-Rite Balancer . . . Vent-Rite Unit Heater Valve. Originators of "Balanced Radiation by Controlled Venting," "The Vent-Vac Method" and "Vacuum Limitation."



No. 2

VENT-RITE RADIATOR VALVES

Vent-Rite Controlled Venting Radiator Valves are made in a wide variety of types, sizes, outlets, and venting capacities. Both Vacuum and Non-Vacuum. All are noiseless in operation, positive in action, close thermostatically under temperature. They may be taken apart for examination and cleaning. Venting is through an adequate straight-line venting orifice, accurately set by a modulating adjustment of the valve pin. The adjustment, being underneath, is not readily disturbed.



No. 66

VENT-RITE CONTROL VALVES

Vent-Rite Control Valve No. 66 is the heart of the Vent-Vac Method of steam control for automatically-fired, one-pipe systems. It takes the place of a main line vent, limits the amount of vacuum created (see below) and breaks the vacuum at the beginning of the firing period. It is entirely mechanical. With the No. 66 and the Vent-Vac Method, a system is "Vacuum" between Firing periods, "Non-Vacuum" during Firing, combining the best of both systems, assuring "Balanced Radiation."

The Vent-Rite Line includes Nos. 1, 51, 3, 5, 5A and 55 (Non-Vacuum); 2, 62, 4, 6, 6A, 66, 68 and the Balancer (Vacuum).

VACUUM LIMITATION

Vacuum has always been right, but there are two kinds of vacuum

In the days of hand-tended coal furnaces, vacuum proved its value in making use of the latent heat in boilers, by reducing the boiling point of the boiler water as vacuum increased. Vacuum can still be used effectively for that purpose, but the Vacuum itself must be limited, as too great a Vacuum tends to inefficiency. Vacuum at 5 in. and Vacuum at 25 in. are very different in characteristics.

At the higher Vacuum, one pound of steam is 344 per cent bigger in volume, and therefore the Btu content of each cubic foot has been reduced 76 per cent. The capacity of the system cannot be increased, and as the heat value of each cubic foot has been so reduced, it is not economical to transfer this steam vapor in the higher Vacuum. Therefore Vacuum Limitation to a degree that will produce the transfer of a satisfactory number of Btu's per cubic foot is desirable.

Vent-Rite units operating under the Vent-Vac Method offer proper Vacuum Limitation.

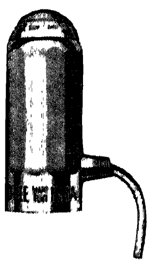
The Dole Valve Company

Main Offices and Factory: 1901-1941 Carroll Avenue, **Chicago, Ill.**

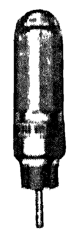
Branch Offices and Representatives **In all Principal Cities**

Dole No. 1A Vari-Vent Valve is designed to be part of a room-thermostat-controlled oil, gas or stoker fired gravity steam heating system. Vents air extra fast. Permits comfortable heating with lower and more economical boiler pressures. On systems using this valve throughout, all radiators are heated quickly with only ounces of pressure rather than the usual pounds.

Also BALANCES distribution of heat to radiators; gives definite straight-line control any venting speed between virtually closed and wide open. Large and distant radiators heat as rapidly as those close to boiler. Tamper-proof — can be set and locked. Easily installed on any cast iron radiator. Finished in bright chromium finish.

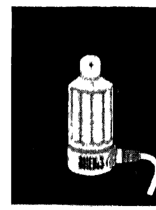


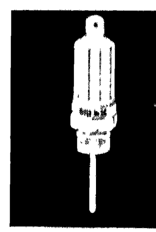
No. 1A Vari-Vent Valve

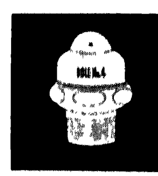


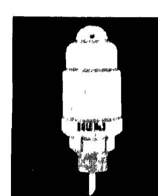
No. 1B Vari-Vent Air Valve

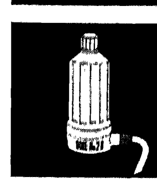
No. 1C Quick Vent Float Valve











No. 1B Vari-Vent Air Valve on each convector for BALANCING automatically fired gravity steam convector installations.

No. 1C Quick Vent Float Valve for mains of automatically fired gravity steam installations.

Valves at Right (top to bottom)

No. 3 Air Valve for all types of gravity steam installations operated at any pressure up to 15 lb. Positive seal against water.

No. 3C Air Valve, all purpose straight type. Particularly recommended for unit heaters.

No. 4 Quick Vent Valve, for mains ending 18 in. or more above boiler water line.

No. 5 Quick Vent Valve. Positive seal against water.

No. 2B Vacuum Valve for vacuumizing gravity steam installations.

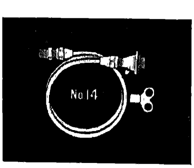
Valves Below (left to right)

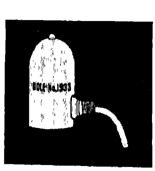
No. 14 Key Valve for venting concealed radiators and convectors of hot water heating systems.

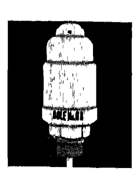
No. 1933 Air Valve. Low cost, high quality and performance.

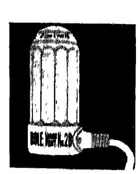
No. 6B Vacuum Valve for quick venting main of vacuumized systems.

No. 2B Vari-Vent Vacuum Valve for BALANCING vacuumized gravity steam installations.









Write the Dole Valve Company for a handy Selector Chart which indicates the Dole Air or Vacuum Valve most suited for a particular need.

Jenkins Bros.

BRONZE - IRON - STEEL VALVES

Mechanical Rubber Goods

80 WHITE ST., NEW YORK, N. Y.; 524 ATLANTIC ST., BOSTON, MASS.; 376 SPRING ST., N. W., ATLANTA, GA.; 133 N. SEVENTH ST., PHILADELPHIA, PA.; 1514 FULTON ST., CHICAGO, ILL.; 1112 WALNUT ST., HOUSTON, TEX.

BRIDGEPORT, CONN. (Office and Factory)

JENKINS BROS., LTD.: LONDON, W.C. 2; MONTREAL, QUE., (Works and Main Office).

IN VALVES Jenkins GIVES YOU EVERYTHING

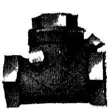


Fig. 702 *Bronze Regrinding Swing Check*

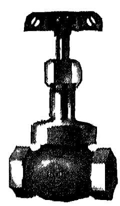


Fig. 106A *Bronze Globe, Renewable Comp. Disc*

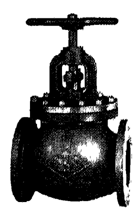


Fig. 613 *Iron Body Regrinding Globe*

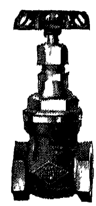


Fig. 370 *Bronze Gate*

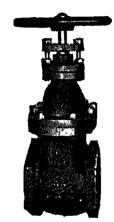


Fig. 325 *Iron Body Gate*

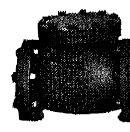


Fig. 624 *Iron Body Regrinding Swing Check*

OVER 500 DIFFERENT JENKINS VALVES COVER EVERY HEATING AND AIR CONDITIONING NEED

To adequately describe the complete Jenkins line of valves requires a Catalog of more than 300 pages. There are over 500 different types and patterns of valves that bear the trusted "Diamond" trade mark. Practically speaking, Jenkins can furnish any valve that you may require for plumbing, heating, air conditioning, general industrial or engineering service.

General Classifications of Jenkins Valves Include—Bronze Valves fitted with Jenkins renewable composition disc. Bronze Regrind-Renew Valves with bevel and plug type seats. Bronze Gate Valves. Iron Body Valves fitted with Jenkins renewable composition disc. Iron Body Regrinding Valves. Iron Body Gate Valves with solid wedge and double disc parallel seats. All-Iron Valves. Cast Steel Gate, Globe and Swing Check Valves. Electrically and Hydraulically Operated Valves. Radiator Valves. Fire Line Valves. Quick-opening and Self-closing Valves, Needle Valves, Y Valves, Solder-End Valves, Stainless Steel Valves.

Other Jenkins Products Are—Colored Valve Wheels with or without service markings molded in relief letters. Composition Valve Discs exactly suited to service conditions. Sheet Packing. Gaskets. Moncrieff Scotch Gage Glasses.

JENKINS VALVES ARE SOLD BY GOOD SUPPLY HOUSES EVERYWHERE

The American Brass Company

Availability—Anaconda Copper Tubes, in all standard sizes, are carried in stock by distributors of Anaconda Pipe, located in the principal trading areas of the country. These tubes, in sizes up to and including 1¼ in. are furnished soft in 30, 45 and 60-ft coils; also hard and soft in 20-ft straight lengths. Sizes over 1¼ in. are furnished, hard or soft, in straight lengths only.

ANACONDA "85" RED BRASS PIPE

Anaconda "85" Red Brass Pipe, in standard pipe sizes, is offered as the highest quality corrosion-resistant pipe commercially obtainable at a moderate price and is recommended for steam return lines.

Anaconda "85" Red Brass Pipe contains 85 per cent copper and conforms to government specifications for Grade "A" water pipe. The words "Anaconda 85" are stamped in the metal at one-foot intervals throughout each length.

EVERDUR*

For many years the efforts of metallurgists have been directed to finding some element or elements which, when added to copper would alloy with it to produce a metal with strength approaching that of steel and, at the same time, retain or augment the non-rusting and corrosion-resistant properties of copper.

The addition of silicon and small amounts of other elements to copper, when their proportions are properly adjusted, produces copper-rich alloys of the solid solution type which attain the desired objective to a remarkable degree. Copper-silicon alloys, made and sold by The American Brass Company under its trademark "Everdur," were the first commercial applications of copper containing substantial proportions of silicon, and mark a decided advance in the metallurgy of copper alloys.

In addition to their non-rusting properties and high strength, Everdur alloys possess many qualities not usually found in metals of this character. They are unusually resistant to general atmospheric conditions and other normally corrosive factors. Everdur alloys have excellent machining and working characteristics and can be fabricated into a variety of forms and shapes. They also weld readily by any of the commercial methods.

*"Everdur" is a trademark of The American Brass Company registered at the U. S. Patent Office.

CORROSION RESISTANCE

The corrosion resistance of Everdur is equal to that of pure copper and in some cases, slightly superior.

However, like copper and all copper alloys, Everdur is not equally resistant to all corroding agents, nor to the same corroding agents under all conditions. As with copper, the resistance to corrosion may be substantially reduced in some instances by the presence of oxidizing agents. Nevertheless, Everdur does offer excellent resistance to the corrosive action of many solutions and atmospheres.

Everdur Tanks—Everdur copper-silicon alloy is an ideal material for durable, rustless water tanks of every description—from domestic range boilers to large storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries.

Everdur is made in all commercial shapes including tank plates which have physical properties as given in A.S.T.M. Tentative Specification B96-40T.

Minimum specification requirements for hot rolled and annealed tank plates are: Tensile Strength, 50,000 psi.; Yield Strength (at 0.5 per cent elongation under load) 18,000 psi.; Elongation, 40 per cent in 2 inches.

Sound, double welded butt joints made on annealed Everdur tank plates have a minimum tensile strength of 47,000 psi. and single welded butt joints have a minimum tensile strength of 42,500 psi. after the beads have been removed.

For additional data and names of fabricators address our nearest office or agency.

EVERDUR FOR AIR CONDITIONING EQUIPMENT

Because of its strength and welding properties, Everdur may be substituted for steel and fabricated by substantially the same methods and with the same equipment as steel.

Everdur metal has been used with marked success for fans and blowers, ducts, humidifiers, cast and wrought parts of other equipment items subject to corrosive influences.

EVERDUR LITERATURE

Descriptive literature containing much pertinent tabular data will be sent upon request.

Arthur Harris & Co.

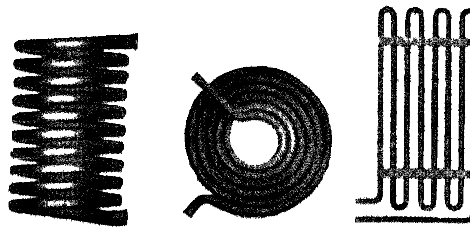
210-218 N. Aberdeen (formerly Curtis) Street **Chicago, Ill.**

ENGINEERS — FABRICATORS OF NON-FERROUS METALS AND STAINLESS STEEL

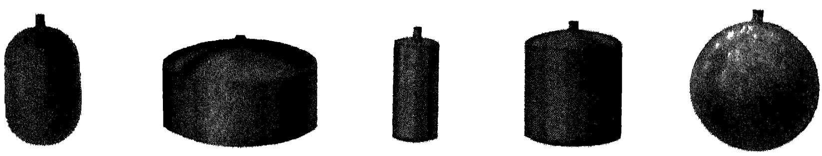
Metals Fabricated—Aluminum, Block Tin, Brass, Bronze, Copper, Everdur, Monel, Nickel, Inconel, Stainless Steel and KA2 SMO. Bulletin on request.

Coils

For heating, cooling and condensing. All shapes made from any size pipe or tube—standard or special connections, of copper, brass, aluminum, stainless steel, KA2 SMO, monel, inconel, nickel, block tin, and Everdur.



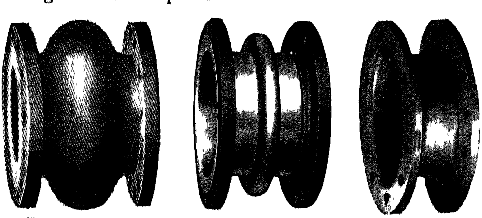
Metal Floats



Column *Flat Cylindrical* *Cylindrical* *Cylindrical* *Ball*

Made of copper, plain steel, stainless steel, KA2 SMO, aluminum, brass, Monel, pure nickel, Admiralty and Everdur, for open tank and all pressures.

Seamless copper ball floats carried in stock in diameters of 3 in., 4 in., 5 in., 6 in., 7 in., 8 in., 10 in., 12 in. for open tank and pressures of 25, 50, 100 and 150 lb. Floats in special sizes and pressures made to order. Stainless steel ball floats 2½ in. to 12 in. for high pressure and corrosion carried in stock special stainless steel floats made to order—stainless steel ball floats larger than 12 in. diameter can be made up specially. Float catalog sent on request.

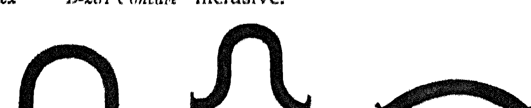


B-280 Convex *B-200 Convex* *B-281 Concave*

Copper Expansion Joints

For low pressure and vacuum. Made in two styles convex and concave. Sizes 4 in. to 60 in. diameter. Cast iron or steel flanges. Flanges drilled to American standard unless otherwise ordered: B-200 available only in sizes 4 in. to 15 in. inclusive.

Bends



We make bends in every shape from all sizes of copper water tube, pipe and tubing in copper, brass, aluminum, stainless steel, monel, tin and nickel. Standard or special connections. U-bends for storage water heaters.

Also special pipe work for industrial installations, plumbing, heating and brewing. Perforated pipe, double pipe coolers, etc.

Non-Ferrous Castings—"Dairywhite" nickel silver for Process Industries Equipment. Suitable for milk and food products machinery. Castings also of 88-10-2 80-10-10, 85-5-5-5 and special mixtures. Many patterns available without charge.

Wolverine Tube Company

1411 Central Avenue, **Detroit, Michigan**

SEAMLESS TUBE

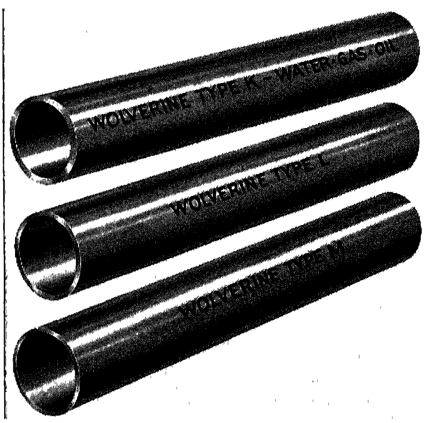
COPPER - BRASS - ALUMINUM

Sales Offices:

ATLANTA, GA. .... 542 Spring Street
*BALTIMORE, MD. .... 121 S. Gay Street
*BOSTON (Cambridge), MASS. .... 195 Albany Street
BUFFALO, N. Y. .... 416 Jackson Bldg.
*CHICAGO, ILL. .... 1632 W. Lake Street
*CHICAGO, ILL. .... 3348 S. Pulaski Road
CLEVELAND, OHIO .... 1740 East 12th Street
*DALLAS, TEXAS .... 2813 Canton Street
DAYTON, OHIO .... Route No. 9
DENVER, COLO. .... 1210 California Street
*LONG ISLAND CITY, N. Y. .... 47-31 31st Place
*LOS ANGELES, CALIF. .... 1015 East 16th Street
LOUISVILLE, KY. .... 125 South 5th Street
MILWAUKEE, WIS. .... 647 W. Virginia Street
*MINNEAPOLIS, MINN. .... 100 North 2nd Street
*NEWARK, N. J. .... 34 Providence Street
NEW YORK, N. Y. .... 420 Lexington Avenue
*PHILADELPHIA, PA. .... 351 North 7th Street
*PITTSBURGH, PA. .... 1000 California Ave., N.S.
*PORTLAND, ORE. .... 524 N.W. 14th Avenue
*TULSA, OKLA. .... 716 S. Troost Street
ST. LOUIS, MO. .... 4565 McRee Avenue
SAN FRANCISCO, CALIF. .... 7 Front Street
SEATTLE, WASH. .... 1005 E. Pike Street
WASHINGTON, D. C. .... 1108 16th Street
WINNIPEG, MAN. .... 80 Lombard Street
EXPORT: ROCKE INTERNATIONAL ELECTRIC CORP., 100 Varick Street, New York, N. Y.

*Warehouse Stock.

WOLVERINE COPPER WATER TUBE



Type K

Recommended for Air Conditioning, Refrigeration, Oil Burner, and Plumbing and Heating installations. Also for Gas, Steam, Oil Lines, and industrial uses. This is the heaviest of the three types and is best for underground use.

Type L

For Oil Burner, Air Conditioning, Refrigeration, and general plumbing uses.

Types K and L furnished in hard or soft temper in straight 20 ft lengths; soft temper in 30, 45, 60 ft coils, longer on special order.

Type M

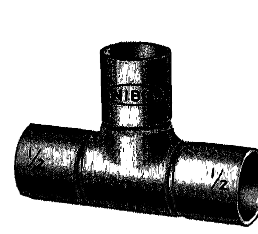
Suitable for Air Conditioning and Refrigeration installations and for interior plumbing and heating purposes.

Furnished only in hard temper—straight 20 ft lengths.

Wolverine Water Tube is made according to U. S. Government and A.S.T.M. specifications. For a complete list of this data, write Detroit for Form 575.

WROUGHT FITTINGS

Wolverine-Nibco Solder Fittings are of the straight-line design—ends not expanded. They make stronger, neater joints more quickly; give trouble-free service, and longer life. A complete range of sizes is available. Write to Detroit or your nearest warehouse for Catalog C.



The experience of 25 years of seamless tube manufacture, the use of the latest equipment, and adherence to Government and customer specifications, are responsible for the uniform, high quality of Wolverine products.

Immediate Shipment From Large Stocks

In the four preceding sub-divisions of this Catalog Data Section, equipment used in heating, ventilating and air conditioning service is illustrated and described.

The sizes and capacities of such equipment required to obtain specific results are directly affected by the characteristics of materials used in building construction and for insulating exposed surfaces of the apparatus itself. Because of their influence upon the efficiency of equipment, selection and application of these materials requires careful study.

In Chapter 3, of the Technical Data Section, the physical properties of commonly used building materials and types of insulating materials are tabulated. From the data given in these tabulations their value and suitability for specific uses may be calculated.

The following sub-division—Insulation—gives manufacturers data on many types of insulating materials used in modern construction. These data, used in conjunction with engineering data given in Chapter 3, will enable engineers to determine the materials most suitable for a specific purpose, and the method of application which will give best results.

INSULATION

Many different materials are used for insulating purposes—in their natural state or processed and fabricated into various forms. They include: Vegetable fibers, wood, tree bark, cork—processed into wools or other fibrous forms, and used in loose bulk or fabricated into boards, paper, blankets or batts. Natural wools, jute, hair—felted into blankets, pads, mats, etc., or used in loose bulk forms. Glass in block, sheet, or wool forms.

Mineral products such as natural rocks and furnace slags—processed into granulated form, or into wool form and used in loose bulk or fabricated into blankets, batts, or pads; and asbestos, asphalt, gypsum and magnesia—used in board form, blankets, felts, or in loose bulk. Many of these types of insulation are also used in plastic form. Metallic insulation, such as aluminum and steel are fabricated into sheet form and used separately or in conjunction with other insulating materials.

INSULATION, Window (p. 1048-1053)

Single-pane and double-pane insulating window sash, metal fabric insulating window screens, weather stripping for windows and for interior and exterior doors.

INSULATION, Building (p. 1054-1084)

Aluminum sheets, paper in sheets and fabricated forms, felts, cork, glass, glass and rock wools, cane fibre boards, wood products in board form and fibrous blankets and pads, or used in loose fibre form—all are utilized as insulation against heat or cold.

Technical data on this type of insulation will be found in Chapter 3.

Insulating materials, in board or slab form are adapted for use in walls as a plastic base, and thus serve as both a heat or cold insulation and a fire-retarding material.

INSULATION, Sound Deadening (p. 1054-1084)

Many of the insulating materials utilized in building construction are also suitable for sound deadening or acoustical control. Some of them are also adapted for use on machinery and in building to counteract or absorb vibration.

Technical data on Sound Control will be found in Chapter 32.

INSULATION, Underground (p. 1056, 1057, 1065)

Asbestos, asphalt, mineral wools, magnesia—used in conjunction with underground piping and conduits of concrete, tile or cast iron.

Technical data is contained in Chapter 42.

INSULATION, Pipes and Surfaces (p. 1060-1084)

Asbestos, magnesia, and mineral wools in loose fibrous forms, blankets, or in plastic forms and suitable for use in extremes of high or low temperature service; also hair and felts, and cork in loose bulk or in molded or plastic forms.

Technical data will be found in Chapter 42.

Some of these insulating materials are also used as refractory materials.

INSULATION, Duct (p. 1054-1084)

Various of the insulating materials which may be fabricated into board or slab forms, and various felts and fibrous materials have been adapted for use as duct insulation—as a duct liner or applied to the outer surfaces. Some have been utilized to construct the walls of the duct itself, serving the dual purpose of duct and insulation.

Technical data is contained in Chapter 42.

Manufacturer's products shown in this division are designed for specific applications. Consult the Index to Modern Equipment for additional products of these manufacturers.

Chamberlin Metal Weather Strip Co., Inc.

General Offices, Detroit, Mich. **Factories, Detroit, Mich., Peru, Ill.**

CHAMBERLIN HEAT-SAVING PRODUCTS
Weather Strips, Calking, In-Dor-Seals, Insulation and Insulate-Windows.

Rock Wool Insulation

Having an average conductance coefficient of 0.24 to 0.27 Btu, Chamberlin Rock Wool is one of the most efficient insulations available today. Long in fibre and clean of "shot", it insures effective insulation with low density and light weight. It is available in several forms adaptable to both new and existing construction—in blowing, commercial or loose wool fibres; in wall-thick or 2 in. thick batts, with or without vapor-proofing coverings. Among its physical properties Chamberlin Rock Wool averages 38 per cent silica, 32 per cent lime and only 0.04 per cent sulphur. Its loss on ignition is less than 0.1 per cent. Wherever possible pneumatic application, by means of such modern high-powered and efficient transient units as illustrated, is preferable. Here again, the engineer will do well to consider the value of faithful, responsible performance in the art of application—the hidden factor that means so much in obtaining the utmost efficiency from insulation.

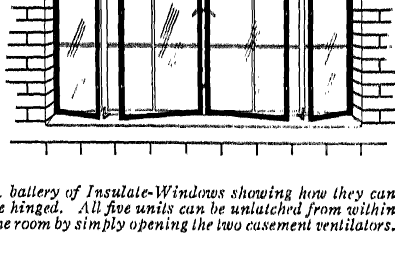
Insulate-Windows (metal Storm Sash)

In addition to other major heat loss controls, Chamberlin has developed a product that reduces 40 to 60 per cent of the heat loss of glass in standard window construction.

Thousands of Chamberlin Insulate-Window (registered tradename) units have been applied to wood and metal windows in existing buildings, including offices, banks, residences, etc.; they are *not* double glazed units. They are single glazed and custom made *to supplement* existing windows of practically any type. Depending upon characteristics of the areas and sash to be "insulated," Insulate-Windows are hinged, stationary or sliding. They are framed with a patented rolled section of antique-finished, cold-rolled bronze embodying ingenious features of mechanical glazing in place of putty. Chamberlin "insulation" of windows is a high grade, mechanical service involving design and installation of durable, secondary window units, readily accessible for cleaning and removal when necessary—no interference with sash operation or ventilation. Insulate-Windows provide a year 'round heat stop that gives practical relief without the technical difficulties involved in double-glazed windows.

Typical Chamberlin Insulation unit. Equipped with powerful, gas-powered, pneumatic blower capable of projecting granular grade "A" rock wool a minimum distance of more than 200 ft through a large rubber hose.

At right—near view of "insulated" residential metal casement windows. Chamberlin Insulate-Windows on outside.



A battery of Insulate-Windows showing how they can be hinged. All five units can be unlatched from within the room by simply opening the two casement ventilators.

Chamberlin Metal Weather Strip Co., Inc.

General Offices, Detroit, Mich. **Factories, Detroit, Mich., Peru, Ill.**

Factory Sales—Installation Branches

Atlanta, Ga.
Baltimore, Md.
Boston, Mass.
Buffalo, N. Y.
Chicago, Ill.
Cincinnati, Ohio
Cleveland, Ohio
Dallas, Texas
Denver, Colorado
Detroit, Mich.
Indianapolis, Ind.
Kansas City, Mo.
Los Angeles, Cal.
Louisville, Ky.
Memphis, Tenn.
Milwaukee, Wis.
Minneapolis, Minn.
Newark, N. J.
New Haven, Conn.
New York, N. Y.
Philadelphia, Pa.
Pittsburgh, Pa.
Portland, Ore.
Providence, R. I.
Raleigh, N. C.
St. Louis, Mo.
San Francisco, Cal.
Schenectady, N. Y.
Washington, D. C.

CHAMBERLIN HEAT-SAVING PRODUCTS
Weather Strips, Calking, In-Dor-Seals, Insulation and Insulate-Windows.

Weather Strips

Modern weather strip service, in which engineers and architects can have greatest reliance, is one that can responsibly fulfill conditions of a contract. For problems of infiltration or circulation of air, gases or mixtures, leakage of rain, filtration of dust, sand or soot—there's a Chamberlin Weather Strip remedy regardless of climatic or construction conditions. Particularly helpful to the engineer is the knowledge that the job can be entrusted, regardless of location, to a Chamberlin Factory Branch employing experienced mechanics.

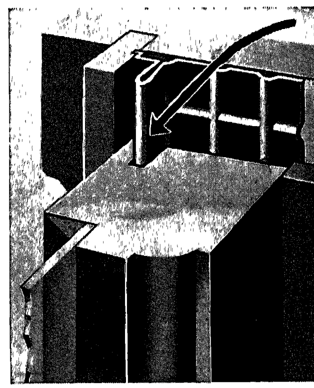
Modern problems of heating, ventilating and air conditioning can be approached with utmost confidence in this field with the aid of Chamberlin's proven 47 years of experience and specialized training. Write for *A.I.A.* catalog of standard details and specifications. Consult nearest branch for equipments, surveys, quotations.

Calking

Chamberlin Plasti-Calk is essential in the sealing of construction joints in wood, metal, glass, stone, tile, concrete and brick. It is waterproof, permanently elastic, nonstaining and noncorrosive. It provides durable adhesion and will not sag, pucker, or shrink under extremes of heat or cold, dryness or moisture. Chamberlin Plasti-Calk is specially prepared with porous pigments capable of retaining oil indefinitely, and does not contain tar or asphalt. Supplied in various colors. It is all important that a calking compound be specified thoroughly on the basis of stringent physical properties. Such specifications will be furnished upon request.

Automatic Door Bottoms

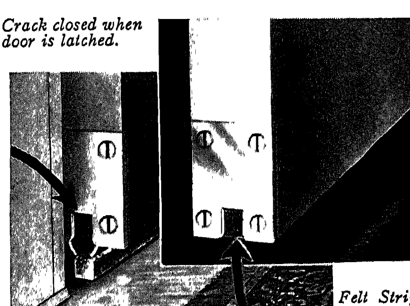
In-Dor-Seals eliminate under-door drafts and heat losses in rooms adjacent to cold areas, corridors and halls—as in a sleeping room door—virtually an "outside" door at night when windows are open. They are used to light-proof X-Ray and dark rooms; to sound-proof private offices, studios, laboratories; to resist room-to-room circulation of odors, fumes and dust.



Oldest and most extensively successful weather strip principle—tongue in groove. 80 to 95 per cent efficient. No failure or upkeep.

Calking applied with hand tools, hand or power gun.

Crack closed when door is latched.



Felt Strip—automatically raised the instant door opens.

Ingersoll Steel & Disc Division

Borg-Warner Corporation

310 So. Michigan Ave., **Chicago, Illinois** **Distributors in the Principal Cities**

Ingersoll
KOOLSHADE* *Sun Screen*

"It's cooler in the shade!"

A Metal Fabric That Reduces the Solar Load Through Windows as much as 80 to 85 Per Cent.

KOOLSHADE Sun Screen has had such ready acceptance among Air Conditioning Engineers because it offers high efficiency in cutting down the Solar Load without shutting off light or vision. Its advantages are: (1) Lowers the cost of cooling equipment necessary by stopping 80 to 85 per cent of the Solar heat outside the window glass. (2) Saves in operating cost—as much as 25 to 50 per cent, where there are large windows. (3) Reduces the need for zoning for sun effect. (4) Forestalls complaints due to extreme sun heat. (5) Maintains lower temperatures in non-cooled rooms.

▲*THIS WINDOW equipped with KOOLSHADE no direct sun comes in but ample light.*

▲*THIS WINDOW is bare and shows how the sun rays stream into the room.*

Actual photo of KOOLSHADE installed. Stops direct Sun Heat . . . kills Sun Glare . . with abundant light and clear vision . . . yet it is practically invisible!

Automatic Sun Shade. KOOLSHADE is always in position . . . functions automatically . . . protects most when most needed. The flat horizontal wires, held at a FIXED angle of 17 deg, fully stop direct heat rays when the sun is 38½ deg or more above the horizon—the heat of the day in all seasons. Needs no adjustment or human attention.

Light and Vision. KOOLSHADE gives sun-exposure rooms the effect of cool "north" light, free from glare. Vision from within is superior to that through windows with 18-mesh fly screen.

Attractive Appearance. Makes the window look smart, and does not mar the outside appearance of the building.

Economy. KOOLSHADE is strong, tough and should last for many years. Maintenance cost is negligible and first cost is low.

Other Advantages include full insect protection, free ventilation, reduction of fading of fabrics and fire-safety.

Specifications of Fabric. Made of fine quality bronze, with 17 horizontal wires per in. and vertical wires spaced ½ in. apart. Supplied in widths up to 72 in.

Framing and Installation. Framed and installed like ordinary insect screens. Frames may be either wood or metal—always full length and always outside the window glass. Re-wiring of present screen frames is entirely practical.

Where to Buy. Authorized KOOLSHADE distributors and dealers in all principal cities offer adequate facilities to assure correct frame design and proper installation.

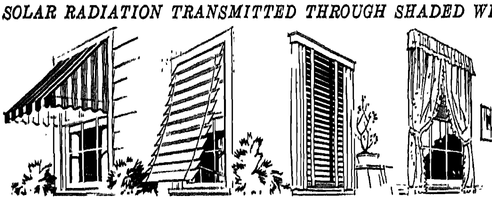
Technical Proof of Performance. Reports of Tests and Calculations by the Pittsburgh Testing Laboratories provide ample evidence as to the performance of KOOLSHADE. A full copy of this report (from which one table is reproduced below) will be sent upon request for Brochure HG102.

For Full Literature and information, write for our Booklet HG101 entitled "KOOLSHADE SUN CONDITIONING."

Framed and Installed like ordinary fly screen — and keeps out insects, too.

KOOLSHADE SUN LOAD 19 PER CENT

SOLAR RADIATION TRANSMITTED THROUGH SHADED WINDOWS



Awnings in Use Sun Load 22 Per Cent to 28 Per Cent — *Outside Venetian Blind or Shutter Sun Load 22 Per Cent* — *Inside Venetian Blind Sun Load 58% or more* — *Half-Drawn Shade Sun Load 68% or more*

DATA FROM PITTSBURGH TESTING LABORATORY

FROM CALCULATIONS BASED ON ACTUAL TESTS

Solar Radiation Transmitted Through Windows Equipped with KOOLSHADE Sun Screen

For 40 deg. latitude, on July 21st **All figures given represent B.t.u. per sq. ft. per hour.**

TIME	SOLAR RADIATION	NE	EAST	SE	SOUTH	
6 AM	Intensity Incident to Vertical Surface (1)	72	80	40		6 PM
	Transmitted thru Window with KOOLSHADE (2)	33	38.5	13		
7 AM	Intensity Incident to Vertical Surface	143	180	112		5 PM
	Transmitted thru Window with KOOLSHADE	38.5	60.5	22		
8 AM	Intensity Incident to Vertical Surface	143	211	155	8	4 PM
	Transmitted thru Window with KOOLSHADE	17	42.5	20.5	0	
9 AM	Intensity Incident to Vertical Surface	104	192	168	46	3 PM
	Transmitted thru Window with KOOLSHADE	8	22	17	2	
10 AM	Intensity Incident to Vertical Surface	46	143	156	77	2 PM
	Transmitted thru Window with KOOLSHADE	1.5	11.5	13	4	
11 AM	Intensity Incident to Vertical Surface		75	121	95	1 PM
	Transmitted thru Window with KOOLSHADE		3.5	8.5	5.5	
12 M	Intensity Incident to Vertical Surface			73	103	12 M
	Transmitted thru Window with KOOLSHADE			3.5	6	
		NW	WEST	SW	SOUTH	TIME

(1) Data from the A.S.H.V.E. Guide 1940.
(2) Figures represent Solar Heat Gain in EXCESS of heat gain by conduction through window glass.

Libbey-Owens-Ford Glass Company

Toledo, Ohio

WINDOW CONDITIONING WITH DOUBLE GLAZING

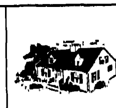
Window conditioning (storm sash) will save approximately 23 per cent of the fuel used in the typical uninsulated suburban residence, according to the results of a study of five representative types of houses. These calculations (see table on this page) were based on the use of outside storm sash and storm doors or the equivalent use of modern weather stripped casement windows with prefit window conditioning panels.

If these houses also had ceiling insulation (in addition to double glazing), the total savings in heating costs would be 40 to 47 per cent as compared with an unprotected house.

Window conditioning does more than merely cut down fuel bills, it helps to eliminate cold drafts and reduces to a minimum the possibility of fogging on windows. It is essential to satisfactory winter air conditioning with its higher healthful humidity. Window conditioning, double glass insulation, keeps the inner pane of glass comparatively warm even though the outer pane may be as cold as the outdoor air. This reduces condensation on the inner pane of glass to a minimum.

There are many forms of prefabricated, double-glazed windows available in both wood and steel sash, including double hung, casement, and horizontal gliding types.

FUEL SAVINGS AND COMPARATIVE HEATING COSTS
FIVE TYPES OF HOMES WITH AND WITHOUT WINDOW CONDITIONING AND ATTIC INSULATION

										
	Attic area vented above insulation	1488.5 sq. ft.		770 sq. ft.		1143 sq. ft.		995 sq. ft.		782 sq. ft.
	Sidewalls net	2447.7 " "		1634 " "		1332 " "		1197.5 " "		695 " "
	Window area	540.3 " "		326 " "		363 " "		285 " "		280.8 " "
	Crack length	590.4 lin. "		389 lin. "		422 lin. "		365 lin. "		436 lin. "
	Unheated floor	None		None		None		None		782 sq. "
ZONE 9	Heating cost—no insulation—oil 7 cents per gallon	$315.50		$190.60		$211.00		$173.80		$178.50
	Heating cost if attic is insulated	254.50		159.50		164.70		133.90		146.90
	Heating cost with window conditioning	241.25		144.55		159.50		131.90		137.00
	Savings due to insulation 3½" minimum wool in attic floor	61.00	19.3%	31.10	[illegible]	[illegible]	21.9%	[illegible]	[illegible]	[illegible]	17.7%
	Savings due to window conditioning	74.25	23.5%	[illegible]	24.2%	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	Savings with both TOTAL	$135.25	42.8%	$ 77.15	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	$ 73.10	[illegible]
ZONE 7	Heating cost no insulation oil 7 cents per gallon	$245.00		$149.30		$161.00		$136.00		$139.55
	Heating cost if attic is insulated	198.00		124.90		125.00		104.40		114.75
	Heating cost with window conditioning	186.25		112.80		123.00		103.80		106.90
	Savings due to insulation 3½" minimum wool in attic floor	47.00	19.2%	24.40	16.3%	36.00	22.4%	[illegible]	23.3%	24.80	17.7%
	Savings due to window conditioning	58.75	24.0%	[illegible]	24.4%	38.00	[illegible]	[illegible]	23.7%	32.65	23.4%
	Savings with both TOTAL	$105.75	43.2%	[illegible]	40.7%	$ 74.00	46.0%	[illegible]	[illegible]	$ 57.45	41.1%
ZONE 5	Heating cost no insulation oil 7 cents per gallon	$176.00		$106.35		$116.30		$ 96.30		$100.50
	Heating cost if attic is insulated	142.50		88.95		91.20		74.00		82.40
	Heating cost with window conditioning	134.25		80.80		88.40		73.65		77.00
	Savings due to insulation 3½" minimum wool in attic floor	33.50	19.0%	17.40	16.4%	[illegible]	[illegible]	22.30	[illegible]	18.10	18.0%
	Savings due to window conditioning	41.75	23.7%	25.55	24.0%	[illegible]	[illegible]	22.65	[illegible]	[illegible]	23.4%
	Savings with both TOTAL	$ 75.25	42.7%	$ 42.95	40.4%	[illegible]	[illegible]	[illegible]	46.7%	$ 41.60	41.4%

INSULATION CALCULATIONS CHECKED BY ALFRED J. OFFNER, CONSULTING ENGINEER

OWENS-ILLINOIS
INSULUX
Glass Block

Owens-Illinois Glass Company

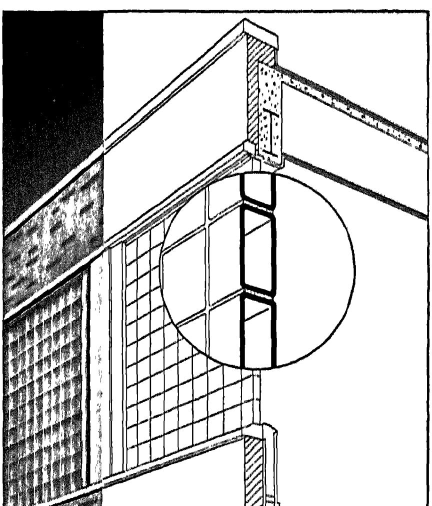
INSULUX PRODUCTS DIVISION

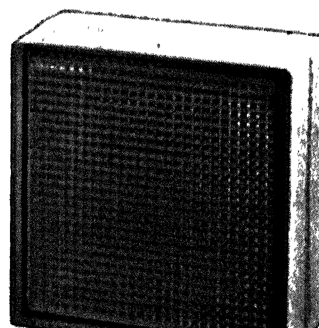
Toledo, Ohio

Dealers in All Principal Cities

Insulux Glass Block Give Better Control of Interior Conditions

Insulux Glass Block are hollow, partially evacuated block, 3⅞ inches thick, with ribbed or patterned walls. Laid up in mortar in solid panels, they form a light-transmitting area that also offers high insulation value. Proper use of Insulux Glass Block results in better control of interior conditions, and therefore greater efficiency and lower initial and operating costs for cooling and heating plants.





Better Insulation

The cross-section drawing above shows why Insulux Glass Block panels give higher insulation than ordinary light-transmitting areas. The glass block, partially evacuated and with thick faces, lower the conductivity and solar heat transmission of the light-transmitting area. Air infiltration is eliminated. The better insulation provided by Insulux is a factor in planning air conditioning and heating equipment and operating costs.

Lower Heat Transmission

Tests on conductivity of Insulux Glass Block show that the heat transmission of Insulux is approximately the same as for a concrete wall 16 inches thick or a brick wall 8 inches thick. This test data is available for inspection by engineers.

Reduction of Solar Heat

In a comparative test of solar heat transmission, a single glazed steel sash transmitted 94 per cent more heat than an Insulux panel. As with sash, however, Insulux panels transmit less solar heat if properly oriented and well shaded. There is variation in the solar heat transmission of different designs of Insulux data will be furnished on request.

Designs, Sizes, Erection

There are 8 designs of Insulux for both residential and industrial use. Block available in three sizes. Panels are easily and quickly erected by bricklayers. We will gladly supply any technical information and advice on installations on request.

Pittsburgh Corning Corporation

Grant Building, Pittsburgh, Pa.

Distribution through Pittsburgh Plate Glass Company warehouses in principal cities and by the W. P. Fuller Company on the West Coast.

Glass Blocks allow the economical use of large glass areas, reduce heat loss in cold weather and materially aid air-conditioning. This is because each PC Glass Block contains a sealed-in dead-air space that is an effective retardant to heat transfer.

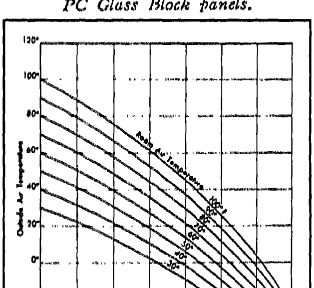
Thermal Insulation

Tests run by nationally recognized laboratories have established the value of glass blocks for insulation of light-transmitting areas. These tests have proved that with glass block panels, heat loss is slightly less than half that experienced with single-glazed windows. In computing heat losses through panels for most design purposes, it is recommended that a "U" value of 0.46 to 0.49 be used for all block sizes and face patterns. For complete data on heat transfer values see the section on heat transfer elsewhere in this Guide—page 97.

Surface Condensation

Due to high insulating value, condensation will not start forming on the room side of glass block panels until outside air has reached a temperature much lower than that necessary to produce condensation on single-glazed windows. The accompanying chart shows at what temperatures condensation will form.

Outdoor temperature required to produce condensation on the room side surface of PC Glass Block panels.



For example, with inside air at 70 F and relative humidity at 40 per cent, condensation will not begin to form on the interior surfaces of a glass block panel until an outdoor temperature of −14 deg is reached. Under similar conditions with single-glazed sash, moisture will begin to form when the outdoor temperature reaches +33 F.

Solar Heat Gain

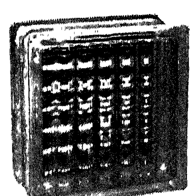
The use of glass blocks for light-transmitting areas results in a marked reduction in total solar heat gain as compared with ordinary windows. This factor is of considerable advantage in buildings that are properly air conditioned, but does not eliminate the need for adequate ventilation or shading in non-air-conditioned rooms.

For data on solar heat gain through glass blocks see the table in the solar radiation section of this Guide—page 135. This table is for standard pattern glass blocks. Where LX-75 blocks (with Fiberglas screen insert) are used, these figures may be reduced approximately 40 per cent.

PC Glass Blocks Aid Air-Conditioning

The three chief aims of air-conditioning—temperature control, humidity control and cleansing of air are all aided by the use of PC Glass Blocks. Heat loss is less in winter—heat gain is less in summer. Ideal humidity conditions are much more easily maintained without undue condensation. Solar heat transmission and radiation are reduced. Dirt can't filter in, for each panel is a tightly sealed unit.

Sizes and Shapes Available



PC Glass Blocks are available in eight attractive patterns, some of the patterns being designed for special control and direction of transmitted daylight. For complete information on the sizes and shapes of PC Glass Blocks, and for illustrations of the many patterns available, write the Pittsburgh Corning Corporation, Pittsburgh, Pa., or call the nearest Pittsburgh Plate Glass Company warehouse.

• • •

Additional technical data, including detailed figures on thermal insulation, solar heat gain, surface condensation, light transmission and construction data, will be furnished on request.

H. W. Porter & Co.

INCORPORATED

Newark, New Jersey

Permanent Protection and Insulation for Underground Pipe Lines

BALTIMORE, MD. CHARLOTTE, N. C. RICHMOND, VA. WASHINGTON, D. C.

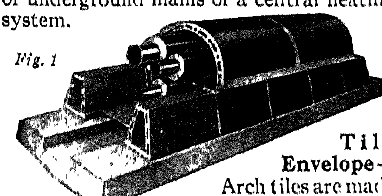
Therm-O-Tile

REG US PAT OFF

STEAM CONDUIT SYSTEMS

For Central Heating—Therm-O-Tile is a complete conduit system for the permanent support, protection, and insulation of underground mains of a central heating system.

Fig. 1



Tile Envelope—Arch tiles are made 6 in. to 24 in. in diameter, and with 5 different size base tiles they produce 27 different conduit cross sections.

Foundation—The base of the system is a thick concrete slab poured directly in the trench bottom, reinforced with steel when installed over a filled or boggy ground. See Figs. 1, 4, 6 and 7.

Drainage—The drainage system of the conduit is entirely internal, open to inspection at manholes, and of ample capacity to keep the pipe space dry at all times without any possibility of becoming clogged with silt or vegetation. See Figs. 6 and 7.

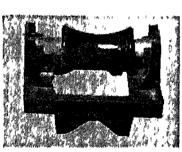


Fig. 2—Pipe Support for Single Pipe.

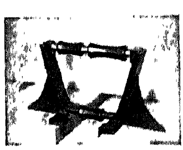


Fig. 3—Pipe Support for Three Pipes.

Pipe Support—All pipes are supported on cast-iron adjustable supports resting directly on the concrete base independent of the tile envelope. Figs. 2, 3, 4, 6 and 7.

Accessibility—All piping is installed before tile is placed, giving complete accessibility for welding, testing and insulation. Pipe fitters work on concrete slab "walkway." Figs. 1 and 4.

Strength—Due to immovable concrete base and arch construction of extra heavy tile members, conduit will sustain any roadway traffic load usually encountered without extra reinforcement.

Insulation Either sectional pipe covering or Thermobestos waterproof fibre filling may be used for insulation, as the insulation space is kept dry at all times, by the internal drain.

For single or double pipe lines, sectional insulation of economical thickness is recommended; for multiple pipe lines, a filler type of insulation is usually more economical in first cost. See Figs. 6 and 7.

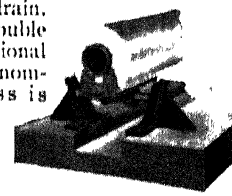


Fig. 4—Pipe Saddle. Permits full thickness of insulation between pipe and roller.

Waterproofing— Under normal soil conditions, this conduit is waterproof. If marshy ground or partially submerged conditions are encountered, the conduit may be made completely waterproof by the use of membrane waterproofing applied under the slab on a sub-base and carried completely over the tile envelope.

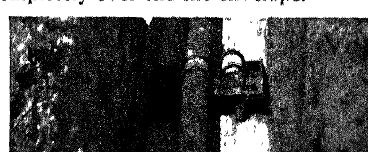


Fig. 5—Anchor Block. Fits directly in line with Base Tiles.

Efficiency—The degree of thermal efficiency secured depends upon the type and thickness of insulation used. This conduit, due to its sealed air chambers in the tile and dry insulation space, adds to the normal efficiency of the insulating material on the pipe lines.

Representatives—Therm-O-Tile is also sold and installed locally by Johns-Manville Construction Units.

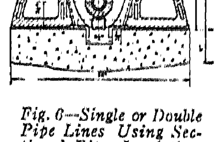


Fig. 6—Single or Double Pipe Lines Using Sectional Pipe Insulation.

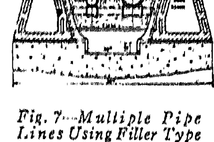


Fig. 7—Multiple Pipe Lines Using Filler Type Insulation.

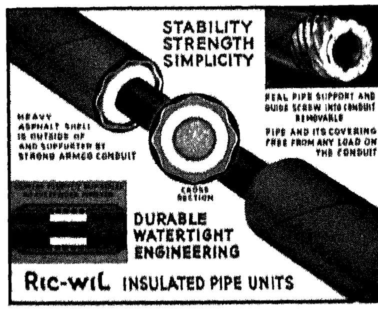
The Ric-wiL Company

Agents in Principal Cities — RIC-WIL (REG. U. S. PAT. OFF.) — ESTABLISHED IN 1910

CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

Union Commerce Bldg., **Cleveland, Ohio** — **New York, Chicago, San Francisco**

Ric-wiL Insulated Pipe Units Pre-fabricated, pre-sealed, ready-to-install units, ideal for speed and economy. Armco Iron Conduit is the foundation supporting heavy asphalt shell of any desired thickness a permanent housing for the insulated pipe which is surrounded with a protective air-space. Ample structural strength, lightweight and watertight. Furnished in any lengths, for single or multiple pipes, with any kind of steam pipe or insulation, for underground or overhead steam lines. *Welded* couplings used if preferred. Write for latest Unit Bulletin.

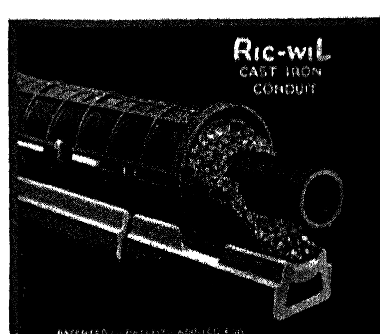


Types of Sectional Conduit Ric-wiL SuperTile Conduit, shown below, with Dry-paC Insulation, is an extra weight, heavy duty system designed for use under highway traffic or in especially wide or deep trenches. Vitrified tile, split on the job, with Loc-liP Side Joints, interlocking construction throughout. Same design also furnished in standard weight tile (Type F). Tile is bell and spigot design, lined or unlined, and comes in 24 in. sections, 4 in. to 27 in. inside diameter. For extra heavy duty under railways, Ric-wiL is made of cast iron in 2 or 4 foot sections. Where continuous concrete base, poured on job, is desired, and reduced labor cost not essential, Ric-wiL Universal Type System is recommended. Each system supplied complete with proper pipe supports, accessories, and insulation as specified. Separate bulletin on any one of these Ric-wiL types supplied on request.

Base Drain Standard Base Drain is vitrified salt glazed tile for tile conduit and extra heavy tile or cast iron for the cast iron conduit, in 24 in. lengths. Made in three sizes to support and drain properly all conduit sizes.

Insulation Ric-wiL Dry-paC Waterproof Insulation is high-grade long fibre asbestos, specially processed. Any grade of commercial hand packed insulation can be furnished, also sectional pipe covering. For lined conduit, diatomaceous earth mixture is molded and keyed inside the tile.

Engineering Service Full cooperation with architects and engineers. Installation supervision if desired. Write for Catalog Bulletin with valuable underground data.



Alfol Insulation Company

Incorporated

155 East 44th St., **New York, N. Y.**

Agents in Principal Cities

HEAT INSULATION for

Buildings
Houses—Summer and Winter
Ships and Naval Vessels
Railroads
Oil Refineries
Heating Equipment

ALFOL

PAT. IN 34 COUNTRIES

ALL PURPOSES

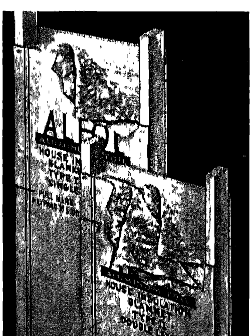
Radiator Reflectors
Cold Storage Rooms
Storage Tanks
Pipes and Ducts
Ovens and Kilns
Trucks and Trailers

Aluminum Foil insulates against heat and cold because of its extremely high reflectivity and low emissivity.

For latest test data and information write us for—

Bulletin and Circular issued by National Bureau of Standards.
Reprints of articles on Reflective Insulation.

For technical data see Table 1, Section C, page 74, and pages 73-76, this volume.

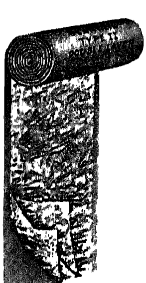


ALFOL HOUSE INSULATION BLANKET

99.4 per cent Pure Aluminum Foil spaced on three ply waterproofed vapor barrier sheet. Provides spaced sheets to reduce conduction and convection. Recent tests show that reflectivity of ALFOL is 97 per cent. Applied between structural members or furring, ALFOL blankets give high value in insulation at a low cost for material and labor.

Specifications

	Description	Widths	Net Area per Roll	Net Weight per Roll
Type 1	1 Layer ALFOL....	16" 20" 24"	250 sq. ft.	17 lbs.
Type 2	2 Layers ALFOL....	16" 20" 24"	250 sq. ft.	19 lbs.

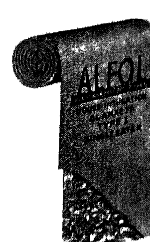


Rolls of Type 1 or Type 2

Rolls of ALFOL House Insulation Blanket weigh less than 20 pounds, are about 7 in. in diameter. Insulation for a house can be carried easily in a passenger automobile.

Permanence

Inspections of our earliest applications prove that insulating value of ALFOL is permanent. Write for Circular.



ALFOL RADIATOR REFLECTOR

ALFOL Insulation in the form of heat reflectors behind radiators reduces heat loss through walls, saves fuel. Temperature gradient to outside reduced 50 per cent.

Ten Years of Service Proves Lasting Value of ALFOL

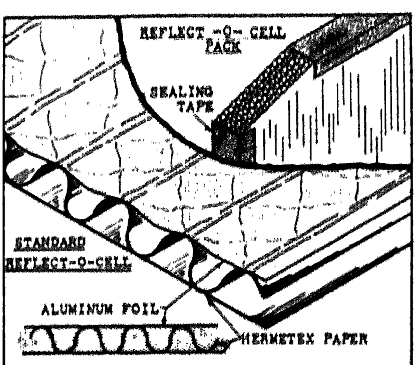
Aluminum Aircell Insulation Co.

Curtis Bldg., **Detroit, Mich.**

INSULATION FOR		INSULATION FOR
Air Conditioners		Automobiles
Brooders		Beehives
Buildings		Busses
Chicken Houses		Floral Shipping Boxes
Dairy Barns		Heaters
Fruit Storage		Ovens
Homes		Ranges
Hot Houses	Licensed under Patents:	Shipping Containers
Incubators	1,757,479 1,890,418 1,934,174	Trailers
Refrigerated Rooms	others pending	Trucks

	Heat Transmission Btu/Hour/sq ft/F			Reflect-O-Cell Saves per cent	
	Not Insulated	1 layer Reflect-O-Cell	2 layers Reflect-O-Cell	1 layer Reflect-O-Cell	2 layers Reflect-O-Cell
Wood Stud Wall · 4 in. studs	0.26	0.13	0.09	50	65
Open Attic Floor	0.69	0.19	0.12	69	82
Wood Shingle Roof (no lath and plaster)	0.56	0.18	0.10	68	81

Thermal Efficiency determined at the University of Detroit under humidity conditions prevailing in actual use and also at the University of Toronto, Toronto, Canada.



REFLECT-O-CELL is the modern insulating material based on the famous Dewar Principle of insulation, the outstanding example of which is the efficient Thermos Bottle.

Light in Weight, not depending on thickness nor density, REFLECT-O-CELL employs this Dewar Principle by means of its polished Aluminum Foil backed by corrugated Kraft paper.

The Structure of Reflect-O-Cell permits both surfaces of the foil to reflect radiant heat.

Moisture Sealing, which effectively reduces summer dehumidifying load and winter humidifying requirements by preventing vapor pressure losses.

Windproofing all studding spaces similar to metallic window weatherstrip effect.

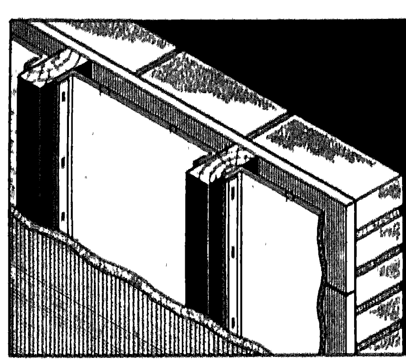
Soundproofing provided by high hysteresis air cell construction.

Especially Indicated for exposed application to eliminate air-conditioning shock and to reduce cooling load.

REFLECT-O-CELL is used in the production of insulated Buses, Trucks, Trailers, Ranges, Heaters and others. Similar satisfactory service has also been rendered in residences, industrial buildings and processing equipment.

REFLECT-O-CELL is supplied in conveniently scored continuous Roll Sheets or in Packs and is easily installed between wall studs, ceiling joists and roof rafters by stapling through its flanged edges.

Weighs only 38 lb per 1000 sq ft.



Single layer wall application.
Also installed in two layers.

Armstrong Cork Company

Building Materials Division

Lancaster, Pennsylvania

Offices

Albany	Cincinnati	Indianapolis	Omaha	**In Canada**
Atlanta	Cleveland	Jacksonville	Philadelphia	
Baltimore	Columbus	Kansas City	Pittsburgh	Montreal, Que.
Boston	Dallas	Louisville	Richmond	Toronto, Ont.
Buffalo	Des Moines	Milwaukee	Rochester	Winnipeg, Man.
Charlotte	Detroit	Minneapolis	St. Louis	
Chicago	Houston	New York	Washington, D. C.	

Distributors

Appleton, Wis.	Northwestern Asbestos and Cork Insulation Co.
Boise, Idaho	Reilly Atkinson and Co.
Boston, Mass.	T. R. Nunan Company
Charleston, W. Va.	Capital City Supply Company
Denver, Colo.	Stearns-Roger Manufacturing Company
Eau Claire, Wis.	Horel-George Company
Fort Wayne, Ind.	Asbestos & Asphalt Products Co.
Glendale, N. Y.	Charles M. Trojahn
Grand Rapids, Mich.	Thomas Batenburg
Jamestown, N. Y.	Laco Roofing & Asbestos Company
Joplin, Mo.	Joplin Cement Company
Kingsport, Tenn.	Kingsport Lumber & Supply Co.
Little Rock, Ark.	Fischer Cement & Roofing Company
Los Angeles, Calif.	Van Fleet-Freear Company
Manitowoc, Wis.	Northwestern Asbestos and Cork Insulation Co.
Memphis, Tenn.	Grant Brothers, Inc.
New Orleans, La.	H. T. Steffee
Portland, Oregon	Asbestos Supply Company
Providence, R. I.	Rhode Island Covering Company
San Antonio, Tex.	General Supply Company, Inc.
San Francisco, Calif.	Van Fleet-Freear Company
Seattle, Wash.	Asbestos Supply Company
South Bend, Ind.	Asbestos & Asphalt Products Co.
Spokane, Wash.	Asbestos Supply Company
Springfield, Mass.	Johnson Asbestos Company
Springfield, Mo.	Southwestern Insulation Company
Tacoma, Wash.	Asbestos Supply Company
Terre Haute, Ind.	The Hartmann Company
Tulsa, Okla.	Kelley Asbestos Products Company
Wichita, Kans.	Ludeman Insulations Company

For detailed technical information, samples, and descriptive literature, ask any office or representative. Complete specifications appear in Sweet's Architectural Catalog.

PRODUCTS—**Armstrong's Corkboard, Cork Covering, Vibracork, Corkoustic, Temlok, Temcoustic, Insulation Sundries.**

Corkboard

Insulating Efficiency

The thermal conductivity of Armstrong's Corkboard is 0.27 Btu per hour, per degree temperature, per inch thickness at 60 F mean temperature.

The value of adequate and efficient insulation is covered in Chapter 3 of this book and the tables on pages 84 to 96 indicate the savings which can be effected by using 1½ in. or 2 in. of corkboard in standard wall and roof construction.

For air conditioning work use 1 in., 1½ in., or 2 in. corkboard insulation on ducts, fans, dehumidifiers, and similar equipment. Erect corkboard in hot asphalt and tie with wires or bands 9 in. on centers or erect with Armstrong's Cements, using ties as above.

Sizes and Thicknesses

Armstrong's Corkboard is furnished in rigid boards 12 in. x 36 in., 18 in. x 36 in., and 24 in. x 36 in., in several thicknesses: 1 in., 1½ in., 2 in., 3 in., 4 in., and 6 in.

Cork Covering

Armstrong's Cork Covering is made of pure cork in sizes to fit all standard pipe sizes. The inside surfaces of each piece are machined to assure an accurate fit, free from moisture-catching air pockets. Cork covering is rigid and will not sag. Thicknesses are: Ice Water (1.20 in. to 1.93 in.); Brine (1.70 in. to 3.00 in.); and Special Thick Brine (2.63 in. to 4.00 in.).

Armstrong's Fitting Covers are rigid and are designed to fit accurately all types of standard ammonia and extra heavy fittings, screwed, flanged, and welded.

Vibracork

Armstrong's Vibracork, made in three densities, is ideal for the elimination of noise and vibration transmission and is of primary importance in air conditioning work. It does not take a set, is not affected by atmospheric moisture, and will not deteriorate under usual service.

For aid in the solution of any technical problems involving insulation, isolation, or acoustical treatment, and for literature and prices, get in touch with an Armstrong district office or distributor or the Armstrong Cork Company, Building Materials Division, Lancaster, Pa.

Conforms in all details to Federal Specification HH-C-561a, March 9, 1939

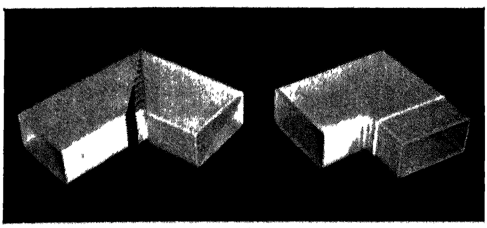
The Philip Carey Company

Manufacturers of Heat Insulation and Asbestos Products

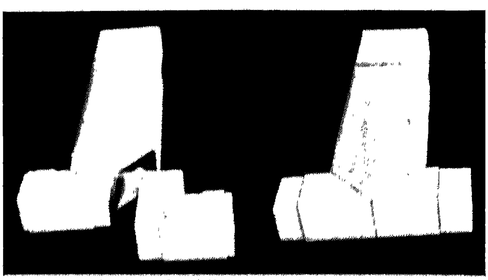
Lockland **Cincinnati, Ohio**

Sales Offices

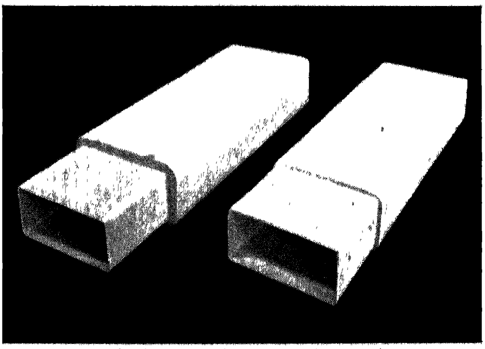
Atlanta, Ga.
Baltimore, Md.
Boston, Mass.
Buffalo, N. Y.
Charlotte, N.C.
Chattanooga, Tenn.
Chicago, Ill.
Cincinnati, Ohio
Cleveland, Ohio
Columbus, Ohio
Dallas, Tex.
Dayton, Ohio
Detroit, Mich.
Indianapolis, Ind.
Kansas City, Mo.
Los Angeles, Cal.
Louisville, Ky.
Minneapolis, Minn.
New York, N.Y.
Philadelphia, Pa.
Pittsburgh, Pa.
Richmond, Va.
St. Louis, Mo.
Seattle, Wash.
Wheeling, W. Va.



Standard 90 deg Elbow assembly. Left: Core opened to show duct vanes. Right: The completed fitting.



Two standard 90 deg Elbows nested in a larger standard section to form a tee.



Standard 1 in. and ½ in. thick Careyduct sections with core extended.

Careyduct is a new prefabricated insulated duct built entirely of asbestos. The double layer construction consists of an inner core of hard, rigid asbestos, and the outer jacket is made of multiple layers of a fine corrugated asbestos structure. The combination results in great strength, is an excellent insulator, and has a definite sound deadening effect.

Careyduct fittings are made from standard sections of duct, and may be made in the field with comparative ease by men without special training. A simple mitre cut plus a few standard accessories make a complete fitting thus keeping costs at a minimum. Prefabricated fittings may be ordered from the factory if desired.

The telescopic assembly method practically eliminates leaks that are commonly found in other construction.

The standard sizes of Careyduct are designed so that a combination of smaller sizes will exactly nest in a larger size. All tees and take-offs are a combination of ells and straight duct.

Grilles and dampers are installed according to the accepted standard practice. Careyduct gives high insulating value. It materially reduces the transmission of extraneous and equipment noises. Careyduct costs decidedly less than properly insulated metal duct and compares very favorably with sheet metal duct of standard quality.

For more detailed information write for Catalog and Erection Manual.

The Eagle-Picher Lead Company

General Offices: Temple Bar Building, **Cincinnati, Ohio**

Offices in Principal Cities

A Remarkable Insulating Wool Made From Minerals

Years ago Eagle-Picher pioneered a method of fusing and fiberizing carefully selected minerals into a dark gray insulating wool. This mineral wool is chemically inert. Fibers are mechanically strong, extremely resilient and flexible. They withstand expansion and contraction without loss of efficiency even at elevated temperatures.

From this mineral wool, Eagle-Picher has fabricated a long list of insulating products to meet a wide range of temperatures and operating requirements.

Eagle H-2 Loose Wool

A clean fill insulation that is highly efficient for temperatures to 1200 F. Averages considerably lighter in weight than many rock and slag wools goes farther. Fibers are soft and flexible. Approved by Underwriters Laboratories as fireproof and a non-conductor of electricity. Retains physical and chemical stability in presence of water. Packed in 40-lb. bags.

Eagle 7-B Granulated Wool

Another grade of fill insulation that has all the advantageous properties of Eagle H-2 Loose Wool. It consists of small pellets averaging 1/8 to 1/2 in. in size. For all fill jobs in irregular spaces. May be poured. Packed in 40-lb. bags.

Eagle Low Temperature Felt

A highly efficient insulating material for subzero and low temperatures (to 400 F). Inherently water-repellent (not specially treated). Available in densities 4-lb to 8-lb per cu ft. Recommended for refrigerator rooms, trucks, refrigerators, stoves, etc. Extensively used in marine field.

Eagle H-5 Felted Pads

Specially designed for use in pre-fabricated equipment where high insulating efficiency is required. Offers low conductivity in easy to apply form. Pads come rolled in paper, 6 to 12 ft in length. Thicknesses 1 1/2 in., 2 in. and 2 1/2 in. Standard width is 24 in.

Paper Encased Batts and Blankets

These light-weight, sturdily constructed batts and blankets are easy to apply. Enclosed on four sides with paper, one side of which is an approved vapor barrier. Strong tacking flanges. Quickly cut with knife or shears. Three thicknesses Ful-Thik, Semi Thik and 1-in.

Eagle Super "66" Cement

A high temperature plastic insulation. Easy to apply and trowels to a smooth finish. Does not cause corrosion. Will stick on any clean, heated surface. Dry coverage 50-55 sq ft per 100 lbs. 100 per cent reclaimable up to 1200 F. Packed in 50-lb bags.

Eagle Supertemp Blocks

An all-purpose block insulation which will withstand elevated temperatures up to 1700 F without loss of efficiency or structural strength. Blocks are water-repellent. Light weight. Easily cut to fit irregularly shaped surfaces. Blocks resist attacks of steam and moisture, and withstand all normal vibration and abrasion. Available in all standard sizes.

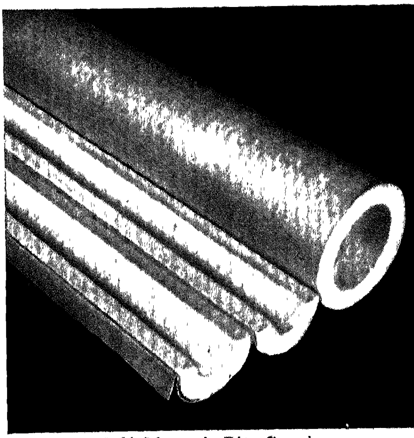
Eagle Insulseal

A protective coating for Blankets, Supertemp, "66" Cement and other kinds of heat insulation. Provides a permanent seal that safeguards insulation against air infiltration, moisture, water, fumes; also against vibration and abrasion. Does not support combustion.

For more complete specifications and technical data on these and other Eagle Insulating Products, see Sweet's Engineering or Power Plant catalogs.

Ehret Magnesia Manufacturing Co.

Valley Forge, Pa.



85% Magnesia Pipe Coverings

A FULL RANGE OF INSULATIONS FOR HEATING AND VENTILATING

The Ehret Company furnishes a broad range of thermal insulations for practically every industrial and architectural requirement. For full details of Ehret products, see the Ehret Insulation Manual.

Ehret's 85 Per Cent Magnesia

Known for nearly half a century in the industrial field, Ehret's 85 per cent Magnesia Pipe Coverings and Blocks are efficient, economical and they last indefinitely. Pipe coverings are available in a full range of sizes and thicknesses, and blocks can be furnished in thicknesses up to 4 in. An ideal material for use on heated pipes or surfaces whose temperatures do not exceed 600 F.

OTHER HEAT INSULATIONS

In addition to 85% Magnesia insulation, the Ehret Company furnishes a full line of other heat insulating materials, in the forms of pipe coverings, flat and curved blocks, sheets, lagging, blankets, cements and loose fills. These materials include Enduro (high temperature), asbestos cellular, asbestos sponge felt, mineral wool and many other products for use on heated pipes and surfaces.

COLD INSULATIONS

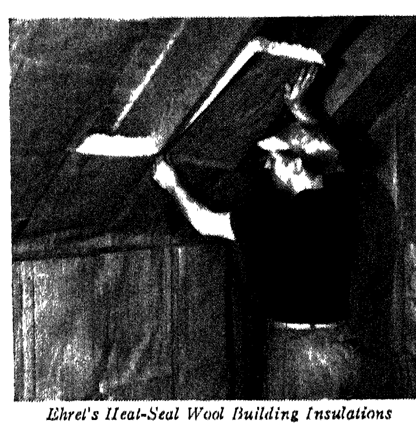
Ehret insulations for use on cold pipes and surfaces are made in a variety of forms and materials. Pipe coverings include cork, wool felt, frostproof and anti-sweat. Standard Hair Felt, Punched Hair Felt and Insultelt, in roll form are used to insulate both curved and flat surfaces. Ehret's Eroduct is a special material in ½ in. thickness that is applied to air conditioning and cold air ducts. Cork blocks, sheets and discs, as well as granulated cork are also furnished.

BUILDING INSULATIONS

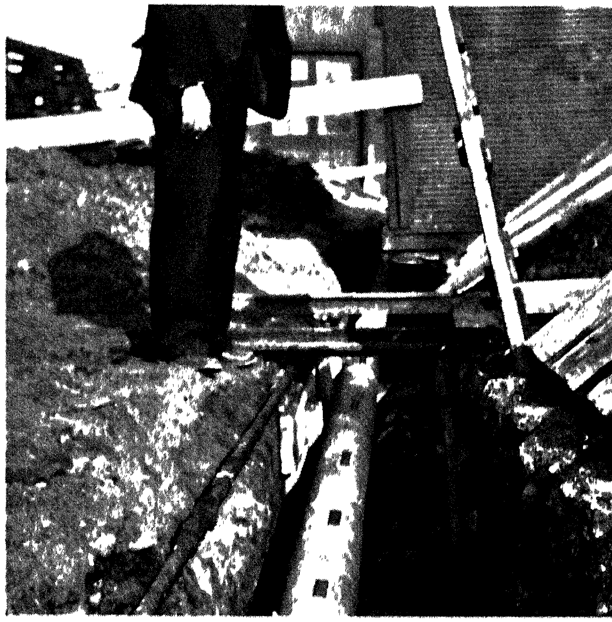
For insulating the walls, floors and ceilings of buildings, Ehret's Heat Seal Wool is made in batts, strips, loose and granular forms. This material is high in insulating efficiency, is easy to install or apply and it will last indefinitely. Batts can be furnished with or without paper backing, as desired.

OTHER EHRET PRODUCTS

In addition to the insulations themselves, the Ehret Company can furnish insulation accessories such as water-proofing compounds, weathertight jackets, bands, wires, adhesives, sewing canvas, asbestos paper, wallboard and many other materials required for the application of insulations. Ehret packings and asbestos products as well as Durant Insulated Pipe (which is briefly described on the opposite page) are fully treated in the Ehret Insulation Manual. Write today for your copy of this 280 page handbook.



Ehret's Heat-Seal Wool Building Insulations



Two [illegible] in. steam lines of Durant Insulated Pipe, being installed one above the other in a narrow trench.

EHRET'S DURANT INSULATED PIPE

. . . *for Underground and Outdoor Service*

This unique system of pipe line protection consists of pipe that is insulated, sealed and protected *at our factory*, and shipped to the job ready for installation. Pipe lengths can be joined with screwed, flanged or welded fittings, and the system provides protection for expansion bends, joints, valves and similar pipeline appurtenances.

Field joints in Durant Insulated Pipe are easy to make, and once made the backfill can be begun and the trench flooded for tamping.

Ehret's Durant Insulated Pipe will not crack or leak and moisture or water is permanently excluded by the thick, time-defying layer of high-melting-point asphalt that encloses all parts of the system. Write for the special Ehret D.I.P. folder — it gives full details.

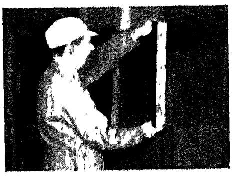
Some Outstanding Advantages of Durant Insulated Pipe:

1. Permanently waterproof.
2. Elimination of electrolysis and corrosion.
3. Requires no sub drains as even complete water submersion does no harm.
4. In multiple lines, individual Durant pipes can be added, removed or replaced without disturbing others.
5. Minimum trenching and field work.
6. No rollers or pipe supports required.
7. No breakage or waste of material during installation.
8. Tile or masonry protection not required.
9. Field costs are much lower than those of tile, tunnel and similar systems.
10. Insulation protection is absolutely dependable.

General Insulating & Mfg. Company

Engineering Offices and Main Plant: ALEXANDRIA, INDIANA | Executive Offices: ST. LOUIS, MISSOURI | Branch Plants: DOVER, N. J. DUBUQUE, IOWA

ROCK WOOL *Gimco* INSULATION



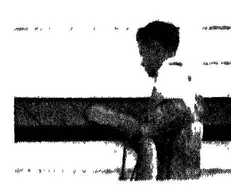
The GIMCO Super Sealal Bat is a completely wrapped "packaged" insulation. The Standard Sealal Bat is furnished with a vapor proof paper backing on one side. Sealal Bats fit between standard studdings and joists and are furnished in one, two and three inch thicknesses.

Gimco Sealal Rock Wool Bats Gimco Rock Wool Bats are made from long, tough rock wool fibres annealed and treated specially by the patented Gimco process. Installed 3⅝ in. thick Gimco's conductivity is only 0.067 Btu per hour per sq ft for that thickness. Gimco provides full "wall-thick" protection . . . keeps inside temperatures as much as 15 deg cooler in summer and pays for itself out of winter fuel savings. Gimco is as fireproof as the rock itself, resists moisture, and will not decay, pack down or dust out. Gimco is as permanent as the house, and offers no attraction to vermin or termites.

Gimco Bats are Self-Supporting—Gimco bats need only to be pushed between studdings or joists. Their own natural resiliency holds them permanently in place without additional support. Application costs are thus cut to a minimum.

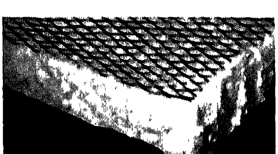
Gimco Insulation for Present Homes—Gimco in granulated form can easily be blown into empty wall and ceiling spaces. It makes a permanent "wall-thick" insulation, and can be installed in any home, regardless of age, size or type of construction. For complete details, write for new book on home insulation.

Specifications—Gimco Sealal Bats are furnished with waterproofed paper backing. They are made in three sizes: (1) 15 x 18 in. x wall thick; (2) 15 x 23 in. x wall thick; (3) 15 x 48 in. x wall thick. Ten small bats insulate approximately 20 sq ft of wall or ceiling area; 10 medium size bats insulate approximately 25½ sq ft; 10 large bats insulate approximately 55 sq ft.

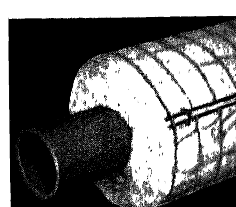


Present homes are easily and quickly insulated by blowing Gimco Rock Wool in empty wall and ceiling spaces. Insures a thick, protective blanket of uniform density.

GIMCO ROCK WOOL PRODUCTS FOR INDUSTRY



Flexfelt Blankets—Efficient and adaptable rock wool insulation for boilers, heaters, furnace breachings, hot tanks and other industrial equipment. Standard sizes: 2 ft x 4 ft and 2 ft x 8 ft. Special sizes are also made to order or as planned by our engineers from purchasers' equipment drawings.



Gimco Flexfelt Pipe Coverings—Flexfelt coverings are suited for every type of pipe insulation . . . hot water, steam process, cold water and refrigerant lines. They are made in a variety of sizes and styles, each individually designed to do an efficient insulating job.

Gimco No. 330 Insulating Cement can be easily and quickly applied to any surface. It dries quickly with little shrinkage and has a smooth, hard surface. Cost per square foot coverage is unusually low.

Free Construction and Engineering Service—Gimco engineers, backed by years of research in our own laboratories, render prompt and efficient service in helping you solve insulation problems. Quotations and suggestions on any job (temperatures up to 1500 F) will be gladly given.

10807 Lyndon
at Meyers Road

IN SULATION DUSTRIES CORPORATED DETROIT

Detroit
Michigan

ROCK WOOL INSULATION PRODUCTS

BUILDING INSULATION PRODUCTS
Loose Rock Wool (paper bags)
Granulated Rock Wool (paper bags)
Rock Wool in Rolls (any length or thickness)
Rock Wool Batts (cartons)
(with or without paper backs)
Rock Wool Batts (bags)
(without paper backs)

INDUSTRIAL INSULATION PRODUCTS
For
Stoves and Ranges
Water Heaters
Industrial Ovens
Bakery Ovens
Large Diameter Pipes
Boiler Settings, etc.

Insulation Industries Incorporated owns and operates one of the most modern, up-to-date Rock Wool plants.

Rock Wool is manufactured by a patented, precision process that produces a superior grade of Rock Wool. It is light in weight, has long, silky and resilient fibers. It is clean and free from foreign particles.

Rock Wool is indestructible and will last as long as the building itself. It is fire proof, vermin and rodent proof and is resistent to moisture.

BUILDING INSULATION

Rock Wool is suitable for all types of building insulation requirements.

It can be applied in the granulated form by the pneumatic method to existing homes or buildings.

For new construction or for unfinished attic or wall spaces, Batts are furnished either 15 x 23 in. or 15 x 48 in. and 2 or 4 in. thick and with or without paper backs, packed in cartons.

Long fiber Rock Wool in loose form is available packed in 35 lb paper bags.

RESULTS

Results obtained in all types of buildings, both old and new, show substantial savings in fuel consumption with elimination of drafts and variation of temperatures between rooms and floors.

BLANKETS

Long fibered, especially treated Rock Wool, felted and secured between metal fabrics of different types. These blankets are made in standard sizes 24 in. x 96 in. and 24 in. x 48 in. and special sizes as required and any thickness from 1 in. to 8 in. Applicable to flat or curved surfaces.

INSULATING BLOCK

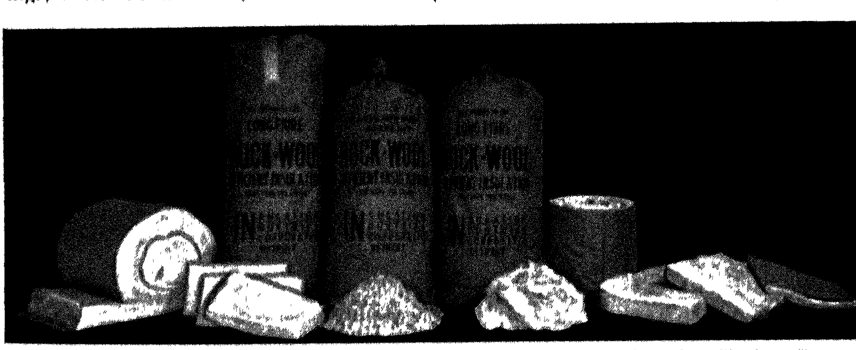
Rock Wool fabricated into sheet or board form from ½ in. to 4 in. thick, 24 in. x 36 in. or special sizes, as required. This block is widely used for insulating boilers, ducts, tanks, stills, etc., and for domestic furnaces, boilers, ranges and hot-water tanks.

INSULATING CEMENT

For finishing block and blanket insulation. For temperature conditions from 100 to 2000 deg. Is very plastic and is quickly and easily applied.

SPECIFICATIONS

Write for complete information and details on Insulation Industries products.



Rolled wool any thickness — *Batts in bags 8" x 15" x 4"* — *Granulated Wool* — *Long Fiber Loose Wool* — *Paper Backed Batts* — *Blanket—Paper both sides*

INSULITE

General Offices

1100 Builders Exchange, **Minneapolis, Minnesota**

Sales Solicitors Offices

New York—101 Park Avenue

Chicago—205 W. Wacker Drive **St. Louis—1206 S. Vandeventer**

TWENTY-SEVEN YEARS PROVEN DURABILITY

For 27 years engineers and architects have specified Insulite materials for structural uses, interior finish, duct lining, and for other thermal insulation and sound control work. Insulite materials have proved themselves practical through their performance on the job.

STRUCTURAL MATERIALS

Lok-Joint Lath—An insulating plaster base, fabricated from Ins-Lite or from Graylite. Patented "Lok" firmly locks the sheets between supporting members. Thicknesses: ½, ¾ and 1 in. Size: 18x48 in.

Sealed Graylite Lok-Joint Lath—An insulating plaster base of Graylite, sealed on stud space side (seal in center of 1 in. thickness) with an effective vapor barrier. Has patented "Lok" on long edges. Furnished in same thicknesses and size as Ins-Lite and Graylite Lok-Joint Lath.

Bildrite Sheathing is an asphalt-containing wood fiber board manufactured under an exclusive process which provides increased strength and moisture resistance. It is $^{25}/_{32}$ in. thick and has a distinctive gray-brown color. Thermal conductivity: 0.36 Btu per inch thickness. Each sheet is marked to indicate proper nail spacing. Available in sizes 4 x 8 ft up to 4 x 12 ft with all edges square. Also available in 2 x 8 ft size with interlocking joint on long edges. Used as a structural sheathing board and as a roof boarding.

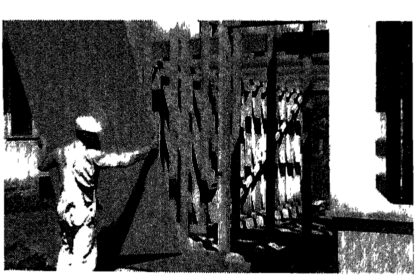
Condensation Control—Where low outside temperatures and high inside humidities may occur, authorities recommend "sealing the hot side and ventilating the cold side" of the wall to prevent condensation. An adequate vapor barrier, Sealed Graylite Lok-Joint Lath, should be used on the warm (room) side of the wall thereby effectively reducing vapor transmission into the stud space. Bildrite Sheathing is designed to allow any surplus vapor in the stud space to "breathe" or be vented harmlessly to the exterior air. If vapor is trapped within the stud space and cannot escape through the sheathing, destructive condensation may occur.

INSULITE WALL OF PROTECTION

This construction consists of Bildrite Sheathing on the exterior of the frame work and either Lok-Joint Lath or Insulite Interior Finish Materials on the interior. Transmission coefficients (U) are shown below.

Exterior Finish and Sheathing	Interior Finish — No Insulation Between Studding: No plaster—Insulite Building Board, Interior Board, Tile Board, or Plank (½ in.)	Plaster (½ in.) on Lok-Joint Lath (½ in.)	Plaster (½ in.) on Lok-Joint Lath (1 in.)
Wood Siding, $^{25}/_{32}$ in. Bildrite Sheathing	0.16	0.15	0.13

The above values are typical of results which can be obtained by utilizing Insulite materials in frame construction. For further (U) values refer to Chapter 3, pages 88 and 89.



Applying Bildrite Sheathing

Applying Lok-Joint Lath

INTERIOR FINISH MATERIALS

Ins-Lite Building Board A wood fiber board with the light color of natural wood burlap and linen textured surfaces. Thermal conductivity: 0.33 Btu/hr/sq ft/in. °F; density: 16 lb/cu ft. Furnished in thicknesses of ½, ¾ and 1 inch and sizes of 4 x 4 ft to 4 x 12 ft. Also available in 6 x 8 ft, 6 x 12 ft and 8 x 12 ft sizes for ½ in. and ¾ in. thicknesses.

Graylite Building Board An integrally treated asphalt containing wood fiber board of grayish brown color burlap and linen textured surfaces. Thermal conductivity 0.35 Btu per inch thickness. Furnished in same thicknesses and sizes as Ins-Lite Building Board.

Smoothcote Interior Board Coated Insulating Board with smooth, hard surface one side, having 68 per cent light reflection. Furnished in ½ inch thickness only and in sizes of 4 x 4 ft to 4 x 12 ft.

Satincote Interior Board Factory finished Insulating Board in colors buff, gray, coral and green. Light reflection from 64 per cent for green to 80 per cent for the buff color. Requires no further decoration. Highly resistant to abrasion and easily washable. In ½ inch thickness and in sizes of 4 x 4 ft to 4 x 12 ft.

TileBoard Available in Ins-Lite, Graylite, Smoothcote and Satincote. TileBoard is furnished with the Lok-Grip Joint that permits concealed nailing and which together with the Lok-Pin (a flat diamond shaped metal dowel) definitely and mechanically safeguards against any falling units even though no face nailing is used. Sizes of TileBoard, 24 in. x 24 in. and smaller, also available with a butt joint and beveled on edges.

Ins-Lite and Graylite TileBoard available in ½ or 1 inch thicknesses and sizes of 12 x 12 inches to 24 x 48 inches. Smoothcote and Satincote TileBoard also available in above sizes in ½ inch thickness only.

Plank Available in Ins-Lite, Graylite, Smoothcote and Satincote. Plank has the Lok-Grip joint which permits concealed nailing and is beveled and beaded both long edges. Ins-Lite and Graylite Plank furnished in ½ and 1 inch thicknesses, widths of 6 to 16 inches and lengths of 8 to 12 ft. Smoothcote and Satincote Plank furnished in ½ inch thickness only and in above sizes.

Acoustilite or Fiberlite effectively quiet and control sound

Acoustilite A high efficiency acoustical material for sound control. Coefficient of sound absorption, at 512 cycles, is 0.79 when mounted on solid background and 0.80 when on furring strips. Noise reduction coefficient is 0.65 when mounted on solid background and 0.75 when on furring strips. Factory painted in buff, (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ¾ in.; sizes, 12x12 in. to 16x32 in.

Fiberlite An efficient sound absorptive and decorative material. Coefficient of sound absorption, at 512 cycles, is 0.53 when mounted on a solid background and 0.72 when on furring strips. Noise reduction coefficient is 0.55 when mounted on solid background and 0.65 when on furring strips. Factory painted in buff (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ½ in.; sizes, 12x12 in. to 16x32 in.

HardBoard Products

HardBoard materials are tough, durable, grainless, pressed wood fiber boards with a hard, smooth surface. Available in a range of densities from 37 to 68 lb/cu ft. Thicknesses are from 1/10 to 5/16 in. and sizes of 4 x 2 ft to 4 x 12 ft.

Industrial Insulation

Industrial Insulation is a wood fiber board for use in all types of manufacturing industries producing items such as refrigerators, coolers, showcases, brooders, partitions and cabinets.

It can be cut-to-size and fabricated to customer's specifications. Three types of industrial board are available.

Lowdensite Industrial Board A 10 to 14 lb density board with an average tensile strength of 100 lb/sq in. and an average conductivity of 0.30 Btu/inch thickness.

Ins-Lite Industrial Board A 14 to 18 lb density board with an average tensile strength of 250 lb/sq in. and an average conductivity of 0.33 Btu/inch thickness.

Graylite Industrial Board Differs from two above products in that it has an integral asphalt treatment which provides increased strength and moisture resistance as well as minimum thickness and linear expansion. A 16 to 20 lb density board with an average tensile strength of 350 lb/sq in. and an average conductivity of 0.35 Btu/inch thickness.

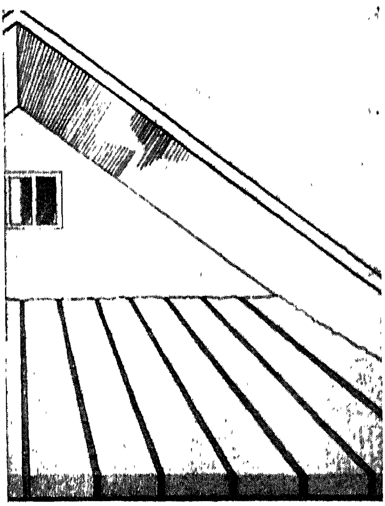
Insul-Wool Insulation Corp.

General Offices, **Wichita, Kansas**

Branches in Principal Cities

Manufacturers and Distributors of Insul-Wool

Insul-Wool is a fibre insulation of the "fill" type made of wood pulp, a natural insulating material. By the exclusive "Insul-Wool" method the wood pulp is converted into a loose fluffy substance which, when installed in a building, forms a soft heat-resisting blanket having millions of tiny air cells capable of resisting passage of either heat or cold.



Insul-Wool Applied over Ceiling

UNIFORMITY OF PRODUCT

Only one grade of Insul-Wool is made and every "run" is tested at the factory to insure uniformity of product and unvarying high quality. It is free from grit, silicon particles, or "shot."

FIRE PROOF AND VERMIN PROOF

A special "Insul-Wool" method of chemical treatment makes Insul-Wool thoroughly vermin proof and fire proof. It has been approved by the National Board of Fire Underwriters.

"INSUL-WOOL" SERVICE

Insul-Wool is distributed and installed only by specially trained men direct factory representatives or men in the organizations of the largest insulation material dealers throughout the United States.

ADVANTAGES OF INSUL-WOOL

1. It is made from wood pulp, a natural insulating material.
2. Chemical treatment makes Insul-Wool safe under all conditions and hazards.
3. Approved by the National Board of Fire Underwriters.
4. Its light weight of 2.5 lb to the cubic foot adds very little load to the ceiling rafters.
5. Does not pack or settle and outlasts the building in which it is installed.
6. Does not draw moisture.
7. Cuts fuel costs and reduces Summer temperatures, indoors.
8. Meets U. S. Government requirements of Federal Construction with a thermal conductivity of 0.24 Btu per hour, per square foot, per degree Fahrenheit, per inch thickness.

Analysis of Insul-Wool in Terms of Commerical Thickness

Material	Commercial Form	Comm'l. Thickness Inches	D. Wt. Per cu. ft.	C. Conductivity
INSUL-WOOL	Wood Fiber-Loose Type, Fire proofed and Germ proofed.	1 4	2.5	0.24* 0.067**

*Kansas City Testing Laboratory, Inc., February 25, 1938.
**J. C. Peebles, Armour Institute of Technology, April 8, 1937.

Complete data on Insul-Wool Insulating Product will be sent upon request.

International Fibre Board Limited

Sales Offices

OTTAWA, MONTREAL, TORONTO. Administrative Offices and Mills: GATINEAU, QUEBEC.

London Office: THE TENTEST FIBRE BOARD CO., LIMITED

75 CRESCENT WEST, HADLEY WOOD, BARNET, HERTFORDSHIRE, ENGLAND.

TEN/TEST

INSULATING BUILDING BOARD

TEN/TEST is a manufactured lumber made from spruce fibres, solidly pressed under hydraulic pressure into a strong, homogeneous board. The fibres are chemically treated and water-proofed during process of manufacture, until the insulation is non-hygroscopic, free from capillary attraction and moisture-resisting in service commensurate with the maximum degree of insulation obtainable.

TEN/TEST Products

TEN/TEST Insulating Building Board. Standard insulation for use as exterior sheathing, interior finish; between walls and under floors for sound deadening. Standard Industrial Insulation for refrigeration and the prevention of condensation. Manufactured in convenient sizes: 4 ft wide and up to 17 ft long, ½ in. to 1 in. thick or laminated to any desired thickness.

TEN/TEST Notch Board Plaster Base. Insulating plaster base having tongue and groove interlocking joints. Provides an effective bond with plaster without use of metal lath at joints. Sizes: 16 in. wide; 32 in. and 47¾ in. long. Thicknesses from ½ in. to 1 in. or laminated to any desired thickness.

TEN/TEST Roof Board. An effective roof insulation. Manufactured in two sizes: 1 x 4 ft and 2 x 4 ft. Thicknesses from ½ in. to 1 in. or laminated to any desired thickness.

TEN/TEST Ashlar Block. For interior decoration and acoustical correction. Can be supplied in a variety of designs and sizes to harmonize with any decorative treatment.

TEN/TEST Acoustical Tile and Panels with sound absorption coefficients ranging up to 0.53 at 512. Specially designed for churches, schools, auditoriums, theatres, etc.

TEN/TEST Moulded and Shiplap Edge Wall Panels. Conceals joints and provides excellent decorative treatment. Featured in widths of 11 in. to 47¼ in., lengths up to 12 ft.

TEN/TEST Mouldings. An effective trim and finish for joints, corners, etc. Available in widths of ¾ in. to 10 in. and lengths up to 12 ft.

HYDRO/TEST. Water proof, insulating building board, designed particularly for low temperature requirements.

Official Tests

Conductivity. TEN/TEST has a conductivity of 0.33 Btu per hour per square foot per degree Fahrenheit per 1 in. thick. Authority: Professor E. A. Allcut, M. Sc. M. I. Mech. E. Mem. A.S.M.E. Professor of Applied Mechanics, University of Toronto. Tests performed by Hot-Plate method. Mean temperature 47.8 deg.

Plaster Bonding Strength 2163 lb per sq ft. Brown and scratch plaster coats were applied to standard 7/16 in. board, and the pull registered in an Olsen Testing Machine. Authority: Columbia University Testing Laboratories, New York.

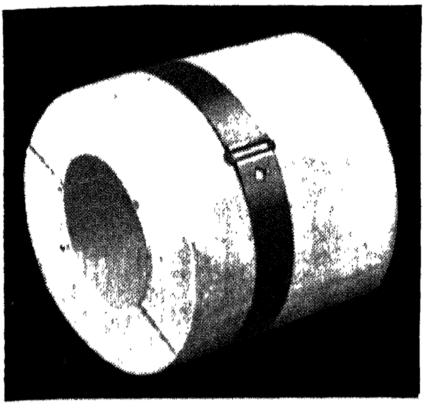
Moisture Resisting. TEN/TEST, after complete immersion in water for 24 hours, registered 37.5 per cent increase in weight.

Tensile Strength 228 lb per sq in. Tests made on 7/16 in. board cut to strips 1 in. wide and tested in a Riehle Tensile Testing Machine, the grips being 2 in. apart. 228 lb is the mean average of seven series of tests.

Transverse Strength (equal deflection) is 28.4 lb. Test made on 7/16 in. board, 6 in. wide, 18 in. long, on 12 in. centers, and load being applied to breaking point.

NOTE. Authority for tensile strength, transverse and moisture tests; J. T. Donald & Co., Ltd., Chemical Analysts and Engineers, Montreal, Que.

Johns-Manville Pipe and Boiler Insulation



J-M 85% Magnesia Pipe Insulation

J-M Pre-Shrunk Asbestocel Pipe Insulation

J-M Pre-Shrunk Asbestocel is a radically improved insulating material for hot water or low pressure steam piping, which, since it is made of moisture proofed asbestos paper, minimizes objectionable shrinkage.

Supplied in canvas, asbestos paper or aluminum finishes. All types furnished in 3-ft sections in standard thicknesses of 2 to 8 plies, each ply approximately ¼ in. thick, for all commercial pipe sizes.*

J-M 85% Magnesia

Recommended as the most widely used insulation of the molded type for temperatures up to 600 F. Pipe insulation is furnished in sectional or segmental form for all commercial pipe sizes,* in thicknesses up to 3 in. Blocks are 3 in. by 18 in. and 6 in. by 36 in., flat or curved, from 1 in. to 4 in. thick. Minimum thickness for curved blocks, 1¼ in.

J-M Pre-Shrunk Wool Felt Pipe Insulation

Due to its Dual-Service Liner an asphalt-saturated felt - J-M Pre-Shrunk Wool Felt is equally effective and durable on either hot or cold water service piping. By the use of waterproofed felts, shrinkage troubles have been minimized.

Supplied in two finishes, the regular canvas and a smooth, dull-coated aluminum. In either finish, it is furnished in 3-ft sections in thicknesses of ½ in. ¾ in., 1 in., Double ½ in., and Double ¾ in., for all commercial pipe sizes.*

J-M Asbesto-Sponge Felted Pipe Insulation

Recommended on all high pressure steam piping at temperatures up to 700 F where insulation may be subjected to rough usage or where maximum efficiency and durability are desired. Furnished in 3-ft sections up to 3 in. thick, for all commercial pipe sizes.

J-M Superex Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 F. Superex and Magnesia are both furnished in sectional and segmental pipe covering, and in block forms.

J-M Asbestocel Sheets and Blocks

Asbestocel Sheets and Blocks are used for insulating warm-air ducts, flues, heater casings and fan housings in the ventilating system. Temperature limit 300 F. Furnished 6, 9, 12, 18 and 36 in. wide by 36 and 72 in. long, from ½ in. to 4 in. thick.

J-M Rock Cork Sheets and Pipe Insulation

J-M Rock Cork is made of mineral wool and a moisture-proof binding ingredient molded into sheets for insulating refrigerated rooms and air conditioning ducts; and into sectional pipe insulation with an integral waterproof jacket, for all low temperature service. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance, its high insulating efficiency is maintained in service.

Furnished in sheets 18 in. by 36 in., in 1½, 2, 3 and 4 in. thicknesses; also 18 in. by 18 in. by 1 in. thick. In lagging form, for curved surfaces, supplied 18 in. long by 1½, 2, 3 and 4 in. thick, 2 to 5 in. wide, depending on diameter. In pipe covering form, in ice water, brine and heavy brine thicknesses, for all commercial pipe sizes.

Details on Request

Write for complete information on any Johns-Manville insulating material.

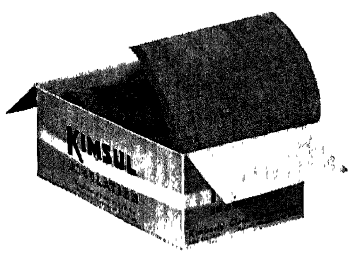
*Can also be supplied in sections to fit straight runs of copper pipe or tubing with outside diameter ⅝ in. and larger.

KIMBERLY-CLARK CORPORATION

ESTABLISHED 1872

Building Insulation Division

Neenah, Wisconsin



A Product of
KIMBERLY-CLARK CORPORATION

KIMSUL*, manufactured by the KIMBERLY-CLARK CORPORATION, is a wood fibre product, made in long, flexible blankets composed of many creped layers or plies, providing a maximum number of dead air cells for efficient insulation. Being flexible and extremely light in weight, it is marvelously easy to install. Each blanket is stitched with rows of strong twine running the length of the blanket. This unique feature holds the KIMSUL blanket securely in place—prevents sagging or "packing down" inside the walls.

KIMSUL is delivered in cartons containing sufficient KIMSUL to insulate 250 sq ft in Commercial thickness; or 125 sq ft in Standard thickness; or 83⅓ sq ft in Double Thickness.

UNIQUE PROPERTIES OF EXPANSION AND STITCHING

Expandability—Delivered to a job in a *compressed form*, each blanket of KIMSUL, when installed, is expanded to about 5½ times its original length without decreasing the intended thickness or lessening its heat-stopping ability.

Speeds Work—Lowers Cost—Because KIMSUL is delivered in compressed form it is easier to handle; storage costs are reduced—and it goes up quickly.

KIMSUL blankets are made in widths to exactly fit standard widths of stud spacing. To apply KIMSUL to side walls: using shipping carton as a dispensing container, the end of KIMSUL blanket is nailed to top plate—blanket expanded and attached at bottom—then cut off. That is all there is to it.

Stitching Controls Efficiency Each blanket of KIMSUL, before being compressed, is stitched its entire length with rows of twine approximately 20 times stronger than necessary to support its entire weight. This prevents KIMSUL from being expanded beyond most efficient density, prevents sagging, and holds KIMSUL securely in place.

PHYSICAL PROPERTIES

1. Thermal Efficiency KIMSUL's conductivity is 27 Btu hr/sq ft/°F/inch (J. C. Peebles) one of the most efficient heat "Blockaders" developed. This conductivity is at the extremely low installed density of 1.57 lb per cu ft . . . which density is maintained by the stitching feature.

2. Flexibility Flexible as a blanket, KIMSUL fits snugly. It can be tucked behind wire and piping, moulded to the shape of non-standard-size openings, pulled over or around corners, and packed into cracks, around doors and window frames.

3. Will Not Shred or Sift—Each blanket of KIMSUL is composed of many creped layers or plies stitched together.

FOR GREATER COMFORT . . . SUMMER AND WINTER

KIMSUL INSULATION

The fibres are fastened to one another in each ply to form a strong yet flexible web.

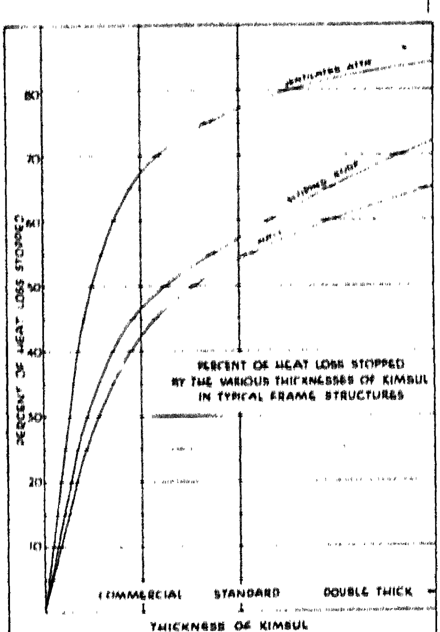
4. Moisture and Fire Resistant KIMSUL resists water and fire. Even when exposed to flame temperature of 2000 F. KIMSUL merely chars. It does not produce spontaneous combustion.

5. Lightness At Standard Thickness (1 in.) KIMSUL has an installed density of 1.57 lb per cu ft. Installed, 1000 sq ft of it weighs only 131 lbs. Thus KIMSUL adds relatively little weight to structural load of a building.

PROPER THICKNESS FOR AVERAGE REQUIREMENTS

To calculate what thickness of insulation will provide the desirable balance between costs and the benefits produced, it is necessary to have information concerning the specific job. The severity of the climate, the price of the fuel to be used are important considerations.

In most cases, however, it will be found that the standard one-inch thickness of KIMSUL is not only sufficient but that this thickness stops the greatest proportion of heat losses in winter, and the greatest proportion of heat infiltration in summer, *at the least cost.*



HEAT LOSSES THROUGH WALLS, ATTICS AND ROOFS

The effectiveness of KIMSUL in reducing heat flow varies somewhat, depending on type of structure insulated. The figure at lower left shows relationship between insulation thickness and the heat stopped, expressed as per cent of heat flowing through an uninsulated structure.

Note that depending upon the structure insulated, the per cent of heat stopped by Standard Thick KIMSUL varies from 54 per cent to 77 per cent.

When a normal frame wall is considered, 1 in. of KIMSUL stops 54 per cent of heat which would be lost through the uninsulated wall. By adding another inch of thickness this percentage is increased to 65 per cent so the second inch of insulation is responsible for stopping only an additional 11 per cent. If wall thick insulation is used, total heat stopped in only increased to about 73 per cent.

Looking at it another way by taking the maximum heat stoppage through wall insulation as 100 per cent 1 inch of KIMSUL stops 74 per cent of all the heat that can be stoppnd by insulation.

So it is evident that it is the first inch which does the most work.

Graph shows how effectively KIMSUL reduces heat flow through typical frame structures. Note that greatest proportion of heat losses are stopped by the first inch of KIMSUL.

It must, however, be borne in mind that stopping 75 per cent of the heat losses, by insulating walls and roof, does not mean a fuel savings of a like amount. Usually half of the wall area is made up of doors and windows and heat losses through them must also be considered.

*Reg. U. S. & Can. Pat. Off.

Mundet Cork Corporation

65 S. Eleventh St. **INSULATION DIVISION** **Brooklyn, N. Y.**

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork. Natural Cork Isolation Mats, Cork Tile and all kinds and varieties of Cork Specialties.

Mundet Branches

*ALBANY, N. Y.	CHICAGO, ILL.	*HOUSTON, TEXAS	PHILADELPHIA, PA.
*ATLANTA, GA.	*CINCINNATI, OHIO	*KANSAS CITY, MO.	*ST. LOUIS, MO.
*BROOKLYN, N. Y.	*DALLAS, TEXAS	*LOS ANGELES, CALIF.	*SAN FRANCISCO, CALIF.
BOSTON (NO. CAMBRIDGE), MASS.	*DETROIT, MICH.	*NEW ORLEANS, LA.	*SYRACUSE, N. Y.

[*Agencies for Keasbey & Mattison Asbestos and Magnesia Insulating Products for High Temperatures]

Mundet Distributors are Located in the Following Cities—Names and Addresses on Request

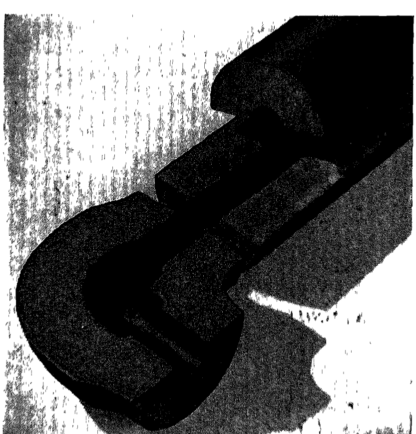
AMANA, IOWA	HARTFORD, CONN.	OKLAHOMA CITY, OKLA.	SEATTLE, WASH
BALTIMORE, MD.	JOHNSON CITY, TENN.	PORTLAND, OREGON	TUCSON, ARIZ.
BUFFALO, N. Y.	MEMPHIS, TENN.	PROVIDENCE, R. I.	TULSA, OKLA.
CHARLOTTE, N. C.	MINNEAPOLIS, MINN.	RICHMOND, VA.	UTICA, N. Y.
CLEVELAND, OHIO	NASHVILLE, TENN.	ROCHESTER, N. Y.	YOUNGSTOWN, OHIO
DENVER, COLO.	NORFOLK, VA.	SALT LAKE CITY, UTAH	

Mundet "Jointite" Corkboard

—for all low temperature insulation and for acoustical correction. 100% pure cork, fabricated in accordance with U. S. Government Master Specifications and unsurpassed in its field. Sold in standard 12 in. x 36 in. sheet. Standard thicknesses, ½ in., 1 in., 1½ in., 2 in., 3 in., 4 in., 6 in.

Mundet "Jointite" Cork Pipe Covering

Shown below, with fitting cover. Protects all types of low temperature lines. Made in 3 thicknesses, with complete line of standard covers, suitable for pipes carrying sub-zero to 50 F temperature.



Section of Mundet Moulded Cork Pipe Covering, with Fitting. The pipe covering is made in sections 36 in. long, to fit all sizes of pipes.

Mundet Cork Vibration Isolation

Machinery vibration encountered in heating and ventilating work is effectively controlled by the use of Mundet Natural Cork Isolation Mats. These consist of blocks of pure cork, held together within a rigid steel frame or bound with asphalt paper applied with hot asphalt top and bottom. Mundet steel bound mats are usually used under exposed mounts; asphalt paper bound mats under concrete foundations of the envelope type. Mats are made to fit under any type of machine foundation. For loads exceeding 2000 lb per square foot, we manufacture Mundet Machinery Isolation Cork, which is a board form of compressed granulated cork, available in 3 densities. All types of isolation are furnished in 1 in., 1½ in., 2 in., 3 in., 4 in., and 6 in. thicknesses, depending on class of service.

Above close-up of Mundet Natural Cork Isolation Mat shows how the blocks of cork are held together within a steel frame.

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers, to assist and advise in the preparation of specifications pertaining to cork. This service is also available without obligation to any one who has a low temperature insulation or a vibration isolation problem. Our complete catalogue is filed in Sweet's Architectural Catalogue and will be sent on request. It is replete with information and data of value to every specification writer whose field touches our products.

Mundet Contract Service

Covers the complete installation of our products, in accordance with best established practice. Divided responsibility is avoided. All materials and workmanship are guaranteed.

The Pacific Lumber Company

PALCO WOOL INSULATION

100 Bush Street, San Francisco — 35 E. Wacker Drive, Chicago — 5225 Wilshire Blvd., Los Angeles — 122 East 42nd St. New York

PALCO INSULATION WOOL

TRADE MARK REG. U. S. PAT. OFFICE

WHAT IT IS

PALCO WOOL is a loose fill insulating material made from the bark of the Redwood tree, the protective covering of the world's oldest living thing. It is highly refined into an insulating material of light weight, wiry fibres of springy resilience. Recent improvements in manufacturing have made it clean, dustless and lighter in weight. In practical use PALCO WOOL has proved to be ideal for all types of construction, large or small, where resistance to conduction of heat is required. It is continuously efficient and reasonably priced, thus assuring economical performance.

USES

PALCO WOOL is suitable for any type of domestic or commercial construction as well as for the various types of Cold Storage construction.

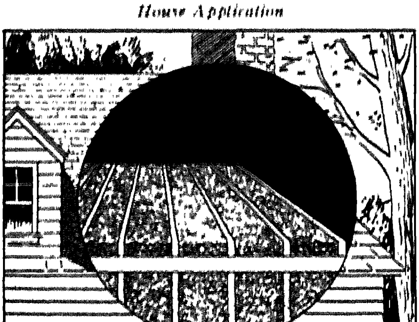
INSTALLATION

Approximately .8 of a lb of PALCO WOOL is required per square foot of 4 in. thickness. It is easily installed by hand or by machine. Between 100 and 150 lbs can be applied per hour per man. It comes in bales weighing approximately 100 lbs. Size 24 in. x 24 in. x 26 in.

Send for Insulation Manuals

Send for new 16-page booklets: "For Comfort Savings" on House Insulation or "Cold Storage Manual." Both give comparative charts and complete information on PALCO WOOL. Free sample on request.

House Application



8 PROPERTIES that make it AN IDEAL INSULATION

1. Thermal Efficiency: The established conductivity of PALCO WOOL is .26 Btu per hour per sq ft per inch of thickness per degree F difference in temperature by the Flat Plate Method.

2. Non-Settling: The fibres of PALCO WOOL possess such resilience that no settlement in a wall can occur under the most severe conditions of vibration.

3. Moisture Resistant: The fibres of PALCO WOOL are entirely lacking in capillarity and have little attraction for moisture, enabling it to remain dry and efficient when in use.

4. Permanent: The inherent antiseptic qualities of PALCO WOOL make the existence of fungus impossible. The fibres retain their resilience indefinitely.

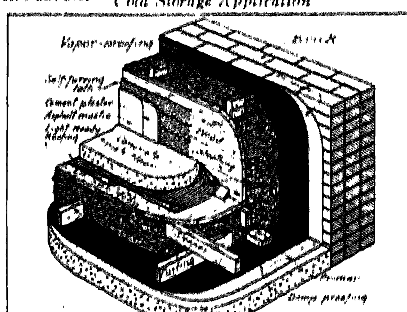
5. Vermin Repellent: PALCO WOOL is distasteful and repellent to rodents and insects.

6. Fire Resistant: PALCO WOOL, like the Redwood bark it comes from, is inherently fire resistant. As an additional protection it is *Saferized* to make it flameproof.

7. Odor Proof: PALCO WOOL is odorless itself and does not absorb or give off odors.

8. Economical: PALCO WOOL is light in weight and low in density, offering exceptional thermal efficiency per dollar invested.

Cold Storage Application



The Ruberoid Co.

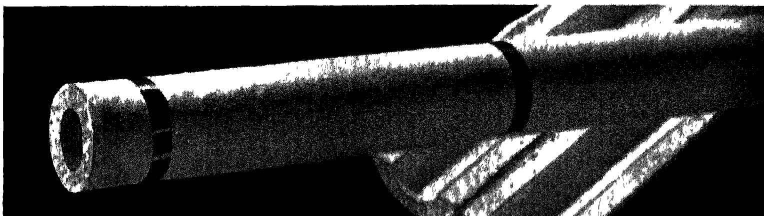
INSULATING PRODUCTS

Executive Offices

500 Fifth Avenue, **New York, N. Y.**

Divisional Offices

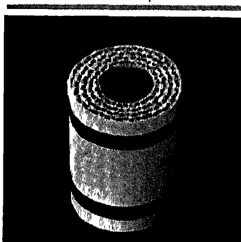
NEW YORK CHICAGO BOSTON (Millis) ERIE BALTIMORE MOBILE MINNEAPOLIS



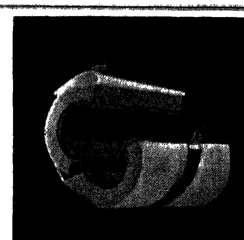
85 per cent Magnesia Sectional Pipe Covering. An accepted standard for general industrial pipe insulation.

The desire for increased efficiency of heating equipment as well as the need for fuel conservation prompts the engineer to seek the product that provides him with the most economical operating plant. The following Ruberoid Insulating Products are tabulated to enable you to choose quickly the one for the correct purpose. Greater detail and description are provided in the Ruberoid Catalog on "Heat or Cold Insulating Products," which will be sent on request.

Product	Temp. Limit	Suggested Use
Hi-Temp	to 1600 F	Protective inner layer for low temperature insulations.
Pyrfelt	to 1000 F	Breechings and flues -- withstands vibration.
Sponge Felt	to 750 F	For vibrating pipes and underground insulation -- excellent efficiency.
85 per cent Magnesia	to 600 F	Combined efficiency and reasonable cost -- General use in industrial work
Imperial	to 500 F	For temporary lines that require efficiency and constant removal of insulation.
Watcocell	to 350 F	For a low-cost, medium pressure industrial steam line.
Air Cell	to 350 F	Standard insulation for residential pipes.
Woolfelt	to 180 F	For cold and hot water lines. Recommended especiallyf or air conditioning work.
Anti-Sweat	to 120 F	For cold water lines to prevent condensation.
Frost-proof	30 F to 100 F	To assist in the prevention of freezing in circulating water pipes exposed to cold.



Air Cell Pipe Covering—A low-cost insulation for residential use.



Woolfelt Pipe Covering—For the insulation of pipes carrying hot or cold water—also prevents condensation under normal operating conditions.

Sheet and Block Insulations

All of the above products are also made in sheet and block form to whatever thickness may be required. Standard sizes are usually 6, 12, 18 or 36 in. wide x 36 in. long. In this form they are used for insulating flat or irregular surfaces, such as tanks, breechings, furnaces, etc.

Insulating Cements

For the finishing of sheet and block insulation and the insulating of irregular surfaces, such as valves, unions, flanges, etc., the Ruberoid line of insulating cements is complete. This group of plastics not only uses as its base asbestos, but also takes advantage of such excellent natural products as magnesium, mineral wool and Vermiculite.

Asbestos Cements—Factory Prepared—Grades AA, A, HF.

Asbestos Cements—Mine Run—Grades 115, 214.

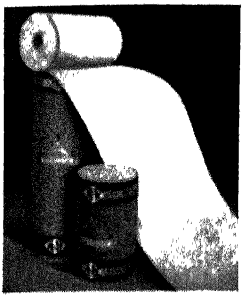
Magnesia Cement—85 per cent Magnesia.

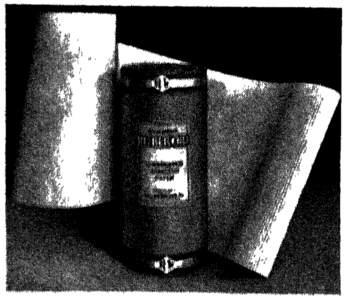
High Temperature Cement—Grade H.T.

Mineral Wool Cement—Grade R-W.

Vermiculite Cement—Grade A-11.

RU-BER-OID Asbestos Insulating Papers and Millboard





Asbestos Paper

Made of pure asbestos fibre and small percentage of binding material. Possesses unusual strength. This fine fire-resisting sheet may be obtained in 8, 10, 12, 14, 16 and 32 lb weights ranging in thickness from 0.019 to 0.0625 or 1/16 in. Rolls 18, 24 and 36 in. wide, weight 50 or 100 lb. Color, blue white.

Asbestos Corrugated Paper

Made entirely from highest quality asbestos felt paper by cementing flat sheet firmly with a 1/4 in. corrugated sheet which forms dead air spaces. Flexible. Efficient for insulating hot air pipes and ducts. 36 in. wide. Weight approximately 46 lb, 250 sq ft.

Asbestos Millboard

Asbestos Millboard is a rigid insulating board made of high quality asbestos fibres and non-organic binder. Has exceptional strength and whiteness. Cuts or drills easily. Standard or embossed finish. For temperatures to 1100 F. Sheets 42 x 48 in. Various thicknesses and weights.

RU-BER-OID ROCK WOOL

Mass or Fill Type **Bats Loose Granulated**

This indestructible wool is an efficient insulating material. Absolutely fireproof, vermin proof and inert toward moisture. Affords excellent sound-deadening and acoustical qualities.

RU-BER-OID Kraflined Rock Wool Bats are carefully fabricated. They are well tailored, uniform and easily handled. They are clean and sufficiently dense to prevent dusting and deterioration. Each bat is backed with a moisture-resistant paper that prevents the infiltration of vapor into the insulated space. RU-BER-OID Kraflined Bats provide "four flap" protection—each edge having an extension that allows adjoining bats to be covered preventing any exposed seam, thus effectively resisting the vapor flow.

Giant Bats
RU-BER-OID
Rock Wool

Recommended for all exposed spaces, such as sidewalls of new houses or under the roof, either in the roof rafters or the floor joist over the top floor ceiling. Bats without the Krafliner can be furnished if desired.

RU-BER-OID Wal-Pac Pads are insulating units 9 in. x 15 in. that can be fluffed up when applied to nearly fill the studding space. Furnished in cartons weighing 25 lb containing 20 pads that should cover 20 sq ft area.

RU-BER-OID Loose and Granulated Rock Wool is also available. Furnished in paper bags containing 35 lb each.

Packages Contain

Kraflined Standard Bat	15 in. x 23 in. x wall thickness	8 bats	19.16 sq ft	27 lb
Kraflined Demi-Bat	15 in. x 23 in x 2 in. thick	12 bats	28.75 sq ft	30 lb
Kraflined Giant Bat	15 in. x 48 in. x wall thickness	5 bats	25 sq ft	45 lb
Kraflined Giant Demi-Bat	15 in. x 48 in. x 2 in. thick	8 bats	40 sq ft	45 lb

Reynolds Metals Company, Incorporated

Federal Reserve Bank Building

Richmond, Virginia

NEW YORK CHICAGO SAN FRANCISCO MINNEAPOLIS LOUISVILLE

REYNOLDS METAL INSULATION

DESCRIPTION

Reynolds Metal INSULATION consists of genuine sheet aluminum foil cemented to both sides of tough kraft paper and is furnished in rolls.

CHARACTERISTICS

Reynolds Metal Insulation is waterproof, windproof, verminproof, and non-absorptive. It is light in weight and flexible, yet stiff enough to conform to angles and curves when not under tension or suspended. Its heat-storage capacity is negligible, owing to its light weight and limited mass. The aluminum used retains its reflectivity under all normal conditions, as the surface is protected by a transparent oxide which forms immediately on exposure. Tests conducted by Prof. Gordon B. Wilkes on Reynolds Metal Insulation exposed to laboratory fumes, moisture, dust and salt water atmosphere show that visual brightness is not essential to good performance. Aged and soiled material maintained reflectivities from 87.5 to 94.5 per cent.

Reynolds Metal Insulation will not absorb moisture; therefore its insulating value is not affected by water or water vapor. An absorbent material loses insulating value as its moisture content increases.

INSULATION VALUES—TYPICAL CONSTRUCTION

Description of Air Space Formations	Number of Air Spaces	Position of Insulation	Heat Flow	*U	†Conductance
0—Uninsulated single Air Space	2	Horizontal	Up	0.15	0.27
1—Air Space faced with Single Mounted Foil	2	Horizontal	Down	0.08	0.10
	3	Horizontal	Up	0.12	0.17
2—Air Space divided in two with one layer Double Mounted Foil	3	Horizontal	Down	0.06	0.07
	(1)	No Insulation		0.26	1.32
3—Air Space divided into three with one layer of Double Mounted Foil and one layer Single Mounted Foil	2	30° Slope	Up	0.16	0.25
	2	30° Slope	Down	0.11	0.15
In all cases the Air Space faced with Foil is ¾ in. or over	3	30° Slope	Up	0.12	0.17
	3	30° Slope	Down	0.08	0.10
	(1)	No Insulation		0.34	1.25
*U—Transmittance in Btu/sq. ft./hr./F. (Inside air to outside air) with a 15 mph. wind velocity outdoors	1	Vertical	Out	0.15	0.34
	2	Vertical	Out	0.12	0.21
†—Air Space conductance in Btu/sq. ft./hr./F. (plaster base to sheathing)	3	Vertical	Out	0.09	0.13
	(1)	No Insulation		0.23	1.06

The Standard Lime & Stone Company

First National Bank Building — **Baltimore, Maryland**

Manufacturers of Capitol Rock Wool Home Insulations

Franchised Distributors in all Principal Cities

The Standard Lime and Stone Company, prominent in the building materials industry since 1888, manufactures CAPITOL ROCK WOOL INSULATIONS, all types of lime products, fluxing and crushed stone, Capitol Portland Cement, etc.

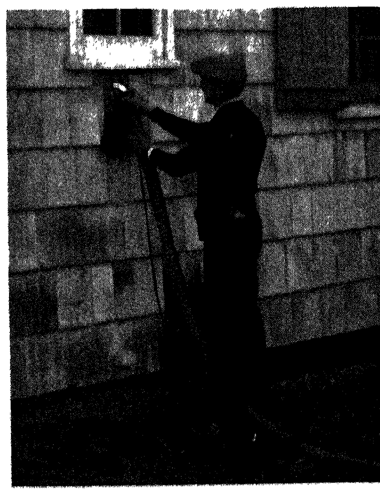
Their new process of manufacturing Capitol Rock Wool produces a refined, longer, more flexible fibre resulting in increased insulating efficiency.

Capitol Rock Wool in Existing Homes

Capitol Rock Wool Grade "A" Blowing Fibre is pneumatically introduced into the wall air spaces and between rafters or joists in roofs or attic floors of existing homes. This "blowing" method of installation is applicable to any type of construction shingle, clapboard, brick veneer, stucco or half timbered. The installation leaves no visible telltale marks. Franchised blowing contractors install Capitol Rock Wool in accordance with the master specifications of the company's home insulation engineers.

Capitol Rock Wool in New Structures

In the past, new homes or buildings were insulated at the time of erection by placing prefabricated batts between the studding and between roof rafters. However, many new structures are now effectively insulated by pneumatically installing Capitol Grade "A" Blowing Fibre after the scratch coat of plaster is applied.



Installing Capitol Rock Wool by Pneumatic Method

CAPITOL ROCK WOOL BATTS

Installing Capitol Rock Wool Batts in a New Structure

Moisture Proofed Special processing renders fibres moisture-resisting.

Vaporproofing Membrane Protects exposed surface of the batts against moisture from wet plaster and excessive interior humidity. Tested membrane is enclosed separately in each carton. It is 17½ in. wide and of sufficient length to give a smooth continuous membrane protected surface without open joints. Tacking is quick and easy.

Cuts Cost Capitol Batts are semi-rigid made in one size, 15 in. x 24 in. This semi-rigid feature permits "spring fit" between framing members spaced either 16 in. or 24 in. on centers. Easily cut to fit irregularly shaped spaces.

Two Thicknesses Wall thickness, for maximum efficiency, also 2 in. thickness.

Permanency All Capitol Rock Wool Products are permanent, non-deteriorating. They bring lasting, year-round comfort fire protection to the home. Fuel savings ultimately return the investment many times over.

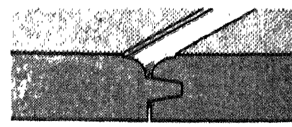
CAPITOL ROCK WOOL IS EFFICIENT

The dead-air cell structure of Capitol Rock Wool virtually eliminates the transmission of heat and sound. In addition CAPITOL ROCK WOOL WILL NOT BURN.

Send for Catalogs and Samples of Capitol Rock Wool Home Insulations.

"Look for the Capitol Dome on every package."

Weatherwood Tile Thicknesses of ½, ¾ and 1 in. in the following sizes: 12 x 12 in., 12 x 24 in., 16 x 16 in., 24 x 24 in., 16 x 32 in. and 24 x 48 in. The "Ogee" design, on all edges, produces a result comparable to that were a completely continuous single piece of board used and likewise enhances the decorative effect. Nailed "blind" to construction in new buildings or they may also be cemented over plaster to both redecorate and increase insulation values in existing buildings.



Ogee Edge

STRUCTURAL INSULATION

Weatherwood Insulating Lath Adds all of the purposes of a good lath to ½ in. of wood fiber board insulation. Each piece is 18 x 48 in. long. The long edges have a steel reinforcing member (optional), which the plaster completely surrounds, reducing possibilities of cracks at the joints between adjacent boards. The excellent bond between the fibrous board face and the plaster eliminates any necessity for plaster keys, saving plaster and labor quantities.

Weatherwood Sheathing Manufactured in pieces 2 ft x 8 in. each 25/32 in. thick, with the long edges tongued and grooved for horizontal application as sheathing material to the exterior walls of frame buildings. Both sides and all edges are asphalt coated, making the board highly water resistant.

Roof Insulation In sheets 22 x 47 in. ½, 1, 1½ and 2 in. thick. All but the ½ in. size are supplied laminated, and may be obtained with square or "ship-lapped" edges.

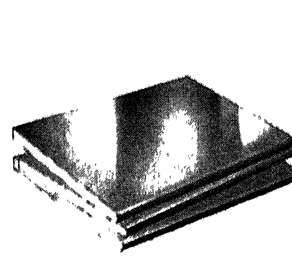
Weatherwood Insulating Lath

REFLECTIVE INSULATION

Insulating Rocklath This universally used plaster base is also available, with a highly polished aluminum foil back, augmenting its well known fireproofing and structural qualities with real insulation value at very slight additional cost.

Insulating Sheetrock Where unit or dry wall materials are used for the interior finish Insulating Sheetrock in ⅜ in. or ½ in. thickness combines its qualities with those of aluminum foil reflective insulation for use in exterior wall construction and top floor ceilings.

Vapor Barriers In existing buildings, where condensation has caused difficulties, Insulating Sheetrock, applied over the plaster will supply a high efficiency aluminum foil vapor barrier and fully restore the interior finish. Sheetrock will take any paint or wall paper decoration.



Reflective Insulation

Wood Conversion Company

First National Bank Building, **St. Paul, Minn.**

NEW YORK   CHICAGO   TACOMA   DALLAS

BALSAM-WOOL AND NU-WOOD INSULATIONS

BALSAM-WOOL
- **Sealed Insulation**
- **Acoustical Blanket**
- **Sound Deadening**
- **Industrial Insulation**
- **Refrigerator Insulation**

NU-WOOD
- **Kolor-Fast Tile**
- **Kolor-Fast Plank**
- **Kolor-Fast Board**
- **Kolor-Fast Wainscot**
- **Kolor-Fast Sheathing**

NU-WOOD
- **Lath**
- **Roof Insulation**
- **Industrial Insulation**
- **Refrigerator Insulation**

KOLOR-TRIM Pre-decorated Moldings

BALSAM-WOOL—The Original Moisture Barrier Insulation

The Moisture Barrier, which is universally recommended by engineers and architects, has been incorporated in Balsam-Wool for 19 years. This Barrier, improved as construction and equipment demanded, now consists of double layers of asphalted kraft a heavier liner being used on the warm side. Encased between this protective covering is an insulating mat of fleecy wood fibres, chemically treated to resist fire, rot, termites and vermin. 92 per cent of the mat volume is dead air.

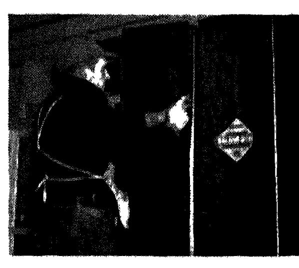
*Balsam-Wool Spacer Flange**

Balsam-Wool SEALED Insulation is fabricated at the factory to a controlled density of 2.2 lb per cubic foot. The mat has a coefficient of .25 Btu per hour, per square foot, per 1 degree F difference in temperature, per 1 in. thickness.

As applied, factory efficiency is assured. The Spacer Flange* on each edge folds over and is fastened to framing members with a staple hammer, assuring important air space, front and back.

Balsam-Wool is available in ½ and 1 in. thicknesses in widths of 12, 16, 20, 24 and 33 in.—Wall-thick in widths of 12, 16, 20 and 24 in.

*Pat. Applied For.



Application is quick and easy

NU-WOOD INTERIOR FINISH—STRUCTURAL INSULATION

Nu-Wood Kolor-Fast and Sta-Lite Interior Finish (Tile, Plank, Board and Wainscot) is applicable either to new construction or to existing buildings. It offers varied and pleasing decoration, also insulation and acoustical value.

Nu-Wood Insulating Lath has several times the bonding strength of wood lath—continuous surface eliminates dirty lath marks, reduces cracks. V-joint resists trowel pressure in both directions assures unbroken insulation value.

Nu-Wood Insulating Sheathing is surfaced on both sides with double coats of special moisture proofing compound. Large boards, marked for nailing—speed erection—stronger, windproof, insulated construction.

PUBLICATIONS

Important to the field of heating, ventilating and air conditioning are the technical journals, trade papers and business publications serving these industries. They include regular monthly editions, special annual numbers and trade catalogs issued by commercial publishers; and many periodical and annual editions published by engineering societies and trade associations.

These publications are a year-round source of information on the many problems involved in the design, production, distribution, operation and maintenance of heating, ventilating and air conditioning equipment, and related problems in refrigeration.

In editorial content and in their advertising pages are given a comprehensive review of developments in their respective branches of the industry. By means of scientific and technical articles they disseminate information of value—they provide valuable data for the engineer, practical helps for the production man, and also serve the distributor, dealer, contractor, and the operating and maintenance man.

PUBLICATIONS (p. 1086-1096)

Specialized trade papers serving a specific branch of the industry; general publications serving the broader field of the entire industry and profession; and technical publications providing the data necessary for scientific development of the industry.

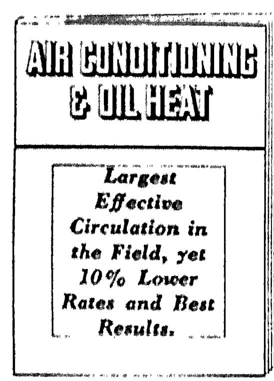
Many publications compile market statistics and provide merchandising suggestions for their readers. These services are of value not only to their readers, but are important to manufacturers who advertise their products in the pages of these publications.

Consistently read—and their contents correlated with private and governmental data on development and distribution of heating, ventilating and air conditioning equipment—these publications afford a comprehensive understanding of the problems and progress of the industry as a whole.

AIR CONDITIONING & OIL HEAT

232 Madison Ave. Lex. 2-4566 New York, N. Y.

Chicago	San Francisco	Baltimore	Los Angeles
A. F. Delgado 929 Forrest Ave. Evanston, Ill. University 7550	Don Harway & Co. 155 Montgomery St. Exbrook 6029	Candler Bldg. Plaza 7065	Don Harway & Co. 318 W. Ninth St. Tucker 9706



Established in 1928 as "OIL HEAT", this paper covered the manufacture, sale, installation and servicing of oil burners its first 7 years. In 1935, its title was changed and the editorial content expanded to cover air conditioning and heating also. This inspired and kept pace with the field itself.

Oil burner manufacturers and dealers, being progressive in both merchandising and technical problems, dominate air conditioning in many sections. Oil fired heating and air conditioning has grown steadily in public favor. 2,300,000 burners are now operating in the U.S.A. and our readers are servicing them.

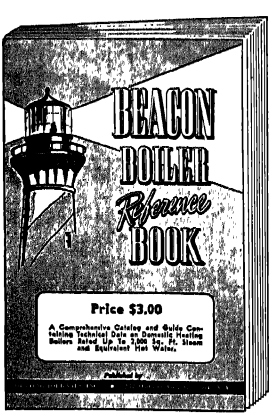
Of the 13,258 oil burner dealers, 7181 handle air conditioning. We reach them and also fuel oil dealers, heating contractors, accessory jobbers, manufacturers, etc. Member, CCA. Total Average Edition shown in Nov. 1940 Report: 16,558 Copies.

Subscription price: $3 a year. Issued monthly.

Many fine booklets and reprints are available at small charges on all phases of oil burners, heating and air conditioning.

Advertisers find this paper a profitable advertising medium. We also have a "Direct Mail Service" covering our readers at low cost for advertisers.

BEACON BOILER REFERENCE BOOK



Contains data on 7116 Heating Boilers and Boiler-Burner Units. Up to 2000 sq ft Steam and Equivalent Hot Water. Covers 195 makes of old, new and obsolete boilers. (405 trade names).

554 pages; 1½ in. thick; 8¼ in. x 5¼ in. Handy for pocket, desk or brief case. Eliminates bulky files.

$3.00 per copy; $3.25 if sent COD.

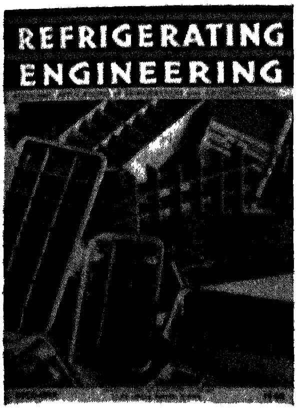
A necessary reference for everybody working with heating boilers and boiler-burner units. Data include ratings, firing rates, combustion chamber measurements, heating surface, floor area, base height, chimney and smoke pipe data, etc.

Also given in the listings are the trade names or numbers of the various series made by the manufactuers; in most cases of older boilers, the date of the catalog and whether it is obsolete; location or disposal of manufacturer; in many cases the source of parts. Shape of boiler, and whether steel, cast iron, etc., also given.

American Society of Refrigerating Engineers

37 West 39th Street, New York, N. Y.

REFRIGERATING ENGINEERING



The most rapidly growing magazine in the refrigeration field

REFRIGERATING ENGINEERING'S

CIRCULATION increased more than 20 per cent during 1940. Long acknowledged the most authoritative periodical in the field, it has added steadily to the practical value of its contents, and its number of readers has grown in proportion. A wide variety of material is presented, all from the viewpoint of its usefulness to the reader in his own business. Up-to-date and attractive in appearance and style, this magazine is a *must* for men who keep in touch with all that is new and important in refrigeration and air conditioning.

THE REFRIGERATING DATA BOOK

THE REFRIGERATING DATA BOOK is now an essential tool in the refrigeration and air conditioning industries. Editions have been published in 1932, 1934, 1936 and 1938. The 1940 Edition (Volume II) is *entirely different* from any preceding volume. It consists wholly of practical, how-it-is-done chapters on all the known applications of air conditioning and refrigeration.

The *Applications Edition* carries information of a scientific and popular nature to the scores of industries using refrigeration processes. The 60 chapters are divided into seven main sections devoted to freezing processes, refrigeration in processing, industrial air conditioning, industrial refrigeration, commercial refrigeration, comfort air conditioning, and storage refrigeration.

APPLICATION DATA BULLETINS

AN outstanding addition to REFRIGERATING ENGINEERING during 1939 and 1940 was the APPLICATION DATA Bulletins which appear regularly in each issue. These bulletins are also available separately at reasonable prices for single copies or quantity orders.

The APPLICATION DATA Bulletins tell precisely how refrigeration is used in various fields, giving examples and specific information on the best practice up to date. These subjects have been covered to date: refrigeration of locker plants, in fur storage, of liquids, of apples and pears, blower coils in refrigeration, refrigeration in restaurants, humidity in refrigeration, refrigeration service charts, refrigeration for skating rinks, butter and cheese making, milk plants, retail stores, citrus fruits, beer dispensing; load calculations, operation of ammonia machines, etc.

CODES AND STANDARDS

THE A.S.R.E. further contributes to refrigeration progress by its participation in establishing codes and standards in the industry. Among the recent codes made available are: No. 13 Rating and Testing Air Conditioning Equipment; No. 14 Rating and Testing Mechanical Condensing Units; No. 15 Mechanical Refrigeration Safety Code; No. 16 Rating and Testing Self-Contained Air Conditioning Units; No. 17 Rating and Testing Refrigerant Expansion Valves; No. 18 Testing Drinking Water Coolers. Other codes are now in preparation.

MEMBERSHIP ACTIVITIES

IT is the policy of the A.S.R.E. to treat in its meetings current subjects touching upon all phases of the art of refrigeration. Membership is in two grades with dues from $7.50 to $15.00. Sections hold meetings in the following cities: Boston, New York, Philadelphia, Detroit, Chicago, Milwaukee, St. Louis, Los Angeles, Baltimore-Washington, Richmond, Pittsburgh, Cincinnati, Cleveland and Kansas City.

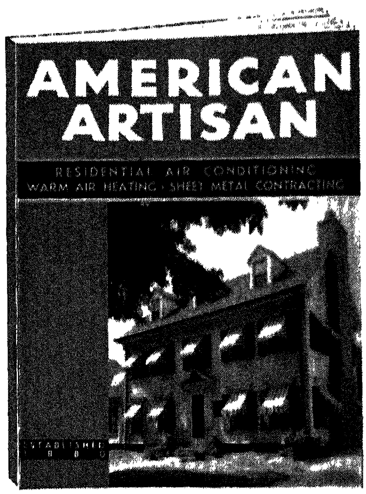
To keep apace with progress in refrigeration and air conditioning, read the publications and follow the activities of THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS, 37 West 39th St., New York, N. Y.

American Artisan

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, **Chicago, Ill.**



AMERICAN ARTISAN, now in its 62nd year of publication, covers the field of warm air heating, residential air conditioning, and sheet metal contracting. A special section of each issue has been devoted to air conditioning since 1932, when it first became apparent that air conditioning for homes was to be along the lines of the central, forced warm air heating system.

Its readers are warm air heating and sheet metal contractors, dealers, jobbers and manufacturers, and also architects, engineers, and public utility companies who take it for its thorough coverage of air conditioning for the home field.

To answer the industry's need for a dependable guide to equipment purchases, it publishes in each January issue a complete and up-to-the-minute directory of warm air heating, air conditioning and sheet metal products and equipment. This directory lists all products used in the field, their trade names, and the full names and addresses of all manufacturers. It is used by readers as a buying reference throughout the year.

Almost from the day interest in residential air conditioning began to develop, the advantages of the warm air type of heating system, with its duct distribution of air, were plain to see. It was adapted to all air conditioning factors, either through a self-contained central unit or through a central furnace to which could be added step-by-step or as a whole, fan, washer, humidifier, filters, controls, cooling, and automatic firing.

Today, as a result of this ready adaptability as well as economy, tens of thousands of homes have winter air conditioning—supplied through forced warm air heating with air cleaning and humidification. Cooling apparatus can be attached to these systems readily whenever complete, year-'round air conditioning is desired.

This trend in residential air conditioning has placed a premium on air handling knowledge, and has brought to the fore the one man experienced in "treating" air at a central place and getting it properly distributed—the warm air heating and sheet metal contractor. The warm air heating industry has, furthermore, undertaken and made notable progress toward the solution of the many new engineering problems involved. All this has helped to put warm air heating in the center of residential air conditioning.

In aiding to develop this trend and assist in the solution of new problems, AMERICAN ARTISAN has provided a service to its field which has made it the recognized authority on residential air conditioning practice.

To manufacturers whose products are used in residential air conditioning, AMERICAN ARTISAN offers full coverage of the leading buying factors. Such manufacturers are invited to write for complete information about this expanding market.

AMERICAN ARTISAN is published monthly. It is a member of the Audit Bureau of Circulations and Associated Business Papers.

Subscription rates—$2.00 per year, $3.00 for two years in U. S., Canada, Mexico, Central and South America. Foreign $4.00 per year.

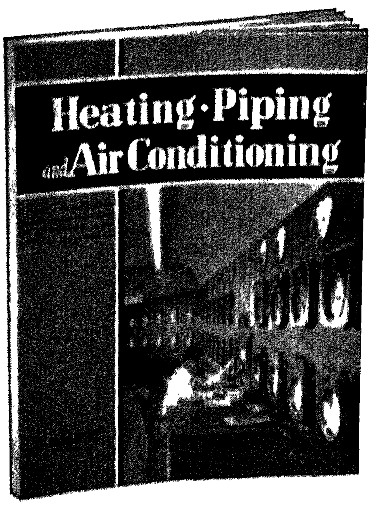
Advertising rates furnished upon request.

Heating, Piping and Air Conditioning

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, **Chicago, Ill.**



HEATING, PIPING AND AIR CONDITIONING is the publication which carries in each issue the official JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in addition to its own regular editorial section.

Its field is that of industry and large buildings. Editorially, it gives specialized attention to the design, installation, operation, and maintenance of heating, piping, and air conditioning systems in such plants and buildings.

In addition, there is published in each January issue a complete Directory of Commercial and Industrial Heating, Piping and Air Conditioning Equipment, which lists all products used in the field, their trade names, and the full names and addresses of all manufacturers. This directory has been established as the industry's buying and specifying guide, and is consulted by readers throughout the year, whenever equipment purchases are up for consideration.

H. P. & A. C. is read by consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping, and air conditioning in industrial plants, large commercial and public buildings, federal, state, and city governments, school boards and public utilities. Among its subscribers are numbered all members of the A.S.H.V.E., who represent about 30 per cent of its total circulation.

Such a coverage means, for the advertiser, consideration at all points in the selling of a heating, piping, or air conditioning product . . . consideration in the selection of a product during the preparation of plans and specifications; consideration in the actual purchase of a product for installation; consideration in the year 'round buying of a product for operating and maintenance requirements.

It has been evident for some time that the air conditioning field is made up of two distinct markets: (1) Industrial and Commercial; (2) Residential.

These two markets are different in equipment used; different in engineering problems involved, different in engineering, distributing, and consuming personnel . . . require, therefore, different selling jobs.

To sell the industrial and large building field for air conditioning, the manufacturer must win acceptance from the engineers who design, specify, install, operate, and select the system to meet the particular requirements of the plant or building. The system may be central, unit, or "split," but it is these engineers who are the influencing or purchasing factors.

It is to such groups that HEATING, PIPING AND AIR CONDITIONING editorially caters exclusively in the industrial and large building field. Without waste, the manufacturer of air conditioning products and accessory equipment, such as motors, drives, controls, etc., can reach through its pages those from whom he is seeking the necessary engineering acceptance.

Manufacturers interested in this field can obtain complete information by writing to the address given above.

HEATING, PIPING AND AIR CONDITIONING is a member of the Audit Bureau of Circulations and Associated Business Papers.

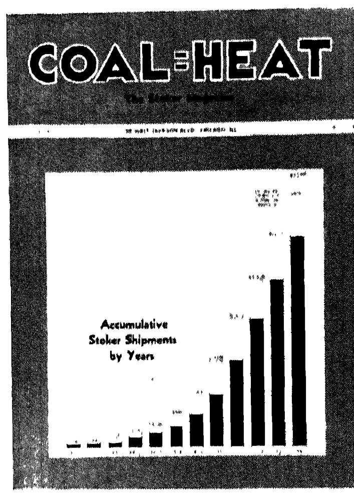
Subscription rates $2.00 per year; $3.00 for two years in U. S., Canada, Mexico, Central and South America. Foreign, $4.00 per year.

Advertising rates furnished upon request.

Coal-Heat

Published at

20 W. Jackson Blvd., **Chicago, Illinois**



COAL-HEAT is unrivalled as a reliable source of information on stoker merchandising and utilization, coal merchandising and utilization, new developments in modern coal-burning equipment, and general news of the stoker industry. Because it was the first magazine to recognize and promote the further sale and use of small stokers, COAL-HEAT has been named "The Stoker Magazine." It has continued to deserve this title by carrying more information, year in and year out, on small stokers than any other publication.

Today, COAL-HEAT's readership throughout the stoker industry and among the more sales-minded groups of the coal industry is unparalleled. In addition, COAL-HEAT now enjoys a friendship and close relationship with many leading fuel engineers, because it has shown its conviction in the value of these men in the coal, stoker and heating industries and has supported them in their work. For a number of years COAL-HEAT has carried many articles *for* and *by* fuel engineers.

COAL-HEAT's primary editorial job is to "further the more satisfactory use and greater sale of coal and modern coal-burning equipment." It is an active exponent of the increased use of scientific and engineering knowledge in the application and sale of stokers, coal and heating equipment. Believing that the sale of fuel and equipment is influenced to a large degree by the efficiency of their use, COAL-HEAT has directed its editorial program to both the merchandising and utilization sides of the stoker, coal and heating industries.

The greatly increased sale of stokers in recent years has given added emphasis to the COAL-HEAT market—has established the importance of COAL-HEAT's readers. As this market continues to grow, COAL-HEAT will become increasingly interesting to men who are concerned with the sale and use of stokers, modern coal-burning equipment and stoker coal.

At the beginning of each year COAL-HEAT issues a new and revised list of stoker manufacturers and assemblers doing business in the United States, Canada and certain important foreign countries.

Each listing includes the name of the company and the address, the executive in charge, the trade name of the stoker, the types and sizes of units available, and whether they are manufactured, assembled or handled as a "private brand" line.

COAL-HEAT also publishes books, booklets, manuals and reprints covering many subjects of interest to men in the stoker, coal and heating industries. These are available to readers at a small cost. In every respect, COAL-HEAT is a valuable index to what is going on in these important fields and a source of considerable information to its readers.

Subscription rates $1.00 a year; $2.00 for three years in U. S. and Canada.

Foreign rates—$2.00 a year; $4.00 for three years.

Advertising rates will be furnished upon request.

COAL-HEAT is published on or about the 15th of each month. There are four special issues each year—the MARKET DATA ISSUE in January; the SPRING STOKER NUMBER in April; the ANNUAL MERCHANDISING NUMBER in August; and the ANNUAL COMBUSTION NUMBER in November.

OILHEATING & AIRCONDITIONING fueloil JOURNAL

Published Monthly at

420 Madison Avenue

New York

MARKET: The oilheating market is a closely knit 4-way market—oilburners, heating, airconditioning, and fueloil. The modern and progressive oilheating dealer sells all four—a good oilburner, using good fueloil, firing a good heating or airconditioning system.

From 1919 to 1930, the only oilheating product sold by burner dealers was the conversion burner. In 1930 the sale of conversion burners represented 77.3 per cent of the dealers' gross income.

By the end of 1939, the average oilheating dealer got only 28.7 per cent of his income from conversion burners. But, beginning in 1932, he had added three other major oilheating lines—heating, fueloil and winter airconditioning. 1939 gross dollar volume of the average dealer was divided:

Conversion burner units	28.7 per cent
Heating equipment, including boiler-burner units	26.2 per cent
Fueloil	28.5 per cent
Winter airconditioning, including furnace-burner units	16.6 per cent

In 1939, 29 per cent or 59,685 conversion oilburners were sold with new cast iron or steel boilers. In addition, dealers sold 18,613 boiler-burner units. Total boiler sales by oilheating dealers increased 10 percent over 1938. These dealers did a winter airconditioning dollar volume in 1939 of $18,216,640.

SERVICES FOR ADVERTISERS

Key Market Studies.
Merchandising News.
Specific Products Reports.
Unit Sale Brand Preference Studies.
Booklets, reprints of special articles.

OILHEATING & AIRCONDITIONING: FUELOIL JOURNAL covers this integrated 4-way market.

It is the oldest paper in the field—established 1922. Editorially, it has consistently fostered every progressive development in the field and it has encouraged the trend to the complete oilheating dealer.

Every issue is carefully balanced editorially to cover the dealers' need for usable information on all four sides of his business.

Heating equipment manufacturers have long known FUELOIL JOURNAL as a powerful sales aid. Its reader interest is unique among trade papers.

CIRCULATION: Like its editorial content, the circulation of FUELOIL JOURNAL is carefully controlled to give complete coverage of this great 4-way market. A detailed breakdown from the latest circulation statement (December, 1940) shows:

Power oilheating and airconditioning dealers and distributors	12,234	
Key heating contractors, plumbing and heating contractors, and engineers	56	
Fueloil distributors, selling fueloil and range oil, and their branches	2,974	
Accessory and heating supply distributors	994	
Total dealers and distributors		16,258
Power oilheating and airconditioning manufacturers and their executives,	648	
Accessory manufacturers	546	
Total manufacturers		1,194
Total dealers and manufacturers,		17,452
Per cent of total circulation		99.19
Other miscellaneous		142
Grand total		17,594

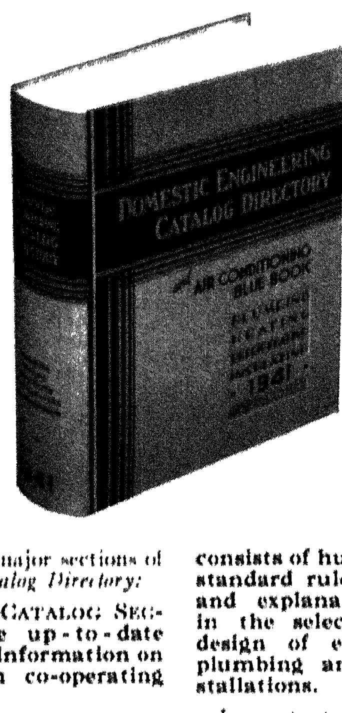
FUELOIL JOURNAL circulation covers the oil heating and air conditioning field at the minimum rate per thousand copies. It will pay you well to get full details. Write, wire or telephone.

Domestic Engineering Catalog Directory

Published by

DOMESTIC ENGINEERING PUBLICATIONS

1900 Prairie Avenue — Chicago, Illinois



NOW in its nineteenth year of service to its industry, the 1941 edition of *Domestic Engineering Catalog Directory* has been designed to meet present day plumbing, heating and air conditioning buying and specifying requirements. A complete, dependable, centralized source of product and technical information, arranged for quick and easy reference, *Domestic Engineering Catalog Directory* has placed buying and specifying on a greatly simplified basis.

The following are the major sections of *Domestic Engineering Catalog Directory:*

I. MANUFACTURERS' CATALOG SECTION Containing the up-to-date buying and specifying information on the products of each co-operating manufacturer.

II. CLASSIFIED DIRECTORY Listing virtually every known product used in heating, plumbing and air conditioning and names of manufacturers who make it.

III. TRADE NAME SECTION This section lists all known trade names of products used in the heating, plumbing and air conditioning field; also gives the names and addresses of the manufacturers.

IV. MANUFACTURERS' NAMES AND ADDRESSES This section gives the name, street address and city of every known manufacturer of heating, plumbing and air conditioning equipment.

V. TABLES AND RULES This section consists of hundreds of charts, tables, standard rules and layout diagrams and explanatory material required in the selection, coordination and design of equipment for heating, plumbing and air conditioning installations.

In constant use, throughout the year, by the leading buyers and specifying engineers in the industry, *Domestic Engineering Catalog Directory* presents to manufacturers a most productive method for the presentation of their product information.

For details of the many services available to manufacturers of heating, plumbing and air conditioning equipment through *Domestic Engineering Catalog Directory*, write Manufacturers' Catalog Service Department, 1900 Prairie Avenue, Chicago, Illinois.

HEATING AND VENTILATING

The Air Conditioning Monthly

THE INDUSTRIAL PRESS . . . *Publisher*
140-148 Lafayette St. New York, N. Y.

HEATING & VENTILATING reaches the "key men" of the industry—the engineers, contractors, and equipment manufacturers who have the final word in the specification, installation, production and maintenance of mechanical equipment for heating, air conditioning and ventilating.

The editorial content is designed to be of practical use to engineers engaged in the design, installation or operating of heating, ventilating or air conditioning equipment, and is prepared under the direction of field-experienced professional engineers. A maximum amount of space is given each month to articles showing how specific problems have been met, authoritative discussions of timely subjects, compilations of useful data, and descriptions of the latest practice, techniques and equipment. Each month an original Reference Data sheet is included for permanent use in a standard binder (back copies are available).

Special issues or special sections are published from time to time as needed. For example, in September 1940 a comprehensive Buyers Guide was included together with a special section on Air Conditioning for Air Defense. The success of the Buyers Guide has resulted in this being scheduled as an annual feature.

CIRCULATION

HEATING & VENTILATING'S total distribution (May 1940)—10,121, classified as follows:

Consulting Engineers (522) and Architects (183) Engineers Employed by Consulting Engineers and Architects (183)	888
Contractors (1646) and Engineers Employed by Contractors (272)	1,918
Governments and School Boards, and their Engineers	468
Public Utility Group	602
Industrial Firms, their Engineers, etc.	1,496
Buildings, Real Estate Management Companies, their Engineers	452
Manufacturers of Air Conditioning, Heating, Piping and Ventilating Equipment, Their Officials and Employees (755) and Designing Engineers (280)	10,35
Manufacturers' Agents and Sales-Engineering Firms (190), Sales Engineers and Salesmen (839)	1,029
Wholesalers (112) and Dealers (290)	402
Educational Institutions, Public Libraries, Associations	734
Miscellaneous and Unclassified	615
	9,639
Field Staff, Correspondents, Exchanges and Advertising Agencies	482
TOTAL	10,121

Subscriptions to HEATING & VENTILATING are $2.00 a year. Advertising rate cards, sample copies and market data will gladly be submitted on request.

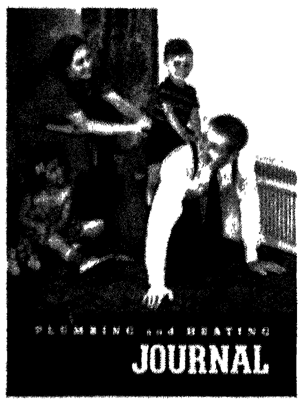
Plumbing and Heating Journal

Published by

THE ANGUS CO., INC.

515 Madison Ave., New York City

PLUMBING and Heating Journal is edited to furnish a well rounded, efficient service to the men engaged in the plumbing, heating, ventilating and air conditioning fields. To this end, it covers both the technical and business phases of their work, as well as many minor but exceedingly important ones.



It gives free technical service through a staff of practical engineers; expert merchandising assistance, and its technical and business articles are by men of recognized competence.

Hundreds of readers come to THE JOURNAL each year for solutions to their technical problems and while some of the questions and answers are published in the Readers' Technical Service section in each issue of the magazine, the vast majority of them having to do with practically every phase of heating, ventilating and air conditioning as well as plumbing are answered by mail, because most of the requests for help are urgent and a delay in answering would, in some cases, entail actual monetary loss to the contractor.

The Readers' Technical Service Department of THE JOURNAL is staffed by editors who have spent their lives in the business; men who were successful plumbing, heating, ventilating and air conditioning engineers before they wrote a line for publication, and who now devote their entire time to keeping abreast of the field's technical developments and using their knowledge and experience for the benefit of JOURNAL subscribers.

The technical service rendered its readers by THE JOURNAL is closely paralleled by what it strives to do for them in a business way, for it also publishes many authoritative articles on and answers questions concerning the various ramifications of business management and prints as part of every issue, special articles devoted to selling.

The JOURNAL'S air conditioning articles are of special value to plumbing-heating contractors. They stress the potentialities in this field, with the express intent of increasing air conditioning installations.

A new department, "Comfort Heating", devoted to equipment for automatic heating with coal, gas or oil, is also an exclusive JOURNAL feature.

Supplementing the business and technical articles is a large amount of exclusive, staff-gathered news that highlights the background of the trade's activities.

This news background is vital. It completes the industrial picture for the reader. It keeps him in intimate touch with what the various important associations and his fellow members of the craft are doing throughout the nation and it charts the trends that are likely to have a very definite influence on the future operation of his business.

THE JOURNAL editorial department draws its news from scores of trained correspondents located at strategic points throughout the country.

In this combination of the technical, business and news aspects of the industry that enables THE JOURNAL to achieve a finely balanced magazine that gives the reader the type of information he wants and needs, in brief, compact form.

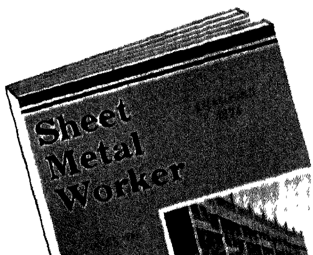
THE JOURNAL subscription price is: 1 year $2.00; special offer, 2 years $3.00; 4 years $5.00.

Sheet Metal Worker

Published by Edwin A. Scott Publishing Company

45 West 45th Street

New York



THE January 1941 issue of SHEET METAL WORKER will be its Sixty-Seventh Anniversary and Directory Number. It is the oldest publication in its field and is of vital importance to men interested in sheet metal work—air conditioning—warm-air heating and ventilation. Founded and published to 1909 by David Williams Company; 1909 to 1920 by United Publishers Corp.; since 1920 by the present publisher, the Edwin A. Scott Publishing Co.

SHEET METAL WORKER is today a monthly merchandising, business and technical journal basic to the use of sheet metal. It serves the various unified merchandising and installing branches of the industry, consuming sheet metal for the erection, maintenance and operating equipment of homes and buildings, including central air conditioning equipment, warm-air heating, ventilating, dust and refuse removal, and systems for handling material by air; kitchen and restaurant work; a wide variety of interior and exterior work for commercial, industrial, institutional, and residential buildings.

Subscribers are mainly merchandising contractors purchasing practically all products and equipment which they fabricate, erect or install. Manufacturers, jobbers and distributors also subscribe.

The market has three main divisions:

(1) Equipment for resale in connection with erection or installation work.
(2) Materials for fabrication.
(3) Shop equipment and supplies.

CIRCULATION

SHEET METAL WORKER is a member of the Audit Bureau of Circulations and the Associated Business Papers. It has a uniform distribution, with the greater part of its circulation centered in states showing the greatest industrial activity. Readers of SHEET METAL WORKER are made up of warm-air heating, air conditioning and sheet metal contractors and dealers. Also wholesalers, manufacturers, branch offices and salesmen. For further details send for ABC statement.

EDITORIAL

SHEET METAL WORKER has been outstanding in the editorial service it has rendered the trade and is noted for the practical usefulness of its articles and the timeliness of its editorials. Its editor is a noted author in this field and the author of several well-known books.

SHEET METAL WORKER also publishes books on heating, ventilating, sheet metal work, air conditioning, etc.

The Annual Issue published in January, contains a comprehensive and valuable Directory Section.

ADVERTISING

SHEET METAL WORKER has an enviable record of long term advertising and is proud of its long list of regular advertisers.

Because of its intimate contact with this field, SHEET METAL WORKER is well qualified to cooperate with manufacturers in their sales and advertising programs.

Subscription rates—$2.00 per year, U.S., and Mexico. Canada $2.50; Foreign $3.00. Advertising rates on request.

INDEX TO MODERN EQUIPMENT

HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1941

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Armstrong Machine Works, 1032-1033
C. A. Dunham Co., 970-971
Hoffman Specialty Co., Inc., 976-977
Illinois Engineering Co., 978-979
Kieley & Mueller, Inc., 1035
Mueller Steam Specialty Co., Inc., 1036
Sarco Company, Inc., 980-981
Trane Company, The, 858-859
Warren Webster & Co., 984-987
Wright-Austin Co., 1037

AIR FILTERS (*See Filters, Air, also Air Cleaning Equipment*)

AIR FILTER GAUGE

Simplex Controls Div., The Herbusch Corp., 960

AIR FITTINGS, Brass

Martocello, Jos. A. & Sons, 882

AIR FILTER CLEANING COMPOUNDS

Air-Maze Corporation, 862-863
Oakite Products, Inc., 877

AIR MEASURING AND RECORDING INSTRUMENTS

American Blower Corp., 818-819
American Moistening Co., 878
Babcock & Wilcox Co., 996
Bristol Company, The, 941
Julien P. Friez & Sons, Div. of Bendix Aviation Corp., 946
Grinnell Co., Inc., 972-974, 1034
Illinois Testing Laboratories, Inc., 950
Johnson Service Co., 948-949
Minneapolis-Honeywell Regulator Company 954-955
Palmer Company, 956
Parks-Cramer Company, 822
Powers Regulator Co., 958-959
Taylor Instrument Companies, 962-963

AIR MOISTENING APPARATUS (*See Humidifiers*)

AIR PURIFYING APPARATUS

Air-Maze Corp., 862-863
American Air Filter Company, Inc., 864-865
American Blower Corp., 818-819
Buffalo Forge Co., 903
Burnham Boiler Corp., 994-995
Carrier Corporation, 820
Coppus Engineering Corp., 867
Davies Air Filter Corp., 868
Delco Appliance Div., General Motors Sales Corp., 832-833
DeBothezat Division, American Machine & Metals, Inc., 905
Ilg Electric Ventilating Co., 906
Owens-Corning Fiberglas Corp., 870-871
H. J. Somers, Inc., 873
Staynew Filter Corp., 874-875
Westinghouse Elec. & Mfg. Co., 826-827, 876
L. J. Wing Mfg. Co., 855-857

AIR RECEIVERS (*See Receivers, Air*)

AIR TUBING, Flexible Metal (*See Tubing, Flexible Metallic*)

AIR VELOCITY METERS (*See Meters, Air Velocity*)

AIR VELOCITY REGULATORS

Anemostat Corp. of America, 919
Barber-Colman Co., 921,940
Johnson Service Co., 948-949
Minneapolis-Honeywell Regulator Co., 954-955
Powers Regulator Co., 958-959

AIR WASHERS

Air-Maze Corp., 862-863
American Blower Corp., 918-919
American Coolair Corp., 900-901
Autovent Fan & Blower Co., 899
Baker Ice Machine Co., 888
Bayley Blower Company, 902
Binks Manufacturing Co., 880
Buffalo Forge Company, 903
Clarage Fan Company, 821
Delco Appliance Div., General Motors Sales Corp., 832-833
Meyer Furnace Company, 840
L. J. Mueller Furnace Co., 838-839
D. J. Murray Co., Unit Heater and Cooler Div., 852
New York Blower Co., 910
Parks-Cramer Company, 822
H. J. Somers, Inc., 873
B. F. Sturtevant Co., 908-909
Trane Company, The, 858-859
United States Air Conditioning Corp., 828
Utility Fan Corporation, 914
Vilter Manufacturing Co., 898
York Ice Machinery Corp., 829

ALARMS, Water Level

Illinois Engineering Co., 978-979
Kieley & Mueller, Inc., 1035
McDonnell & Miller, 990-991
Mercoid Corporation, 953
Minneapolis-Honeywell Regulator Co., 954-955
Mueller Steam Specialty Co., 1036
Warren Webster & Co., 984-987
Wright-Austin Company, 1037
Yarnall-Waring Company, 1038

ALGAE PREVENTION (*See also Slime Prevention*)

Oakite Products, Inc., 877

ALUMINUM FOIL, Insulation

Alfol Insulation Co., 1058
Reynolds Metals Co., Inc., 1080
Ruberoid Co., The, 1078-1079
United States Gypsum Co., 1082-1083

ALUMINUM FOIL VAPOR BARRIER (*See Aluminum Foil*)

AMMONIA COILS (*See Coils, Ammonia*)

ANEMOMETERS

Julien P. Friez & Sons, Div. of Bendix Aviation Corp., 946
Illinois Testing Laboratories, Inc., 950
Taylor Instrument Companies, 962-963

ASBESTOS PRODUCTS (*See also Insulation*)

Carey, Philip, Co., 1061
Ehret Magnesia Manufacturing Co., 1064-1065
Johns-Manville, 1072-1073
H. W. Porter & Co., 1056
Ric-wiL Company, The, 1057
Ruberoid Co., The, 1078-1079
United States Gypsum Co., 1082-1083

ATTIC FAN COOLERS (*See Fans, Attic; Ventilators, Attic*)

AUTOMATIC FUEL BURNING EQUIPMENT (*See Burners, Automatic; Coal Burners, Automatic; Furnace Burners; Gas Burners; Oil Burners; Stokers*)

AUTOMATIC SHUTTERS (*See Shutters, Automatic*)

AUTOMOBILE HEATER FANS

Torrington Mfg. Co., The, 911-913

BACTERIA CONTROL

Oakite Products, Inc., 877

BEARING, Bronze

Arthur Harris Co., 1044
Hastings Air Conditioning Co., 845

BENDS, Pipe, Ferrous and Non-Ferrous

Baker Ice Machine Co., 888
Crane Co., 998-999
Frick Company, 891
General Refrigeration Corp., 892
Grinnell Co., Inc., 972-974, 1034
Arthur Harris & Co., 1044
Parks-Cramer Co., 822
Vilter Manufacturing Co., 898
York Ice Machinery Corp., 829

BENDS, Return (*See Pipe, Return Bends*)

BLOCKS, Asbestos

Carey, Philip, Co., 1061
Ehret Magnesia Manufacturing Co., 1064-1065
Johns-Manville, 1072-1073
Ruberoid Co., The, 1078-1079

BLOWERS, Fan (*See Fans, Supply and Exhaust*)

BLOWERS, Forced Draft

American Blower Corp., 818-819
American Coolair Corp., 900-901
Autovent Fan & Blower Co., 899
Bayley Blower Company, 902
Buffalo Forge Company, 903
Champion Blower & Forge Co., 904
Clarage Fan Company, 821
Coppus Engineering Corp., 867
Curtis Refrigerating Machine Co., Div. of Curtis Manufacturing Co., 890
DeBothezat Division American Machine & Metals, Inc., 905
New York Blower Co., 910
Servel, Inc., 894
B. F. Sturtevant Co., 908-909
Westinghouse Elec. & Mfg. Co., 826-827, 876
L. J. Wing Mfg. Co., 855-857

BLOWERS, Heating and Ventilating

American Blower Corp., 818-819
American Coolair Corp., 900-901
Autovent Fan & Blower Co., 899
Bayley Blower Company, 902
Buffalo Forge Company, 903
Champion Blower & Forge Co., 904
Clarage Fan Company, 821
DeBothezat Division American Machine & Metals, Inc., 905
C. A. Dunham Co., 970-971
Fedders Manufacturing Co., 842-843
Hastings Air Conditioning Co., 845

Gar Wood Industries, 834-835
International Boiler Works Co., 1006
Kewanee Boiler Corp., 1002-1005
Pacific Steel Boiler Div., U. S. Radiator Corp., 1007
Smith Twin Tubular Boiler Co., 1012
Spencer Heater Division, 1008-1009

BOILERS, Water Tube

Babcock & Wilcox Co., 996
Burnham Boiler Corp., 994-995
Combustion Engineering Co., 1019
Fitzgibbons Boiler Co., 1000-1001
Frick Company, 891
International Boiler Works Co., 1006
Smith Twin Tubular Boiler Co, 1012
Spencer Heater Division, 1008-1009

BREECHINGS AND CHIMNEYS

Bethlehem Steel Co., 934
Farrar & Trefts, 997
International Boiler Works Co., 1006

BURNERS, Automatic (*See also Coal Burners, Stokers*)

Airtemp Div., Chrysler Corp., 830-831
American Radiator & Standard Sanitary Corp., 992-993
Barber Gas Burner Co., 1014
Crane Company, 998-999
Combustion Engineering Co., 1019
Delco Appliance Div., General Motors Sales Corp., 832-833
Detroit Stoker Company, 1020-1021
Gar Wood Industries, Inc., 834-835
General Electric Company, 916-917
Iron Fireman Mfg. Co., 1022-1023
MotorStoker Div., Hershey Machine & Fdy. Co., 1024
Herman Nelson Corp., 853
Spencer Heater Division, 1008-1009
Webster Engineering Co., 1015
Westinghouse Elec. & Mfg. Co., 826-827, 876
Williams Oil-O-Matic Heating Corp., 836

BURNERS, Coal (*See Coal Burners*)

BURNERS, Combination Gas and Oil

Coppus Engineering Co., 867
Todd Combustion Equipment, Inc., 1017
Webster Engineering Co., 1015

BURNERS, Gas (*See Gas Burners*)

BURNERS, Oil (*See Oil Burners*)

CALKING, Building

Chamberlin Metal Weather Strip Co., 1048-1049
Johns-Manville, 1072-1073

CASTINGS, Bronze and Dairy Nickel Silver Metal

Arthur Harris & Co., 1044

CEMENT, Asbestos

Carey Philip, Co., 1061
Eagle-Picher Lead Co., 1063
Ehret Magnesia Manufacturing Co., 1064-1065
Johns-Manville, 1072-1073
Ruberoid Co., The, 1078-1079

CEMENT, Refractory (*See Refractories*)

CEMENT, Rock Wool

Carey, Philip, Co., 1061
Eagle-Picher Lead Co., 1063
EhretMagnesia Manufacturing Co., 1064-1065
General Insulating & Mfg. Co., 1066
Insulation Industries, Inc., 1067
Johns-Manville, 1072-1073
Ruberoid Co., The, 1078-1079
Standard Lime & Stone Co., 1081

CHAIN, Furnace Pulleys (*See also Pulleys, Chain*)

Auer Register Co., The, 920
Barber-Colman Co., 921,940
Hart & Cooley Manufacturing Co., 922-923
Hendrick Mfg. Co., 924
Independent Register Co., 925
L. J. Mueller Furnace Co., 838-839
Trane Company, The, 858-859
Tuttle & Bailey, Inc., 927-929
United States Register Co., 930
Waterloo Register Co., 931

CIRCULATORS, Hot Water Heating

Bell and Gossett Co., 969
Crane Company, 998-999
General Electric Company, 916-917
Triplex Heating Specialty Co., 982-983
Westinghouse Elec. & Mfg. Co., 826-827, 876

CLEANERS, Air (*See Air Cleaning Equipment*)

COAL BURNERS, Automatic, Anthracite

Buffalo Forge Company, 903
Crane Company, 998-999
Combustion Engineering Co., 1019
Iron Fireman Mfg. Co., 1022-1023
MotorStoker Div., Hershey Machine & Fdy. Co., 1024
Spencer Heater Division, 1008-1009

COAL BURNERS, Automatic, Bituminous

Brownell Company, 1018
Crane Company, 998-999
Combustion Engineering Co., 1019
Delco Appliance Div., General Motors Sales Corp., 832-833
Detroit Stoker Company, 1020-1021
Iron Fireman Mfg. Co., 1022-1023
Meyer Furnace Company, 840
Spencer Heater Division, 1008-1009

COILS, Aluminum

Aerofin Corporation, 883-885
Baker Ice Machine Co., 888
Delco Appliance Div., General Motors Sales Corp., 832-833
Arthur Harris & Co., 1044
Kramer Trenton Co., 847
McQuay Incorporated, 848-849
J. J. Nesbitt, Inc., 854
Refrigeration Economics Co., 823
B. F. Sturtevant Co., 908-909
Trane Company, The, 858-859
Young Radiator Company, 860

COILS, Ammonia

Aerofin Corporation, 883-885
Baker Ice Machine Co., 888
Carrier Corporation, 820
Crane Company, 998-999
Frick Company, Inc., 891
G & O Manufacturing Co., 886
General Refrigeration Corp., 892
Kramer Trenton Co., 847
McQuay Incorporated, 848-849
D. J. Murray Mfg. Co., Unit Heater and Cooler Div., 852
Refrigeration Economics Co., 823
Trane Company, The, 858-859
Vilter Manufacturing Co., 898
Worthington Pump & Machinery Corp., 896-897
Yarnall-Waring Co., 1038
York Ice Machinery Corp., 829
Young Radiator Company, 860

COILS, Blast

Aerofin Corporation, 883-885
American Blower Corp., 818-819
American Radiator & Standard Sanitary Corp., 992-993
Autovent Fan & Blower Co., 899
Bayley Blower Company, 902
Buffalo Forge Company, 903
Carrier Corporation, 820
Clarage Fan Company, 821
C. A. Dunham Co., 970-971
Fedders Manufacturing Co., 842-843
G & O Manufacturing Co., 886
McQuay, Incorporated, 848-849
Modine Mfg. Co., 850-851
D. J. Murray Mfg. Co., Unit Heater & Cooler Div., 852
John J. Nesbitt, Inc., 854
Refrigeration Economics Co., 823
B. F. Sturtevant Co., 908-909
Trane Company, The, 858-859
Westinghouse Elec. & Mfg. Co., 826-827, 876
Young Radiator Co., 860

COILS, Brass

Aerofin Corporation, 883-885
E. B. Badger & Sons Co., 1030
Carrier Corporation, 820
Crane Company, 998-999
Grinnell Co., Inc., 972-974, 1034
Arthur Harris & Co., 1044
B. F. Sturtevant Co., 908-909

COILS, Cooling

Aerofin Corporation, 883-885
Baker Ice Machine Co., 888
Carrier Corporation, 820
Crane Company, 998-999
Curtis Refrigerating Machine Co., Division of Curtis Manufacturing Co., 890
Fedders Manufacturing Co., 842-843
Frick Company, 891
G & O Manufacturing Co., 886
General Refrigeration Corp., 892
McQuay, Incorporated, 848-849
Refrigeration Economics Co., 823
Trane Company, The, 858-859
Vilter Manufacturing Co., 898
Worthington Pump & Machinery Corp., 896-897
York Ice Machinery Corp., 829
Young Radiator Company, 860

COILS, Pipe, Copper

Aerofin Corporation, 883-885
American Radiator & Standard Sanitary Corp., 992-993
E. B. Badger & Son Co., 1030
Baker Ice Machine Co., Inc., 888
Bayley Blower Company, 902
Bell and Gossett Co., 969
Crane Company, 998-999
Curtis Refrigerating Machine Co., Division of Curtis Manufacturing Co., 890

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Bell & Gossett Co., 969
Crane Company, 998-999
Fedders Manufacturing Co., 842-843
Fitzgibbons Boiler Co., Inc., 1000-1001
Kewanee Boiler Corp., 1002-1005
L. J. Wing Mfg. Co., 855-857

HEATERS, Refuse Burning

Kewanee Boiler Corp., 1002-1005
L. J. Mueller Furnace Co., 838-839

HEATERS, Storage

American District Steam Co., 1029
American Radiator & Standard Sanitary Corp., 992-993
Bell & Gossett Co., 969
Brownell Company, 1018
Burnham Boiler Corp., 994-995
Crane Company, 998-999
General Electric Company, 916-917

HEATERS, Tank

American District Steam Co., 1029
American Radiator & Standard Sanitary Corp., 992-993
Bell & Gossett Co., 969
Burnham Boiler Corp., 994-995
Crane Company, 998-999
Fitzgibbons Boiler Co., Inc., 1000-1001
Grinnell Co., Inc., 972-974, 1034
International Boiler Works Co., 1006
Kewanee Boiler Corp., 1002-1005
L. J. Mueller Furnace Co., 838-839
Pacific Steel Boiler Div., U. S. Radiator Corp., 1007
Spencer Heater Division, 1008-1009
United States Radiator Corp., 1010-1011
Weil-McLain Company, 1013
Westinghouse Elec. & Mfg. Co., 826-827, 876

HEATERS, Unit

Airtherm Manufacturing Co., 841
American Blower Corp., 818-819
Autovent Fan & Blower Co., 899
Bayley Blower Company, 902
Buffalo Forge Company, 903
Burnham Boiler Corp., 994-995
Carrier Corporation, 820
Clarage Fan Company, 821
Crane Company, 998-999
Delco Appliance Div., General Motors Sales Corp., 832-833
C. A. Dunham Co., 970-971
Electric Air Heater Co., Div. of American Foundry Equipment Co., 844
Fedders Manufacturing Co., 842-843
Grinnell Co., Inc., 972-974, 1034
Hastings Air Conditioning Co., 845
Ilg Electric Ventilating Co., 906
Iron Fireman Mfg. Co., 1022-1023
F. Jaden Manufacturing Co., 846
Kramer Trenton Co., 847
McQuay, Incorporated, 848-849
Modine Mfg. Co., 850
D. J. Murray Mfg. Co., Unit Heater and Cooler Div., 852
Herman Nelson Corp., 853
John J. Nesbitt, Inc., 854
New York Blower Co., 910
Refrigeration Economics Co., 823
B. F. Sturtevant Co., 908-909
Surface Combustion Corp., 824-825
Trane Company, The, 858-859
United States Air Conditioning Corp., 828
United States Radiator Corp., 1010-1011
Warren Webster & Co., 984-987
L. J. Wing Mfg. Co., 855-857
Young Radiator Company, 860

HEATERS, Unit, Gas Fired

Airtherm Manufacturing Co., 841
American Radiator & Standard Sanitary Corp., 992-993
Buffalo Forge Company, 903
Crane Company, 998-999
C. A. Dunham Co., 970-971
McQuay, Incorporated, 848-849
L. J. Mueller Furnace Co., 838-839
Surface Combustion Corp., 824-825
Trane Company, The, 858-859

HEATING SYSTEMS, Air

Acme Heating & Ventilating Co., 837
Airtemp Div., Chrysler Corp., 830-831
Airtherm Manufacturing Co., 841
American Blower Corp., 818-819
American Radiator & Standard Sanitary Corp., 992-993
Autovent Fan & Blower Co., 899
Buffalo Forge Company, 903
Burnham Boiler Corp., 994-995
Carrier Corporation, 820
Clarage Fan Company, 821
Crane Company, 998-999
Delco Appliance Div., General Motors Sales Corp., 832-833
C. A. Dunham Co., 970-971
Fedders Manufacturing Co., 842-843
Gar Wood Industries, Inc., 834-835
General Electric Company, 916-917
Ilg Electric Ventilating Co., 906
McQuay, Incorporated, 848-849
Meyer Furnace Co., The, 840
Modine Mfg. Co., 850-851
L. J. Mueller Furnace Co., 838-839
Herman Nelson Corp., 853
John J. Nesbitt, Inc., 854
Spencer Heater Division, 1008-1009
B. F. Sturtevant Co., 908-909
Surface Combustion Corp., 824-825
Trane Company, The, 858-859
United States Radiator Corp., 1010-1011
Westinghouse Electric & Mfg. Co., 826-827, 876
L. J. Wing Mfg. Co., 855-857
Young Radiator Company, 860
York Ice Machinery Corp., 829

HEATING SYSTEMS, Automatic

Airtemp Div., Chrysler Corp., 830-831
American Radiator & Standard Sanitary Corp., 992-993
Anderson Products, Inc., 1039
Barber Gas Burner Co., 1014
Burnham Boiler Corp., 994-995
Carrier Corporation, 820
Crane Company, 998-999
Delco Appliance Div., General Motors Sales Corp., 832-833
C. A. Dunham Co., 970-971
Gar Wood Industries, Inc., 834-835
General Electric Company, 916-917
Hoffman Specialty Co., Inc., 976-977
Illinois Engineering Co., 978-979
Iron Fireman Mfg. Co., 1022-1023
Meyer Furnace Company, 840
MotorStoker Division Hershey Machine & Fdy. Co., 1024
L. J. Mueller Furnace Co., 838-839
Herman Nelson Corp., 853
Sarco Co., Inc., 980-981
Spence Engineering Co., 961
Spencer Heater Division, 1008-1009
Surface Combustion Corp., 824-825
Trane Company, The, 858-859
United States Radiator Corp., 1010-1011
Westinghouse Elec. & Mfg. Co., 826-827, 876
Williams Oil-O-Matic Heating Corp., 836
Warren Webster & Co., 984-987
L. J. Wing Mfg. Co., 855-857
York Ice Machinery Corp., 829

HEATING SYSTEMS, Furnace

Acme Heating & Ventilating Co. 837
Airtemp Div., Chrysler Corp., 830-831
Airtherm Manufacturing Co., 841
American Radiator & Standard Sanitary Corp., 992-993
Carrier Corporation, 820
Crane Compzny, 998-999
Delco Appliance Div., General Motors Sales Corp., 832-833
Electric Air Heater Co., Div. of American Foundry Equipment Co., 844
Gar Wood Industries, Inc., 834-835
General Electric Company, 916-917
Meyer Furnace Company, 840
L. J. Mueller Furnace Co., 838-839
Herman Nelson Corp., 853
Spencer Heater Division, 1008-1009
Surface Combustion Corp., 824-825
United States Radiator Corp., 1010-1011
Westinghouse Elec. & Mfg. Co., 826-827, 876
Williams Oil-O-Matic Heating Corp., 836
L. J. Wing Mfg. Co., 855-857

HEATING SYSTEMS, Gas Fired

Acme Heating & Ventilating Co., 837
Airtemp Div., Chrysler Corp., 830-831
Airtherm Manufacturing Co., 841
American Blower Corp., 818-819
American Radiator & Standard Sanitary Corp., 992-993
Barnes & Jones, Inc., 968
Burnham Boiler Corp., 994-995
Carrier Corporation, 820
Crane Company, 998-999
Delco Appliance Div., Genera Motors Sales Corp., 832-833
C. A. Dunham Co., 970-971
Gar Wood Industries, Inc., 834-835
General Electric Company, 916-917
Kewanee Boiler Corp., 1002-1005
Meyer Furnace Co., The, 840
L. J. Mueller Furnace Co., 838-839
Herman Nelson Corp., 853
Spencer Heater Division, 1008-1009
Surface Combustion Corp., 824-825
Trane Company, The, 858-859
United States Radiator Corp., 1010-1011
Westinghouse Elec. & Mfg. Co., 826-827, 876

HEATING SYSTEMS, Hot Water

Airtemp Div., Chrysler Corp., 830-831
American Blower Corp., 818-819

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

SHEETS, Stainless Steel
American Rolling Mill Co., 933
Carnegie-Illinois Steel Corp., 935
Columbia Steel Co., 935

SHEETS, Steel
American Rolling Mill Co., 933
Bethlehem Steel Co., 934
Carnegie-Illinois Steel Corp., 935
Columbia Steel Co., 935
Tennessee Coal, Iron & Railroad Co., 935
United States Export Co., 935

SHUTTERS, Automatic
American Coolair Corp., 900-901
Autovent Fan & Blower Co., 899
Barber-Colman Co., 921, 940
Champion Blower & Forge Co., 904
Ilg Electric Ventilating Co., 906
Minneapolis-Honeywell Regulator Co., 954-955
New York Blower Co., 910
B. F. Sturtevant Co., 908-909
L. J. Wing Mfg. Co., 855-857

SLIME PREVENTION (*See also Algae Prevention*)
Oakite Products, Inc., 877

SMOKE DENSITY RECORDING
Bristol Company, The, 941
Leeds & Northrup Co., 951
Westinghouse Elec. & Mfg. Co., 826-827, 876

SOOT DESTROYER
Vinco Company, Inc., 988-989

SOUND DEADENING, Insulation
Aluminum Aircell Insulation Corp., 1059
Armstrong Cork Company, 1060
Celotex Corporation, The, 1062
Eagle-Picher Lead Co., 1063
Ehret Magnesia Mfg. Co., 1064-1065
Insulite Company, The, 1068-1069
International Fibre Board, Ltd., 1071
Johns-Manville, 1072-1073
Kimberly-Clark Corp., 1074-1075
Mundet Cork Corp., 1076
Pacific Lumber Co., The, 1077
Ruberoid Co., The, 1078-1079
Standard Lime & Stone Co., 1081
United States Gypsum Co., 1082-1083
Wood Conversion Co., 1084

SPRAY EQUIPMENT
April Showers Company, 879
Binks Manufacturing Co., 880
Martocello, Jos. A. & Co., 882
Yarnell-Waring Co., 1038

SPRAY NOZZLES
American Moistening Co., 878
American Blower Corp., 818-819
Baker Ice Machine Co., 888
Bayley Blower Company, 902
Binks Manufacturing Co., 880
Buffalo Forge Company, 903
Clarage Fan Company, 821
Marley Co., The, 881
Martocello, Jos. A. & Co., 882
D. J. Murray Mfg. Co., Unit Heater & Cooling Div., 852
Parks-Cramer Co., 822
B. F. Sturtevant Co., 908-909
Trane Company, The, 858-859
Westinghouse Elec. & Mfg. Co., 826-827, 876
Yarnell-Waring Co., 1038
York Ice Machinery Corp., 829

SPRAY NOZZLE COOLING SYSTEM
American Blower Corp., 818-819
April Showers Company, 879
Baker Ice Machine Co., 888
Bayley Blower Co., 902
Binks Manufacturing Co., 880
Buffalo Forge Co., 903
Clarage Fan Company, 821
Marley Company, 881
B. F. Sturtevant Co., 908-909
Trane Company, The, 858-859
Yarnall-Waring Co., 1038
York Ice Machinery Corp., 829

STACKS, Steel
Bethlehem Steel Co., 934
Brownell Co., 1018
Farrar & Trefts, Inc., 997
Hendrick Mfg. Co., 924

STAINLESS STEEL COILS (*See Coils, Stainless Steel*)

STEAM HEATING SYSTEMS (*See Heating Systems, Steam*)

STOKERS, Mechanical, Anthracite
Babcock & Wilcox Co., 996
Combustion Engineering Co., 1019
Crane Company, 998-999
Iron Fireman Mfg. Co., 1022-1023
MotorStoker Division Hershey Machine & Fdy. Co., 1024

STOKERS, Mechanical, Bituminous
Babcock & Wilcox Co., 996
Brownell Company, 1018
Combustion Engineering Co., 1019
Crane Company, 998-999
Delco Appliance Div., General Motors Sales Corp., 832-833
Detroit Stoker Co., 942-943
Iron Fireman Mfg. Co., 1022-1023
Meyer Furnace Company, 840

STOKER MOTORS (*See Motors, Electric*)

STRAINERS, Dirt
Barnes & Jones, Inc., 968
Crane Company, 998-999
C. A. Dunham Co., 970-971
Grinnell Co., Inc., 972-974, 1034
Henry Valve Company, 947
Hoffman Specialty Co., Inc., 976-977
Mueller Steam Specialty Co., Inc., 1036
Sarco Company, Inc., 980-981
Spence Engineering Co., 961
Staynew Filter Corp., 874-875
Trane Company, The, 858-859
Warren Webster & Co., 984-987
Wright-Austin Co., 1037

STRAINERS, Oil
Automatic Products Corp., 939
Crane Company, 998-999
Detroit Lubricator Co., 942-943
General Electric Company, 916-917
Henry Valve Company, 947
Kieley & Mueller, Inc., 1035
Mueller Steam Specialty Co., Inc., 1036
Sarco Company, Inc., 980-981
Spence Engineering Co., 961
Staynew Filter Corp., 874-875
Todd Combustion Equipment, Inc., 1017
Wright-Austin Co., 1037

STRAINERS, Refrigerant
Alco Valve Company, 938
Frick Company, Inc., 891
Henry Valve Company, 947
Westinghouse Elec. & Mfg. Co., 826-827, 876

STRAINERS, Steam
Crane Company, 998-999
Detroit Lubricator Co., 942-943
Illinois Engineering Co., 978-979
Kieley & Mueller, Inc., 1035
Mueller Steam Specialty Co., Inc., 1036
Powers Regulator Co., 958-959
Sarco Company, Inc., 980-981
Spence Engineering Co., 961
Trane Company, The, 858-859
Wright-Austin Co., 1037

STRAINERS, Water
Crane Company, 998-999
Detroit Lubricator Co., 942-943
Illinois Engineering Co., 978-977
Kieley & Mueller, Inc., 1035
McDonnell & Miller, 990-991
Mueller Steam Specialty Co., Inc., 1036
Powers Regulator Co., 958-959
Sarco Company, Inc., 980-981
Spence Engineering Co., 961
Staynew Company, Inc., 874-875
Wright-Austin Co., 1037
Yarnall-Waring Co., 1038

SWITCHES, Float
Alco Valve Company, 938

SWITCHES, Flow Control
McDonnell & Miller, 990-991

TANK COILS (*See Coils, Tank*)

TANK COVERING (*See Covering, Pipes and Surfaces*)

TANK HEATERS (*See Heaters, Tank*)

TANKS, Blow-off
Brownell Company, The, 1018
Farrar & Trefts, Inc., 997
Kewanee Boiler Corp., 1002-1005

TANKS, Pressure
Baker Ice Machine Co., 888
Bell and Gossett Co., 969
Bethlehem Steel Co., 934
Binks Manufacturing Co., 880
Brownell Company, The, 1018
Burnham Boiler Corp., 994-995
Farrar & Trefts, Inc., 997
Frick Company, 891
Kewanee Boiler Corp., 1002-1005

TANKS, Storage
American Radiator & Standard Sanitary Corp., 992-993
E. B. Badger & Sons Co., 1030
Bethlehem Steel Co., 934
Brownell Company, The, 1018
Burnham Boiler Corp., 994-995
Farrar & Trefts, Inc., 997
Frick Company, 891
Kewanee Boiler Corp., 1002-1005

TEMPERATURE CONTROL

THERMOMETERS, Distance Type

THERMOMETERS, Indicating

THERMOMETERS, Recording

THERMOSTATS

TIN PLATE

TOWERS, Cooling (*See Cooling Towers*)

TRANSFORMERS

TRAPS, Bucket

TRAPS, Float

TRAPS, Float and Thermostatic

TRAPS, Radiator

TRAPS, Return

TRAPS, Scale

TRAPS, Steam

TRAPS, Thermostatic

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Roll of Membership

AMERICAN SOCIETY *of*
HEATING *and* VENTILATING ENGINEERS

1941

Contains Lists of Members Arranged Alphabetically and Geographically, also Lists of Officers and Committees, Past Officers and Local Chapter Officers

Corrected to January 15, 1941

Published at the Headquarters of the Society
51 Madison Avenue, New York, N. Y.

ENGINEERS OF HUMAN COMFORT

THE Heating, Ventilating and Air Conditioning Engineer through his work and research brings to our homes, our offices and our factories, in both summer and winter, that climate best suited to our comfort and health. He is truly an *Engineer of Human Comfort.*

In September 1894, a little group of engineers, educators and manufacturers gathered in New York and agreed that the great art of heating and ventilating deserved and required recognition as an essential, distinctive and highly specialized division of modern engineering. These men realized the basic importance of heating and ventilating as the primary element in the well-being of civilized mankind, living and working mostly indoors.

They foresaw the need for research and one of the first acts of the organized body was to establish a Committee on Standards. That the Charter Members had great faith in their enterprise is evident, although little did they dream that progress would be so rapid in their profession.

During the intervening years since that little group of 75 pioneers unfurled the banner of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—3,032 of the real leaders of thought and action in heating, ventilating, and air conditioning have gathered about that standard and carried it proudly before them far along the way of real accomplishment. They may be identified among engineering groups by the distinctive emblem which was adopted by the Charter Members.

The first Annual Meeting was held in New York, N. Y., January 22-24, 1895, and the organization was incorporated under the laws of the State.

The Society now has 3,032 members on its rolls, including engineers, educators, scientists, physicians, architects, contractors, and leaders of industry. There are four classes of active members, namely: MEMBER, ASSOCIATE, JUNIOR and STUDENT.

The management of the Society is entrusted to 4 Officers and a Council of 13 members. Continuity of policy is insured by electing 4 men annually for a 3-year term and retaining the retiring president on the Council for 1 year. Research work is in charge of the Committee on Research consisting of 15 members, 5 being elected annually for a period of 3 years.

The three major activities of the Society are: Membership Service, Publication and Research, the record of its accomplishments being permanently recorded in the annual TRANSACTIONS.

Headquarters of the Society are maintained in The New York Life Building, 51 Madison Avenue, New York, N. Y.

Special Committees

Committee on Admission and Advancement: L. W. Moon, *Chairman* (*two years*); Earle W. Gray (*three years*); C. H. Pesterfield (*one year*).

Committee on Constitution and By-Laws: F. C. McIntosh, *Chairman;* M. C. Beman, W. E. Stark.

A.S.H.V.E.—*I.E.S. Committee on Lighting in Air Conditioning:* H. M. Sharp, *Chairman;* E. E. Ashley, A. A. Brainerd, A. D. Cameron, F. H. Faust, W. F. Friend, W. E. Stark and Walter Sturrock.

A.S.H.V.E—*A.S.T.M. Committee on Thermal Conductivity:* F. C. Houghten, *Chairman;* C. B. Bradley, H. C. Dickinson, R. H. Heilman, E. R. Queer, F. B. Rowley, T. S. Taylor and G. B. Wilkes.

F. Paul Anderson Award Committee: W. L. Fleisher, *Chairman;* D. S. Boyden, John Howatt, W. T. Jones and Perry West.

Publication Committee: W. H. Severns, *Chairman* (*one year*); P. Nicholls (*two years*); James Holt (*three years*).

Guide Publication Committee: S. S. Sanford, *Chairman;* P. D. Close, C. M. Humphreys, A. J. Offner and M. F. Rather.

Speakers Bureau Committee: W. A. Russell, *Chairman;* L. F. Kent, M. F. Rather, A. J. Rummel, A. F. H. Scott, G. M. Simonson and William Worton.

Committee to Study the Method of Selecting Society Officers and Council Members: H. H. Erickson, *Chairman;* Tom Brown, R. H. Carpenter, W. H. Driscoll, F. C. McIntosh and W. A. Russell.

Advisory Committee for Pacific Heating and Air Conditioning Exposition: W. L. Fleisher, *Chairman;* H. H. Douglas, E. O. Eastwood, J. H. Gumz, M. J. Hauan, Daniel Hayes, J. E. McNevin, N. H. Peterson, T. E. Taylor and Dr. B. M. Woods.

Committee on Code for Testing Surface Coils for Heating and Cooling: Tom Brown, *Chairman;* C. A. McKeeman, *Vice-Chairman;* E. L. Hogan, H. F. Hutzel, L. E. Moody, A. F. Nass, S. F. Nicoll, M. Noble, L. P. Saunders and D. C. Wiley.

Committee on Safety Regulations for Heating, Ventilating and Air Conditioning Systems: N. A. Hollister, *Chairman;* F. H. Buzzard, B. F. McLouth, G. P. Nachman and C. H. Randolph.

Nominating Committee

Chapter	*Representative*	*Alternate*
Atlanta	S. W. Boyd	L. F. Kent
Cincinnati	R. E. Kramig, Jr.	H. E. Sproull
Golden Gate	G. M. Simonson	J. A. Hill
Illinois	Tom Brown	M. W. Bishop
Kansas City	W. A. Russell	M. A. Disney
Manitoba	G. C. Davis	R. L. Kent
Massachusetts	H. C. Moore	D. S. Boyden
Michigan	W. C. Randall	G. H. Tuttle
Western Michigan	B. F. McLouth	L. G. Miller
Minnesota	R. E. Backstrom	H. M. Betts
Montreal	C. W. Johnson	F. G. Phipps
New York	O. O. Oaks	C. S. Pabst
Western New York	L. P. Saunders	C. A. Gifford
North Carolina	R. B. Rice	E. R. Harding
Northern Ohio	C. M. H. Kaercher	C. F. Eveleth
Ontario	H. R. Roth	C. Tasker
Oregon	E. E. Carroll	E. C. Willey
Pacific Northwest	M. J. Hauan	S. D. Peterson
Philadelphia	H. H. Erickson	R. F. Junger
Pittsburgh	J. F. Collins, Jr.	F. C. McIntosh
St. Louis	C. F. Boester, Jr.	J. H. Carter
Southern California	J. F. Park	H. H. Douglas
South Texas	D. S. Cooper	A. J. Rummel
Washington, D. C.	S. P. Eagleton	

COMMITTEE ON RESEARCH

A. E. STACEY, JR., *Chairman*

C.-E. A. WINSLOW, *Vice-Chairman*
F. C. HOUGHTEN, *Director*
J. H. WALKER, *Technical Adviser*
A. C. FIELDNER, *Ex-Officio Member*

One Year	*Two Years*	*Three Years*
M. K. FAHNESTOCK	R. E. DALY	PHILIP DRINKER
W. L. FLEISHER	W. A. DANIELSON	AXEL MARIN
R. J. TENKONOHY	H. J. ROSE	A. E. STACEY, JR.
T. H. URDAHL	C. TASKER	J. H. VAN ALSBURG
B. M. WOODS	C.-E. A. WINSLOW	J. H. WALKER

Executive Committee

A. E. STACEY, JR., *Chairman*

M. K. FAHNESTOCK   T. H. URDAHL   C.-E. A. WINSLOW   B. M. WOODS

Finance Committee

E. P. HECKEL, *Chairman*

J. A. GOFF   H. B. HEDGES   J. A. KIESLING   C. H. RANDOLPH

Technical Advisory Committees

Sensations of Comfort—Thomas Chester, *Chairman;* C. R. Bellamy, E. P. Heckel, W. S. Kilpatrick, Dr. W. J. McConnell, A. B. Newton, B. F. Raber, C. Tasker*.

Physiological Reactions—C.-E. A. Winslow*, *Chairman;* Dr. T. Bedford, Dr. E. F. DuBois, Dr. M. B. Ferderber, E. P. Heckel, John Howatt, Dr. R. W. Keeton, C. S. Leopold, André Missenard, Dr. R. R. Sayers, Dr. Charles Sheard, C. Tasker*.

Treatment of Disease—Dr. M. B. Ferderber, *Chairman;* Dr. C. J. Barone, Dr. A. R. Behnke, Dr. B. Z. Cashman, W. L. Fleisher*, Dr. T. L. Hazlett, C. S. Leopold, Dr. C. D. Selby, Dr. A. W. Sherrill, Dr. H. F. Smyth, Dr. C. S. Stephenson, Dr. B. L. Vosburgh.

Air Conditioning in Industry—W. L. Fleisher*, *Chairman;* Philip Drinker*, Dr. Leonard Greenburg, H. P. Greenwald, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, H. B. Matzen, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, Dr. Charles Sheard, C. Tasker*, R. M. Watt, Jr.

Air Pollution and Purification—C.-E. A. Winslow*, *General Chariman.*

(*a*) *Air Pollution*—H. J. Rose*, *Chairman;* Philip Drinker*, H. B. Meller, J. S. Owens, Sol Pincus, R. J. Tenkonohy*, E. C. Webb, E. H. Whitlock.

(*b*) *Removal Atmospheric Impurities*—Dr. Leonard Greenburg, *Chairman;* R. S. Dill, Theodore Hatch, L. R. Koller, F. H. Munkelt, G. W. Penney, Dr. E. B. Phelps, F. B. Rowley, C. Tasker*, W. O. Vedder, J. H. Waggoner, R. P. Warren, W. F. Wells, Dr. Rene Wiener.

Radiation and Comfort—J. C. Fitts, *Chairman;* A. H. Barker, L. M. K. Boelter, R. E. Daly*, E. R. Gurney, L. N. Hunter, A. P. Kratz, C. S. Leopold, D. W. Nelson, W. J. Olvany, G. W. Penney, W. R. Rhoton, C.-E. A. Winslow*.

Instruments—D. W. Nelson, *Chairman;* F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, M. K. Fahnestock*, A. P. Gagge, J. A. Goff, F. W. Reichelderfer, G. L. Tuve, C. P. Yaglou.

Weather Design Conditions—T. H. Urdahl*, *Chairman;* J. C. Albright, P. D. Close, John Everetts, Jr., C. M. Humphreys, J. B. Kincer, O. A. Kinzer, J. W. O'Neill, F. W. Reichelderfer.

Radiation with Gravity Air Circulation—M. K. Fahnestock*, *Chairman;* R. E. Daly*, R. S. Dill, A. G. Dixon, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, T. A. Novotney, W. A. Rowe.

Heat Transfer of Finned Tubes with Forced Air Circulation—W. E. Heibel, *Chairman;* William Goodman, H. F. Hutzel, Ferdinand Jehle, S. F. Nicoll, R. H. Norris, L. P. Saunders, R. J. Tenkonohy*, G. L. Tuve, C. F. Wood.

Cooling Load in Summer Air Conditioning—C. M. Ashley, *Chairman;* John Everetts, Jr., M. G. Kershaw, A. E. Knapp, C. S. Leopold, L. S. Morse, R. M. Stikeleather, J. H. Walker*.

Solid Fuels—R. A. Sherman, *Chairman;* C. E. Bronson, W. A. Danielson*, R. S. Dill, H. N. Eavenson, A. C. Fieldner, L. N. Hunter, A. J. Johnson, R. E. Kerr, H. K. Kugel, F. L. Meyer, P. Nicholls, V. F. Parry, H. J. Rose*, L. E. Seeley, C. E. Shaffer, C. Tasker*, E. C. Webb.

*Member of Committee on Research.

Summer Air Conditioning for Residences—M. K. Fahnestock*, *Chairman;* C. F. Boester, E. A. Brandt, John Everetts, Jr., H. F. Hutzel, E. D. Milener, K. W. Miller, F. G. Sedgwick, J. H. Walker*.

Air Distribution and Air Friction—J. H. Van Alsburg*, *Chairman;* S. H. Downs, A. E. Hershey, R. D. Madison, L. G. Miller, D. W. Nelson, C. H. Randolph, D. J. Stewart, M. C. Stuart, Ernest Szekely, R. J. Tenkonohy*, G. L. Tuve.

Heat Requirements of Buildings—P. D. Close, *Chairman;* E. K. Campbell, J. F. S. Collins, Jr., E. F. Dawson, W. H. Driscoll, H. M. Hart, E. C. Lloyd, H. H. Mather, C. H. Pesterfield, F. B. Rowley, R. K. Thulman, J. H. Walker*.

Air Conditioning Requirements of Glass—M. L. Carr, *Chairman;* C. M. Ashley, L. T. Avery, F. L. Bishop, W. A. Danielson*, H. C. Dickinson, J. D. Edwards, J. E. Frazier, S. O. Hall, E. H. Hobbie, C. L. Kribs, Jr., Axel Marin*, R. A. Miller, F. W. Parkinson, J. H. Plummer, W. C. Randall, L. T. Sherwood, J. T. Staples, G. B. Watkins, F. C. Weinert.

Insulation—R. T. Miller, *Chairman;* E. A. Allcut, R. E. Backstrom, Wharton Clay, R. E. Daly*, W. A. Danielson*, H. C. Dickinson, J. D. Edwards, W. V. Hukill, E. C. Lloyd, Paul McDermott, E. W. McMullen, W. T. Miller, E. R. Queer, F. B. Rowley, W. S. Steele, J. H. Waggoner, G. B. Wilkes.

Sound Control—J. S. Parkinson, *Chairman;* E. L. Anderson, C. M. Ashley, A. L. Kimball, V. O. Knudsen, R. D. Madison, R. F. Norris, C. H. Randolph, A. E. Stacey, Jr.*, G. T. Stanton, A. G. Sutcliffe, R. M. Watt, Jr.

Cooling Towers, Evaporative Condensers and Spray Ponds—B. M. Woods*, *Chairman;* J. C. Albright, W. W. Cockins, S. C. Coey, E. H. Hyde, E. H. Kendall, S. R. Lewis, J. Lichtenstein, H. B. Nottage, J. F. Park, E. T. Selig, Jr., E. H. Taze.

Psychrometry—F. R. Bichowsky, *Chairman;* D. B. Brooks, W. H. Carrier, H. C. Dickinson, A. W. Gauger, J. A. Goff, William Goodman, A. M. Greene, Jr., L. P. Harrison, F. G. Keyes, A. P. Kratz, D. M. Little, Axel Marin*, D. W. Nelson, W. M. Sawdon, F. O. Urban.

Corrosion—A. R. Mumford, *Chairman;* H. E. Adams, J. F. Barkley, L. F. Collins, W. H. Driscoll, T. J. Finnegan, W. Z. Friend, R. R. Seeber, E. T. Selig, Jr., F. N. Speller, J. H. Walker*.

Flow of Fluids Through Pipes and Fittings—S. R. Lewis, *Chairman;* L. A. Cherry, G. C. Davis, T. M. Dugan, Earle W. Gray, R. T. Kern, H. A. Lockhart, Axel Marin*, R. F. Taylor, E. L. Weber.

---

*Member of Committee on Research.

Officers of Local Chapters, 1940-41

Atlanta
Headquarters, Atlanta, Ga.
Meets: First Monday in Month
President, T. T. Tucker
260 Peachtree St., N. W.
Secretary, L. F. Kent
P. O. Box 1673

Cincinnati
Headquarters, Cincinnati, Ohio
Meets: Second Tuesday in Month
President, R. E. Kramig, Jr.
2224 East 14th St.
Secretary, C. E. Hust
Fourth and Main Sts.

Connecticut
Headquarters, Hew Haven, Conn.
President, L. E. Seeley
Yale School of Engrg.
Secretary, E. J. Rodee
290 Congress St.

Delta
Headquarters, New Orleans, La.
President, G. E. May
317 Baronne St.
Secretary, F. G. Burns
317 Baronne St.

Golden Gate
Headquarters, San Francisco, Calif.
Meets: First Tuesday in Month
President, N. H. Peterson
1129 Folsom St.
Secretary, J. A. Hill
245 Market St.

Illinois
Headquarters, Chicago, Ill.
Meets: Second Monday in Month
President, V. L. Sherman
643 Hillside Ave., Glen Ellyn
Secretary, M. W. Bishop
228 N. La Salle St., Chicago

Iowa
Headquarters, Des Moines, Iowa
Meets: Second Tuesday in Month
President, R. A. Norman
715 Ridgewood Ave., Ames
Secretary, N. B. Delavan
414–12th St., Des Moines

Kansas City
Headquarters, Kansas City, Mo.
Meets: Second Monday in Month
President, W. L. Cassell
912 Baltimore Ave.
Secretary, E. M. Jolley
3001 Fairfax Rd., Kansas City, Kans.

Officers and List of Chapters, 1940-41—(*Continued*)

Manitoba

Headquarters, Winnipeg, Man.
Meets: Third Thursday in Month
President, P. L. CHARLES
406 Tribune Bldg.
Secretary, IVAN McDONALD
44 Princess St.

Massachusetts

Headquarters, Boston, Mass.
Meets: Third Tuesday in Month
President, C. P. YAGLOU
55 Shattuck St.
Secretary, C. M. F. PETERSON
77 Massachusetts Ave., Cambridge

Michigan

Headquarters, Detroit, Mich.
Meets: First Monday after 10th of Month
President, G. H. TUTTLE
2000 Second Ave.
Secretary, W. H. OLD
1761 Forest Ave., W.

Minnesota

Headquarters, Minneapolis, Minn.
Meets: First Monday in Month
President, M. H. BJERKEN
533 S. Seventh St.
Secretary, D. B. ANDERSON
1981 First National Bank Bldg.
St. Paul, Minn.

Montreal

Headquarters, Montreal, Que.
Meets: Third Monday in Month
President, C. W. JOHNSON
630 Dorchester St., W.
Secretary, F. G. PHIPPS
5431 Earnscliffe Ave.

New York

Headquarters, New York, N. Y.
Meets: Third Monday in Month
President, C. S. PABST
55 West 42nd St.
Secretary, T. W. REYNOLDS
100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

North Carolina

Headquarters, Durham, N. C.
President, ARVIN PAGE
1001 S. Marshall St., Winston-Salen, N. C.
Secretary, T. C. COOKE
400 E. Peabody St., Durham, N. C.

North Texas

Headquarters, Dallas, Tex.
Meets: Second Monday in Month
President, L. S. GILBERT
1314 Liberty Bank Bldg.
Secretary, T. H. ANSPACHER
702 Tower Petroleum Bldg.

Northern Ohio

Headquarters, Cleveland, Ohio
Meets: Second Monday in Month
President, C. A. McKEEMAN
Case School of Applied Science
Secretary, C. M. H. KAERCHER
3030 Euclid Ave.

Oklahoma

Headquarters, Oklahoma City, Okla.
Meets: Second Monday in Month
President, S. L. ROLLAND
321 N. Harvey Ave.
Secretary, A. R. MORIN
Box 1197

Ontario

Headquarters, Toronto, Ont.
Meets: First Monday in Month
President, J. W. O'NEILL
8 Springmount Ave.
Secretary, H. R. ROTH
57 Bloor St., W.

Oregon

Headquarters, Portland, Ore.
Meets: Thursday after First Tuesday in Month
President, T. E. TAYLOR
307 Postal Bldg.
Secretary, B. W. MOORE
7408 Southeast 35th Ave.

Pacific Northwest

Headquarters, Seattle, Wash.
Meets: Second Tuesday in Month
President, M. J. HAUAN
3412–16th South
Secretary, H. T. GRIFFITH
324-1411 Fourth Ave. Bldg.

Philadelphia

Headquarters, Philadelphia, Pa.
Meets: Second Thursday in Month
President, C. B. EASTMAN
530 Brookview Lane, Brookline, Pa.
Secretary, A. C. CALDWELL
550 South 48th St., Philadelphia, Pa.

Pittsburgh

Headquarters, Pittsburgh, Pa.
Meets: Second Monday in Month
President, F. C. McINTOSH
1238 Brighton Rd.
Secretary, T. F. ROCKWELL
Carnegie Institute of Technology

St. Louis

Headquarters, St. Louis, Mo.
Meets: First Tuesday in Month
President, C. E. HARTWEIN
231 W. Lockwood Ave., Webster Groves, Mo.
Secretary, C. F. BOESTER
101 E. Essex, Kirkwood, Mo.

South Texas

Headquarters, Houston, Tex.
Meets: Third Friday in Month
President, A. J. RUMMEL
201 N. St. Mary's St., San Antonio
Secretary, R. M. SPENCER
329 M & M Bldg., Houston

Southern California

Headquarters, Los Angeles, Calif.
Meets: Second Wednesday in Month
President, H. H. DOUGLAS
601 W. Fifth St.
Secretary, H. H. BULLOCK
212 N. Vignes St.

Washington, D. C.

Headquarters, Washington, D. C.
Meets: Second Wednesday in Month
President, F. E. SPURNEY
28 W. Baltimore St., Kensington, Md.
Secretary, E. H. LLOYD
1626 K St., N. W., Washington, D. C.

Western Michigan

Headquarters, Grand Rapids, Mich.
Meets: Second Monday in Month
President, T. D. STAFFORD
954 Ogden Ave., S. E.
Secretary, F. C. WARREN
200 Division Ave., N.

Western New York

Headquarters, Buffalo, N. Y.
Meets: Second Monday in Month
President, C. A. GIFFORD
78 Roycroft Blvd., Snyder, N. Y.
Secretary, S. M. QUACKENBUSH
597 Michigan Ave., Buffalo, N. Y.

Wisconsin

Headquarters, Milwaukee, Wis.
Meets: Third Monday in Month
President, A. S. KRENZ
5114 W. Center St.
Secretary, H. W. SCHREIBER
507 E. Michigan St.

HOW TO APPLY FOR MEMBERSHIP

The real accomplishments of life are usually measured by the service one has rendered to his fellows and the true cultural refinement of mind, the finest sense of personal and professional ethics, factors transcending all material elements in what man calls "success," are developed through association with those of high ideals and cherished ambitions in the same field of activity. THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS offers to him whose work is definitely within its province an opportunity for such association and an opportunity for real service to his profession.

Every man in the heating, ventilating and air conditioning profession needs the Society—

1—Because of the contacts that it brings through national and local meetings.

2—Because of the information supplied by Society Publications.

3—Because of the opportunities that research reveals in new applications for engineering services and equipment.

4—Because of the satisfaction to be derived in contributing to human comfort and well being.

A Candidate must make application on the printed form "Membership Application" which is available at the headquarters office or from Chapter Officers and members. A statement of qualifications and engineering experience is required and four members must act as sponsors except under certain conditions noted in *Article B*-III of the By-Laws.

Initiation Fees for 1941 are: Members and Associate Members $10.00; Junior Members $5.00. The Initiation Fee must accompany application.

For 1941 the annual dues of the Society are: Members and Associate Members $18.00; Junior Members $10.00; and Student Members $3.00. Dues of new members are pro-rated on a quarterly basis.

ARTICLE C-II—MEMBERSHIP

Section 1. Persons connected with the arts and sciences related to heating, ventilating or air conditioning are eligible for admission into the Society.

Section 4. A Member shall be thirty (30) years of age or over and shall be a person of experience in the science of heat transfer in its application to the art of heating, ventilating or air conditioning, and shall have been in active practice of his profession and in responsible charge of important work for five (5) years and shall be qualified to design as well as to direct such engineering work.

Section 5. A Junior Member shall be a person over twenty (20) years and under thirty (30) years of age, who has been actively engaged in the work of heating, ventilating or air conditioning for three (3) years, or is a graduate of a school of engineering of recognized standing.

Section 6. An Associate Member shall be twenty-five (25) years of age or over. He need not be an engineer, but must have been so connected with some branch of engineering or the art of heating, ventilating, air conditioning or the industries relating thereto, that he may be considered as qualified to co-operate with heating and ventilating engineers in the advancement of professional knowledge.

Section 7. A Student Member shall be a person between the ages of 18 and 25 years, who is regularly attending courses in an engineering college or technical school at the time of applying for membership.

Roll of Membership

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1941

(Corrected to January 15, 1941)

HONORARY MEMBERS

BALDWIN, WM. J. (1915), New York, N. Y. (Deceased May 7, 1924.)
BILLINGS, DR. J. S. (1896), New York, N. Y. (Deceased March 10, 1913.)
BOLTON, REGINALD PELHAM (1897), New York, N. Y.
BRECKENRIDGE, L. P. (1920), North Ferrisburg, Vt. (Deceased August 22, 1940.)
GORMLY, JOHN (Charter Member), Norristown, Pa. (Deceased January 31, 1929.)
NEWTON, C. W. (Charter Member), Baltimore, Md. (Deceased August 6, 1920.)
HOOD, O. P. (1929), Washington, D. C. (Deceased April 22, 1937.)
JELLETT, STEWART A. (Charter Member), (Presidential Member), Philadelphia, Pa. (Deceased April 5, 1935.)

LIST OF MEMBERS Arranged Alphabetically

(Asterisk indicates authorship of papers)

(*M* 1923; *A* 1918; *J* 1916) indicates, Election as Member 1923; Associate 1918; Junior 1916.
(Pres. 1923) indicates, Elected President in 1923 and is now a Presidential Member.

A

ABBOTT, Thomas J. (*M* 1938) Vice-Pres. (for mail) Geo. C. Abbott, Ltd., 119 Harbord St., and 42 Ardmore Rd., Toronto., Ont., Canada.

ABRAMS, Abraham (*M* 1927; *J* 1924) Pres., Abbey Heating Co., Inc., 81 Centre Ave., and (for mail) 100 Clove Rd., New Rochelle, N. Y.

ABRAMSON, Ralph J. (*A* 1938) Estimating & Design Engr., Hipskind Heating & Plumbing Co., 1725 Winter St., and (for mail) 2040 Henrietta St., Fort Wayne, Ind.

ACHESON, Albert R. (*M* 1919) Prof., Mech. Engrg., Syracuse University, and 501 Eckel Theatre Bldg., and (for mail) 852 Ostrom Ave., Syracuse, N. Y.

ADAIR, James S. (*A* 1941; *J* 1940) Owner (for mail) Southern Equipment Co., 205 Balter Bldg., and 520 Rue Chartres, Vieux Carre, New Orleans, La.

ADAM, Ray W. (*A* 1938) Owner (for mail) William A. Adam Co., 8810 Grinnell Ave., and 5911 Courville Ave., Detroit, Mich.

ADAMS, Bruce P. (*A* 1936) Gen. Mgr. (for mail) McDonnell & Miller, 400 N. Michigan Ave., Chicago, and 2211 Greenwood Ave., Wilmette, Ill.

ADAMS, Chester Z. (*M* 1939) Br. Mgr. (for mail) Ilg Electric Ventilating Co., Box 1356, 504 Piedmont Bldg., and A4 Dolly Madison Apts., Greensboro, N. C.

ADAMS, Eugene E. (*A* 1938) Sales Engr. (for mail) Garden City Fan Co., Room 1508A, 55 West 42nd St., New York, and 35-46 79th St., Jackson Heights, L. I., N. Y.

ADAMS, Frank L. (*M* 1939) Htg. & Air Cond. Engr. (for mail) Public Service Co. of Colorado, 900-15th St., and 1185 Grape St., Denver, Colo.

ADAMS, Harold E. (*M* 1930) Chief Engr. (for mail) The Nash Engineering Co., South Norwalk, and Merrill Heights, Norwalk, Conn.

ADAMS, Neil D. (*M* 1929; *A* 1925; *J* 1922) (Council, 1938-40) Supt. (for mail) Franklin Heating Station, 220 Second Ave. S.W., and 836 Eighth Ave. S.W., Rochester, Minn.

ADDAMS, Homer (*Charter Member*; *Life Member*) (*Presidential Member*) (Pres., 1924; 1st Vice-Pres., 1923; Treas., 1915-22; Council, 1915-25) Pres., Fitzgibbons Boiler Co., Inc., and Kewanee Boiler Co., Inc., 101 Park Ave., New York, N. Y.

ADDINGTON, Herbert B. (*M* 1938) Consulting Engr. (for mail) 13 East 37th St., New York, and 25 Lafayette Ave., Brooklyn, N. Y.

ADDINGTON, H. M. (*M* 1939) Assoc. Engr., War Dept., Furniture Warehouse, Ft. Myer, Va., and (for mail) 2025 Eye St., N.W., Washington, D. C.

ADEMA, George E. (*M* 1939) Prop. (for mail) N. M. Adema & Son, 39 W. Balcom St., and 260 Warren Rd., Buffalo, N. Y.

ADLAM, T. Napier (*M* 1932) Vice-Pres. and Gen. Mgr., Sarco Manufacturing Co., 183 Madison Ave., New York, N. Y., and (for mail) 64 Wellington Ave., West Orange, N. J.

ADLER, Herman (*J* 1940; *S* 1938) Purchasing Agent, Armo Cooling & Ventilating Co., Inc., 30 West 15th St., and (for mail) 485 Central Park West, New York, N. Y.

AEBERLY, John J.* (*M* 1928) (Council, 1937-39) Chief of Div. of Htg., Vtg. and Ind. Sanitation, Chicago Board of Health, 707 City Hall, and (for mail) 6225 N. Newcastle Ave., Norwood Park P. O., Chicago, Ill.

AHEARN, William J. (*M* 1929) Htg. and Vtg. Engr., 131 Windermere Rd., Auburndale, Mass.

AHLFF, Albert A. (*M* 1923; *A* 1918) Sales Supvr., Tuttle & Bailey, Inc., 1600 Arch St., and (for mail) 1421 Greywall Lane, Overbrook Hills, Philadelphia, Pa.

AHRENS, Clarence F. (*A* 1940) Sales Engr., R. A. Dubuque Supply Co., 3960 Duncan Ave., and (for mail) 4151 Toenges Ave., St. Louis, Mo.

AINSWORTH, Samuel E. (*A* 1939) Sales Engr., Roche Newton & Co., 1316 Texas Ave., and (for mail) 1524 25th St., Lubbock, Tex.

AKERMAN, Joseph Reid (*A* 1941; *J* 1937) Engr. (for mail) Phoenix Oil Co., 700 Twiggs St., and 831-15th St., Augusta, Ga.

AKERS, Arthur W. (*A* 1940) Dir. (for mail) Technicians Institute, 244 West 14th St., New York, N. Y.

AKERS, George W. (*M* 1929) Secy.-Treas., George W. Akers Co., 16525 Woodward Ave., Detroit, and (for mail) R. F. D. No. 3, Birmingham, Mich.

ALBEN, Edward A. (*A* 1939) Htg. Contractor (for mail) Alben Equipment Co., 114 E. Tenth St., and R-4, Box 282C, Vancouver, Wash.

ALBRIGHT, C. Barton (*J* 1938) Air Cond. Engr., 69 Lane Ave., Caldwell, N. J.

ALEXANDER, Samuel W. (*M* 1935) Pres., S. W. Alexander Co., Ltd., 2494 Danforth Ave., and (for mail) 124 Kingsmount Park Rd., Toronto, Ont., Canada.

ALFAGEME, Braulio (*M* 1935) Engr. & Mgr., B. Alfageme, Almagro 1, Madrid, Spain.

ALFERY, Henry F. (*M* 1938) Chief Engr. (for mail) Milwaukee Gas Specialty Co., 2025 W. Clybourn St., and 1819 W. Center St., Milwaukee, Wis.

ALFSEN, Nikolai (*M* 1933) Engr., Alfsen & Gunderson, Prinsensgate 2b, Oslo, and (for mail) Utsiktsveien 22, Stabekk, Norway.

ALGREN, Axel B.* (*M* 1930) Asst. Prof. Mech. Engrg., Asst. Dir. Engrg. Experiment Station, University of Minnesota, and (for mail) 5109-17th Ave., S., Minneapolis, Minn.

ALLAN, William (*A* 1938) Pres.-Treas. (for mail) Allan Engineering Co., 724 E. Mason St., and 2735 N. Farwell Ave., Milwaukee, Wis.

ALLCUT, Edgar A.* (*M* 1937) Prof., Mech. Engrg. (for mail) University of Toronto, and 48 Foxbar Rd., Toronto, Ont., Canada.

ALLEN, A. Walter (*M* 1936) Sales Engr., Pease Foundry Co., Ltd., 151 Glen Ave., Ottawa, Ont., Canada.

ALLEN, DeWitt M. (*M* 1936; *J* 1922) Dist. Mgr. Ilg Electric Ventilating Co., 310 Board of Trade Bldg., Kansas City, Mo., and (for mail) 3924 Cambridge, Kansas City, Kan.

ALLEN, William W. (*A* 1936) Pres., American Coolair Corp., Box 2300, Jacksonville, Fla.

ALLENSWORTH, James E. (*S* 1939) Student, Carnegie Institute of Technology, Box 290, Pittsburgh, Pa., and (for mail) Amsterdam, O.

ALLONIER, Howard R. (*A* 1936) Dist. Sales Mgr., John J. Nesbitt Co., 425 W. Town St., Columbus, and (for mail) R. R. No. 1, Powell, O.

ALLSOP, Rowland P. (*A* 1940; *J* 1934) Consulting Engr. (for mail) 1221 Bay St., and 168 Courcellette Ave., Toronto, Ont., Canada.

ALT, Harold L.* (*M* 1913) Mech. Engr., Voorhees, Walker, Foley & Smith, 101 Park Ave., New York, and (for mail) 115-27 225th St., St. Albans, L. I., N. Y.

ALTEMUELLER, George F. (*A* 1940) Chief Engr. (for mail) Homeland Investment Co., 111 Corcoran St., and 2601 Highland Ave., Durham, N. C.

AMBROSE, Eugene R. (*M* 1940) Air Cond. Engr. (for mail) American Gas & Electric Service Corp., 30 Church St., New York, and 49 Sherman St., Brooklyn, N. Y.

AMMERMAN, Andrew S., Jr. (*A* 1941; *J* 1937) Engr. (for mail) Aerofin Corp., Room 544, 111 W. Washington St., and 4332 N. Hermitage Ave., Chicago, Ill.

AMMERMAN, Charles R. (*M* 1916) Consulting Engr. (for mail) 772 Century Bldg., and 3908 Guilford Ave., Indianapolis, Ind.

ANDEREGG, R. H. (*M* 1920) Vice-Pres. & Chief Engr., The Trane Co., and (for mail) 420 N. Losey Blvd., LaCrosse, Wis.

ANDERSON, Carroll S. (*M* 1920) Mgr. (for mail) American Blower Corp., 1105 Architects Bldg., 816 W. Fifth St., Los Angeles, and 4267 Holly Knoll Dr., Hollywood, Calif.

ANDERSON, David B. (*A* 1939; *J* 1936; *S* 1933) Mgr., Sales Engrg. Dept. (for mail) Wood Conversion Co., 1981 First National Bank Bldg., and 1999 Pinehurst Ave., St. Paul, Minn.

ANDERSON, Edwin J. (*A* 1939) Mfrs. Agent (for mail) 14 Smith St., and 274 Lenox, Detroit, Mich.

ANDERSON, Einar (*A* 1940) Sales Engr., Vulcan Iron Works, Ltd., and (for mail) 152 Bannerman Ave., Winnipeg, Man., Canada.

ANDERSON, George A. M. (*A* 1939; *J* 1936) Pres. (for mail) King Ventilating Co., and 717 S. Cedar, Owatonna, Minn.

ANDERSON, John W. (*J* 1937) Engrg. Dept., The Conditioning Co., 368 Broad St., Newark, and (for mail) 548 Westminster Ave., Elizabeth, N. J.

ANGERMEYER, Albert H. (*A* 1936) Owner (for mail) A. H. Angermeyer Plumbing & Heating, 119 N. Commercial St., and 245 Webster St., Neenah, Wis.

ANGUS, Frank M. (*M* 1937) Dist. Sales Mgr., General Refrigeration Corp., Beloit, Wis., and (for mail) 4936 Booth Ave., Kansas City, Kan.

ANGUS, Harry H.* (*M* 1918) (Council, 1927-29) Consulting Engr., 1221 Bay St., and (for mail) 34 Farnham Ave., Toronto, Ont., Canada.

ANOFF, Seymour M. (*J* 1940) Asst. Gen. Mgr. in charge of Prod., Albert Pick Co., Inc., 2159 Pershing Rd., and (for mail) 46 N. Central Ave., Chicago, Ill.

ANSPACHER, Thomas H. (*M* 1939; *J* 1936) Dist. Mgr. (for mail) Buffalo Forge Co., 702 Tower-Petroleum Bldg., and 4512 Arcady, Dallas, Tex.

ANTHES, Lawrence L. (*A* 1935) Pres. (for mail) Imperial Iron Corp., Ltd., 30 Jefferson Ave., and 117 Dowling Ave., Toronto, Ont., Canada.

APT, Sanford R. (*M* 1935) Chief Mech. Engr., Caribbean Architect-Engineer, 41 East 42nd St., New York, and (for mail), 36-20 168th St., Flushing, L. I., N. Y.

ARCHAMBAULT, Joseph A. (*A* 1939) Sales Office Mgr. (for mail) C. A. Dunham Co., Ltd., 22 Wellington St., N., and 55A Council St., Sherbrooke, Que., Canada.

ARCHER, David M. (*M* 1934) Sales Repr. (for mail) Young Radiator Co., 143 Federal St., Boston, and 10 Harding Ave., Braintree, Mass.

ARENBERG, Milton K. (*A* 1920) Pres. (for mail) Robert Barclay, Inc., 122 N. Peoria St., Chicago, and Wildwood Lane, Highland Park, Ill.

ARGUE, Edgar J. (*A* 1935) Sales Engr., Anthes Foundry, Ltd., Saskatchewan Ave., and (for mail) 858 Fleet Ave., Winnipeg, Man., Canada.

ARKLEY, Lorne M. (*M* 1922) Head of Dept., Mech. Engrg. (for mail) Queen's University, and 22 Kensington Ave., Kingston, Ont., Canada.

ARMBRUSTER, Frank T. W. (*M* 1936) Sales Engr., American Radiator & Standard Sanitary Corp., 503 S. Front St., Columbus, and (for mail) 105 First Ave., Waverly, O.

ARMISTEAD, William C. (*M* 1937) Sales Engr. (for mail) 205 Church St., and Murfreesboro Rd., Nashville, Tenn.

ARMOUR, Edson G. (*J* 1940; *S* 1939) Sales Engr. (for mail) Armour's, 156 Market St., and 55-A Sheridan St., Brantford, Ont., Canada.

ARMSPACH, Otto W.* (*M* 1919) Vice-Pres. and Chief Engr., Kroeschell Engineering Co., 215 W. Ontario St., Chicago, and (for mail) 205 S. Summit Ave., Villa Park, Ill.

ARMSTRONG, Charles E. (*M* 1939) Chief Engr. (for mail) Armstrong Heat Control Co., 1626 N. E. Union Ave., and 1307 N. E. 11th Ave., Portland, Ore.

ARMSTRONG, Edward T. (*J* 1941; *S* 1939) Instructor, Mech. Engrg. Dept. (for mail) Clarkson College of Technology, and 22 Main St., Potsdam, N. Y.

ARMSTRONG, Walter J. (*M* 1938) Consulting Engr. (for mail) 1010 St. Catherine St., West, Montreal, and 15 Willow Ave., Westmount, Que., Canada.

ARNDT, Heinrich W. (*A* 1935) 2034 Wrightson Rd., Augusta, Ga.

ARNOLD, Robert S. (*A* 1926; *J* 1922) Owner, Robt. Arnold Sales & Engineering Co., 409 Otis Bldg., and (for mail) 6391 Sherwood Rd., Philadelphia, Pa.

ARONSON, Henry H. (*A* 1939; *J* 1929) Field Engr., Premier Furnace Co., Dowagiac, Mich., and (for mail) 6145 Winthrop Ave., Chicago, Ill.

ARROWSMITH, John O. (*M* 1934) Plant Engr. (for mail) Canadian Kodak Co., Ltd., Toronto 9, and 9 Humberview Rd., Toronto, Ont., Canada.

ARTHUR, John M., Jr. (*M* 1923) Commercial Sales Mgr. (for mail) Kansas City Power & Light Co., 1330 Baltimore Ave., Kansas City, Mo., and 3311 State Ave., Kansas City, Kan.

ASH, Robert S. (*J* 1940) Htg. Engr. (for mail) James B. Clow & Sons, 201 N. Talman Ave., Chicago, and 848 Washington Blvd., Oak Park, Ill.

ASHLEY, Carlyle M.* (*M* 1931) Dir. of Development (for mail) Carrier Corp., S. Geddes St., and 207 Brattle Rd., Syracuse, N. Y.

ASHLEY, Edward E. (*M* 1912) Partner (for mail) Edward E. Ashley, Consulting Engr., 10 East 40th St., New York, N. Y., and Noroton Heights, Conn.

ATHERTON, Alfred E. (*A* 1937) Dir. (for mail) A. E. Atherton & Sons Pty., Ltd., 383 Latrobe St., Melbourne, C. 1, and 39 Ormond Esplanade, Elwood, S. 3, Melbourne, Australia.

ATHERTON, George R. (*M* 1930) Exec. Dept., The Trane Co., and (for mail) 323 South 17th St., LaCrosse, Wis.

AUER, George G. (*A* 1939) Pres. (for mail) Auer Register Co., 3608 Payne Ave., Cleveland, and 1021 Homewood Drive, Lakewood, O.

AUSTIN, William H. (*J* 1940; *S* 1937) Tenderay Engr., Westinghouse Lamp Div., and (for mail) 32 Fremont St., Bloomfield, N. J.

AVERY, Ledyard (*A* 1939) Engr., Carrier Corp., and (for mail) 594 Roberts Ave., Syracuse, N. Y.

AVERY, Lester T. (*M* 1934) Pres. (for mail) Avery Engineering Co., 1906 Euclid Ave., Cleveland, O.

AXEMAN, James E. (*M* 1932; *A* 1931; *J* 1925) Gen. Sales Mgr. (for mail) Spencer Heater Div. and N. Campbell St., Williamsport, Pa.

AY, Edward L. (*J* 1940) Asst. Air Cond. Engr., Annex Library Congress, Second & Pennsylvania Ave. S. E., and (for mail) 2008 Shepherd St. N. E., Washington, D. C.

B

BABER, John E. (*A* 1940) Sales Engr. (for mail) T. C. Heyward, 1408 Independence Bldg., and 1240 Romany Rd., Charlotte, N. C.

BACHMAN, Fred (*M* 1936) Owner (for mail) Fred Bachman, 1608 N. Carlisle St., Philadelphia, and 906 Bell Ave., Yeadon, Pa.

BACHMANN, Arthur J. (*J* 1940; *S* 1939) Preferred Utilities Co., Inc., 5 Irma Ave., Port Washington, and (for mail) 139-15 86th Rd., Jamaica, L. I., N. Y.

BACHOFER, Henry A., Jr. (*J* 1938) Mgr., Htg. and Air Cond. Dept., Mid-West Plumbing & Heating Co., 111 S. Fifth St., and (for mail) 534 S. Eighth St., Salina, Kan.

BACKSTROM, Russell E.* (*A* 1931; *J* 1928) Mgr., Ind. Sales Dept. (for mail) Wood Conversion Co., First National Bank Bldg., and 1655 Hillcrest Ave., St. Paul, Minn.

BACKUS, Theodore H. L. (*M* 1916) Schumacher & Backus, 200-208 Hill St., and (for mail) 1018 Vaughn St., Ann Arbor, Mich.

BADGETT, W. Howard* (*M* 1937; *J* 1932) Capt., Infantry, U. S. Army, Hdq., Houston Military District, 324 Post Office Bldg., Houston, Tex.

BADHE, Jaikrishna M. (*A* 1940) Asst. Engr. (for mail) Volkart Bros., Ballard Estate, Bombay, and Flat No. 11, "Palm View," Gokhale Rd., Dadar, Bombay, India.

BAGGALEY, Walter (*M* 1938) Dist. Mech. Elec. Engr. (for mail) Austin Co., 16112 Euclid Ave., Cleveland, and 3390 Glencairn Rd., Shaker Heights, O.

BAHLMANN, William F. (*A* 1940) Br. Mgr. (for mail) Holland Furnace Co., 700 W. Broad St., and Stratford Hills, Richmond, Va.

BAHNSON, Frederic F.* (*M* 1917) Consulting Engr., The Bahnson Co., 1001 S. Marshall St., Pres., Southern Steel Stampings Inc.,, Cherry St. at Brookstown Ave., and for mail 28 Cascade Ave., Winston-Salem, N. C.

BAILEY, Albert E., Jr. (*A* 1938) Sales Engr., Frigidaire Div., General Motors Sales Corp., 29 Franklin Rd., and (for mail) 1624 Patterson Ave. S. W., Roanoke, Va.

BAILEY, Charles F. (*J* 1939) Virginia Engineering Co., N. O. B., Norfolk, and (for mail) Windsor, Va.

BAILEY, Edward P. (*M* 1925) Sales Engr., Detroit Stoker Co., 5-125 General Motors Bldg., Detroit, and (for mail) 151 Crocker Blvd., Mt. Clemens, Mich.

BAILEY, F. A., Jr. (*A* 1939) Prop., Bailey's, 130 King St., Charleston, S. C.

BAILEY, James Luther (*A* 1940; *J* 1930) Asst. Chief Engr., Parks-Cramer Co., Charlotte, N. C.

BAILEY, W. Mumford (*M* 1930) Managing Dir., British Trane Co., Ltd., Vectair House, Clerkenwell Close, London, E. C. 1, England.

BAIRD, Floyd E. (*M* 1929) So. Dist. Mgr. (for mail) The Trane Co., 314 Palmer Bldg., Atlanta, and 400 Campbell Hill, Marietta, Ga.

BAKER, C. T.* (*M* 1935) Consulting Engr. (for mail) 1070 Spring St. N. W., and 31 The Prado, Atlanta, Ga.

BAKER, Donald L. (*A* 1940) Office Mgr. (for mail) Messrs. Martin & Co., New Delhi, India, and 4 Canning Rd., New Delhi, India, and 1931 Chapel St., New Haven, Conn.

BAKER, George R. (*M* 1936) Pres. (for mail) G. R. Baker Co., Ltd., 224 Adelaide St., W., and 37 Lappin Ave., Toronto, Ont., Canada.

BAKER, Harold S. (*A* 1937) Sales Engr. (for mail) Bakersfield Hardware Co., 2015 Chester Ave., and 241 Jefferson St., Bakersfield, Calif.

BAKER, Harry L., Jr. (*J* 1935) Sales Engr. (for mail), American Blower Corp., 135 Spring St., Rochester, and Van Tassel Apts., N. Tarrytown, N. Y.

BAKER, Irving C. (*M* 1921) Vice-Pres. in Charge of Sales (for mail) Chrysler Corp., Airtemp Div., 1119 Leo St., and Box 386, Route 7, Dayton, O.

BAKER, Roland H. (*M* 1928; *A* 1924) Lt. Commander, U. S. Naval Reserve, U. S. Navy Yard, and (for mail) 74 Revere St., Boston, Mass.

BAKER, Thomas (*M* 1938) 416 Clifton Terrace S., 14th and Clifton St. N. W., Washington, D. C.

BAKER, William C. (*M* 1938) Pres. and Treas. (for mail) Electric Appliances, Inc., 155 Seventh Ave., N., and Westover Drive, Nashville, Tenn.

BAKER, William H., Jr. (*A* 1935) Gen. Mgr. (for mail) Standard Air Conditioning, Inc., 50 West 40th St., and 307 East 44th St., New York, N. Y.

BALDI, Giuseppe (*A* 1936) Engr. (for mail) Compagnia Italiana Westinghouse, Via Pier Carlo Boggio 20, Torino, Italy.

BALDWIN, Karl F., Jr. (*A* 1941; *J* 1938) Engr., McCrea Equipment Co., 516 Second St. N. W., Washington, D. C., and (for mail) 4810 Cedar St., Hyattsville, Md.

BALDWIN, William H. (*M* 1921) Sales Engr. (for mail) C. A. Dunham Co., 5757 Cass Ave., and 2432 Atkinson Ave., Detroit, Mich.

BALL, Frederick T. (*A* 1940) Mgr., Stoker Refrigeration Depts., Canadian Fairbanks Morse Co., Ltd., 324 Main St., and (for mail) 374 Brock St., Winnipeg, Man., Canada.

BALL, William (*A* 1936) Pres. (for mail) Inter-State Heating & Plumbing Co., 521 Southwest Blvd., Kansas City, Mo., and 1026 Shawnee Rd., Kansas City, Kan.
BALLANTYNE, George L. (*A* 1936) Mgr., Htg. Section (for mail) Crane, Ltd., 1170 Beaver Hall Sq., Montreal, and 140 Ballantyne Ave., S., Montreal, W., Que., Canada.
BALLMAN, William H. (*M* 1937) Mgr. & Chief Engr., Air Cond. Div., Westinghouse Electric & Manufacturing Co., 1419 N. Broad St., Philadelphia, Pa.
BALSAM, Charles P. (*M* 1932) Owner, National Home Equipment Co., 11 West 42nd St., New York, and (for mail) 324 Fourth St., Brooklyn, N. Y.
BANACH, C. J. (*J* 1939) Chief Draftsman, Johnson Fan & Blower Corp., 1319 W. Lake St., and (for mail) 1427 N. Leavitt St., Chicago, Ill.
BANKS, John B. (*A* 1937) Br. Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 2405 N. Maryland Ave., and 2928 N. Maryland Ave., Milwaukee, Wis.
BANNER, F. L. Dan (*M* 1937) Dist. Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 3101 Gilham Plaza, and 615 East 73rd Terrace, Kansas City, Mo.
BANOWSKY, Aubra B. (*M* 1938) Commercial Sales Mgr., United Gas Corp., United Gas Bldg., and (for mail) 3735 Ingold, Houston, Tex.
BANTA, Guy L. (*A* 1939) Br. Mgr. (for mail) The Trane Co., 310 Postal Bldg., and Chasselton Apts., Northeast 28th St., Portland, Ore.
BARBIERI, Patrick J. (*J* 1936; *S* 1933) Air Cond. Engr., Armo Cooling & Ventilating Co., 30 West 15th St., and (for mail) 2237 Belmont Ave., New York, N. Y.
BARNARD, M. Everett (*A* 1931; *J* 1929) Sales Engr. (for mail) Carrier Corp., 12 South 12th St., and 380 Vernon Rd., Philadelphia, Pa.
BARNES, Arthur F. (*M* 1920) Owner (for mail) Texas Engineering Co., 726 Electric Bldg., and 3015 Jarrard St., Houston, Tex.
BARNES, Arthur R. (*M* 1924) Chief Engr. (for mail) U. S. Supply Co., 1315 West 12th St., and 326 East 70th Terrace, Kansas City, Mo.
BARNES, Hugh S. (*J* 1940) Sales Engr. (for mail) American Blower Corp., 1211 Commercial Bank Bldg., and 2152 Sherwood Ave., Charlotte, N. C.
BARNES, Lewis L. (*J* 1937) Air Cond. Engr., Carrier Atlanta Corp., 348 Peachtree St., and (for mail) 3995 N. Stratford Rd., Atlanta, Ga.
BARNES, N. W. (*A* 1940) Sales Repr. (for mail) The Fulton Sylphon Co., 568 Wrigley Bldg., and 505 N. Michigan Ave., Chicago, Ill.
BARNES, R. W. (*M* 1939) Htg. & Air Cond. Contractor, 1208 Main Ave., San Antonio, Tex.
BARNES, Walter E. (*M* 1933) Pres., Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain, Boston, and (for mail) 7 Woodlawn Ave., Wellesley Hills, Mass.
BARNES, Willis L. (*J* 1940) Sales Engr. (for mail) The H. L. Thompson Co., 2207 Second National Bank Bldg., and 1111 Richmond Rd., Apt. No. 8, Houston, Tex.
BARNEY, William E. (*M* 1936) Consulting Engr. & Mgr. (for mail) Hydraulic-Press Brick Co., Ohio & Michigan Div., South Park, and 4929 East 108th St., Cleveland, O.
BARNSLEY, Frank Richard (*A* 1936) Mgr., Air Cond. Div. (for mail) Canadian General Electric Co., Ltd., 1000 Beaver Hall Hill, and 4418 Oxford Ave., Montreal, Que., Canada.
BARNUM, Marvin C. (*M* 1930; *A* 1928) Eastern Repr. (for mail) Waterman- Waterbury Co., P. O. Box 284, Suffern, and Cherry Lane, Tallman, N. Y.
BARNUM, Willis E., Jr. (*M* 1933; *J* 1930) Mgr., Air Cond. Div., York Ice Machinery Corp., and (for mail) 154 Rathton Rd., York, Pa.
BARR, George W. (*Life Member*; *M* 1905) (Board of Governors, 1910) Dist. Mgr., Aerofin Corp., 2030 Land Title Bldg., Philadelphia, and (for mail) Woods End, Villa Nova, Pa.
BARRY, Patrick I. (*M* 1920) (Peace Commissioner) Managing Director (for mail) M. Barry, Ltd., 4 Marlboro St., and 8 Sidney Park, Cork, Ireland.
BARTELS, Everett M. (*A* 1941; *J* 1939) Supvr. of Mech. Equip. (for mail) Independent School Dist. 629 Third St., and 824 E. Sheridan St., Des Moines, Ia.
BARTH, Herbert E. (*M* 1920) Vice-Pres. (for mail) American Blower Corp., 6000 Russell St., and Wardell Apts., 15 E. Kirby, Detroit, Mich.
BARTH, John W. (*J* 1939) (for mail) P. O. Box 169, and 410 N. E. Fifth St., Ft. Lauderdale, Fla.
BARTLETT, Amos C. (*M* 1919) Mgr., Htg. and Vtg. Dept. (for mail) B. F. Sturtevant Co., Hyde Park, Boston, and 30 Hollingsworth Ave., Braintree, Mass.
BARTLETT, C. Edwin (*M* 1922) Pres. (for mail) Bartlett & Co., Inc., 3112 North 17th St., and 3111 W. Coulter St., Philadelphia, Pa.
BARTLEY, Henry E. (*M* 1938) Dir. and Works Mgr., Matthews & Yates, Ltd., "Cyclone Works," Swinton, and (for mail) "The Grange," Hospital Rd., Pendlebury, Lancs., England.
BARTON, Edmund H. (*A* 1939) Mgr. Htg. Dept. (for mail) Moosomin Hardware, and Box 308, Moosomin, Sask., Canada.
BARTON, Jay (*M* 1937) Pres., National Manufacturing & Engineering Co., 416 Stephenson Bldg., Detroit, Mich.
BASSETT, James W. (*A* 1938) Sales Engr. (for mail) McQuay, Inc., 2832 E. Grand Blvd., Detroit, and 123 Merrill St., Birmingham, Mich.
BASTEDO, Albert E. (*M* 1919) Vice-Pres. and Treas., Burnham Boiler Corp., Irvington, N. Y.
BASTEDO, George R. (*J* 1937) Elec. Engr., Board of Transportation, 250 Hudson St., New York, and (for mail) 102-36 86th Rd., Richmond Hill, L. I., N. Y.
BATES, John H. (*J* 1941; *S* 1939) Service Dept., 731A, Sears Roebuck & Co., 925 S. Homan St., and (for mail) 3210 Arthington St., Chicago, Ill.
BATTAN, Stuart W. (*M* 1940) Owner (for mail) S. W. Battan, Avondale, Pa., and Kennett Square, Pa.
BAUER, Albert E. (*M* 1935) Chief Engr. (for mail) Brown Sheet Iron & Steel Co., 964 Berry Ave., and 59 S. Victoria St., St. Paul, Minn.
BAUM, Albert L. (*M* 1916) Member of Firm (for mail) Jaros, Baum & Bolles, 415 Lexington Ave., and 600 West 111th St., New York, N. Y.
BAUMGARDNER, Carròll M. (*M* 1928) Br. Mgr. (for mail) United States Radiator Corp., 3254 N. Kilbourn Ave., Chicago, and 416 Cumnor Rd., Kenilworth, Ill.
BAXTER, William E. (*A* 1939) Pres. (for mail) W. E. Baxter, Ltd., 87 Vitre St. W., Montreal, and 89-51st Ave., Lachine, Que., Canada.
BAY, Charles H. (*A* 1938) In charge of Steam Sales (for mail) The Detroit Edison Co., 2000 Second Ave., Rm. 322, and 17323 Wildemere, Detroit, Mich.
BAYLES, Robert W. (*A* 1940) Mgr. Htg. Div., James Morrison Brass Manufacturing Co., Ltd., 276 King St. W., and (for mail) 34 Gormley Ave., Toronto, Ont., Canada.
BAYSE, Harry V. (*M* 1923) Pres. (for mail) American Furnace Co., 2719-31 Delmar Blvd., and 6959 Hancock Ave., St. Louis, Mo.
BEACH, Walter R. (*A* 1936) Sales Engr. (for mail) Cleveland Electric Illuminating Co., 75 Public Sq., Cleveland, and 1185 Yellowstone Rd., Cleveland Heights, O.
BEAIRD, Benjamin J. (*M* 1939) Estimator (for mail) Chas. G. Heyne & Co., 2002 Rothwell, and 1621 Kipling, Houston, Tex.
BEALS, Dowell E. (*A* 1941; *J* 1940) Chief Plbg. Estimator & Engr., Brown & Root, W. S. Bellows, Columbia Construction Co., Naval Air Station, and (for mail) 1205 Florida Ave., Corpus Christi, Tex.
BEAN, George S. (*A* 1935) Mgr. Stoker Div. (for mail) North Western Fuel Co., 2196 University Ave., St. Paul, and 4949-16th Ave. S., Minneapolis, Minn.

BETLEM, Henriette T. (*J* 1934) (for mail) Betlem Heating Co., 1926 East Ave., and 1293 Park Ave., Rochester, N. Y.

BETTS, Howard M. (*M* 1927) Sr. Htg. Engr. (for mail) Dept. of Buildings, City of Minneapolis, 213 City Hall, and 4923 S. Russell Ave., Minneapolis, Minn.

BETZ, Harry D. (*M* 1928) Pres. (for mail) Betz Air Conditioning Corp., 1820 Wyandotte St., and 1610 Valentine Rd., Kansas City, Mo.

BEVINGTON, Curtis H. (*M* 1936) Owner (for mail) C. H. Bevington Co., 600 S. Michigan Ave., Chicago, and 310 S. Stone Ave., La Grange, Ill.

BIBER, Herbert A. (*A* 1937) Engr., Mellon National Bank, 514 Smithfield St., Pittsburgh, and (for mail) 323 Barnes St., Wilkinsburg, Pa.

BIBLE, Hollis U. (*M* 1940) Consulting Engr. (for mail) 6551 Main, and 1609 Marshall, Houston, Tex.

BICHOWSKY, F. Russell* (*M* 1935) Pres., Bichowsky & Rice, 7356 Lake St., River Forest, Ill.

BIERINGER, Fred A. (*A* 1941; *J* 1939) Engr. (for mail) Baldwin-Hill Co., Trenton, N. J., and 1613 East 23rd St., Brooklyn, N. Y.

BIGELOW, Edward S. (*M* 1938) Eastern Sales Mgr., Advance Insulating Co., Inc., Pittsburgh, and (for mail) 413 Jericho Rd., Abington, Pa.

BIGGERS, Richmond H. (*A* 1939) Mfrs. Agent (for mail) 2217 E. Jefferson Ave., and 2237 E. Jefferson Ave., Detroit, Mich.

BIGOLET, Louis (*M* 1939) Louis Bigolet, 40 New Chambers St., New York, and (for mail) 1145 Ocean Pkwy., Brooklyn, N. Y.

BILLINGSLEY, Oliver F., 2nd (*J* 1937) Htg. & Vtg. Engr., Douglas Aircraft Co., 3000 Ocean Park Dr., Santa Monica, and (for mail) P. O. Box 1003, Hollywood, Calif.

BINDER, Charles G. (*M* 1920) Mgr. Htg. Dept., Warren Webster & Co., 17th & Federal St., Camden, and (for mail) 115 Oak Terrace, Merchantville, N. J.

BIRD, Charles (*A* 1934) Gen. Mgr., The Doermann Roehrer Co., 450 E. Pearl St., Cincinnati, O.

BIRD, George L. H. (*A* 1941; *J* 1937) 12 Lawn Rd., London, N. W. 3, England.

BIRKETT, Harold S. (*M* 1940) Htg. Engr., The Brooklyn Union Gas Co., 176 Remsen St., and (for mail) 815 East 19th St., Brooklyn, N. Y.

BISHOP, Charles R. (*Life Member*; *M* 1901) 22 Sagamore Rd., Bronxville, N. Y.

BISHOP, Frederick R. (*M* 1921) Mgr. of Sales, The Brundage Co., Kalamazoo, and (for mail) 8011 Dexter Blvd., Detroit, Mich.

BISHOP, Jacob A. (*M* 1939) Dist. Mgr. (for mail) American Blower Corp., 619 Mercantile Bldg., and 1115 N. Windomere, Dallas, Tex.

BISHOP, Joseph W. (see Special Service Roll, p. 73).

BISHOP, M. W. (*A* 1939; *J* 1935) Sales Engr. (for mail) American Blower Corp., 228 N. LaSalle St., and 2641 Estes Ave., Chicago, Ill.

BISPALA, John T. (*A* 1940) Partner, Mgr., Bispala Brothers, 2328 First Ave., Hibbing, Minn.

BJERKEN, Maurice H. (*M* 1938; *A* 1927) N. W. Repr. (for mail) Hoffman Specialty Co., 533 S. Seventh St., and 4952-17th Ave. S., Minneapolis, Minn.

BLACK, Edgar N., 3rd (*M* 1922) Philadelphia Mgr. (for mail) Fitzgibbons Boiler Co., Inc., Presser Bldg., Philadelphia, and 111 Woodside Rd., Haverford, Montgomery Co., Pa.

BLACK, Fred. C. (*Life Member*; *M* 1919) Pres. (for mail) F. C. Black Co., 622 W. Randolph St., and 4535 N. Ashland Ave., Chicago, Ill.

BLACK, Harry G. (*M* 1917) Prop. (for mail) P. Gormly Co., 155 N. Tenth St., and 927 N. 65th St., Philadelphia, Pa.

BLACK, James M. (*J* 1940; *S* 1939) Sales Engr., Avery Engineering Co., 1029 Chamber of Commerce, and (for mail) L. B. Harrison Club, Cincinnati, O.

BLACKBURN, E. C., Jr. (*M* 1929) Consulting Engr., 5 Kenwood Rd., Garden City, L. I., N. Y.

BLACKHALL, Wilmot R. (*M* 1922) Partner (for mail) McKellar & Blackhall, 1104 Bay St., and 332 Waverley Rd., Toronto, Ont., Canada.

BLACKMAN, Alfred O. (*M* 1911) Mech. Engr. (for mail) Robert & Co., and 2223 Dellwood Ave., Jacksonville, Fla.

BLACKMORE, F. H. (*M* 1923) Mgr. Mfg. Dept. (for mail) U. S. Radiator Corp., 1056 Natl. Bank Bldg., Detroit, and 515 Tooting Lane, Birmingham, Mich.

BLACKMORE, George C. (*Charter Member*; *Life Member*) Pres., Automatic Gas Equipment Co., 301 Brushton Ave., Pittsburgh, Pa.

BLACKMORE, J. J.* (*Charter Member*; *Life Member*) Retired, 32 West 40th St., New York, N. Y.

BLACKMORE, Joseph J. (*A* 1939; *J* 1937) Sales Engr., Mfrs. Agency, 6327 Clayton Ave., St. Louis, and (for mail) 312 S. Fillmore, Edwardsville, Ill.

BLACKSHAW, J. L.* (*M* 1937; *J* 1929) Br. Mgr., Air and Refrigeration Corp., 268 McDonough Blvd., Atlanta, and (for mail) 247 W. Mercer Ave., College Park, Ga.

BLAIR, Donald W. (*A* 1940) Steam Service Engr., Boston Edison Co., 39 Boylston St., Boston, and (for mail) 32 Summer St., Saugus, Mass.

BLAKELEY, Hugh J. (*M* 1935) Consulting Engr. (for mail) Hubbard Rickerd & Blakeley, 110 State St., Boston, and 145 Greaton Rd., West Roxbury, Mass.

BLAKER, Alfred H. (*A* 1939) Secy. & Treas. (for mail) National Korectaire Co., 1619 Cortland St., and 6018 N. Francisco Ave., Chicago, Ill.

BLANDING, Robert L. (*M* 1938) Vice-Pres. (for mail) Taco Heaters, Inc., 123 South St., and 1385 Smith St., Providence, R. I.

BLANKIN, Merrill F. (*M* 1927; *A* 1926; *J* 1919) (Treas., 1939-40; Council, 1939-40) Pres. (for mail) Haynes Selling Co., Inc., S. E. Cor. Ridge Ave. and Spring Garden St., and 528 E. Gates St., Roxboro, Philadelphia, Pa.

BLAS, Romualdo J. (*M* 1936) Mgr. & Chief Engr., Blas & Co., Apartado Postal 1006, Caracas, Venezuela, South America.

BLAYNEY, W. Ronald (*A* 1939) Secy. & Treas., W. B. Graves Heating Co., 162 N. Desplaines St., and (for mail) 4327 Monticello Ave., Chicago, Ill.

BLAZER, Benjamin V. (*A* 1940) Owner (for mail) M. Blazer & Son, 173 Market St., Passaic, and 48-13th Ave., Paterson, N. J.

BLIZZARD, Bruce C. (*A* 1939) Asst. Mgr. Fuel Oil Dept. (for mail) Imperial Oil, Ltd., 56 Church St., and 1 Mallory Gardens, Toronto, Ont., Canada.

BLOOM, Louis (*M* 1935) Partner, Freeport Plumbing and Heating Engineers, 84-A Broadway, Freeport, L. I., N. Y.

BLUM, Herman, Jr. (*J* 1936) 3050 Santa Fe, Corpus Christi, Tex.

BLUM, Richard J., Jr. (*A* 1940) Sales Engr. (for mail) The Kirk & Blum Mfg. Co., 2850 Spring Grove Ave., and 3909 Vine St., Cincinnati, O.

BLUMENTHAL, M. I. (*M* 1936) Engr. & Instructor, Refrig. & Air Cond. Dept. (for mail) National Schools, 4000 S. Figueroa, and 651 West 40th Pl., Los Angeles, Calif.

BOALES, William G. (*M* 1936; *A* 1923) Owner (for mail) Wm. G. Boales & Associates, 6439 Hamilton Ave., Detroit, and 263 McMillan Rd., Grosse Pointe Farms, Mich.

BODEN, Walter F. (*A* 1937) Br. Mgr. (for mail) Modine Mfg. Co., 424 E. Wells St., Milwaukee, and 606 Milwaukee Ave., South Milwaukee, Wis.

BODINGER, Jacob H. (*M* 1931) Pres. (for mail) J. H. Bodinger Co., Inc., 530 Tenth Ave., New York, and 1902 Avenue L, Brooklyn, N. Y.

BODMER, Emmanuel (see Special Service Roll, p. 73).

BOESTER, Carl F.* (*M* 1939; *A* 1936) Dir. Housing Research, Purdue Research Foundation, Purdue University, Lafayette, Ind., and (for mail) 101 E. Essex, Kirkwood, St. Louis, Mo.

BOGATY, Hermann S. (*M* 1921) 735 E. Phil-Ellena St., Philadelphia, Pa.

BOLAND, Roy O. (*A* 1938) Mgr., Insulation Div. (for mail) Alexander Murray & Co., Ltd., 4035 Richelieu St., Montreal, and 348 Kensington Ave., Westmount, Que., Canada.

BOLTON, Reginald P.* (*Honorary Member; Life Member*; *M* 1897) (*Presidential Member*) (Pres. 1911; 1st Vice-Pres., 1905, 1910; 2nd Vice-Pres. 1903; Board of Governors, 1901, 1905, 1910-13) The R. P. Bolton Co., 116 East 19th St., New York, N. Y.

BOND, Harry H. (*M* 1938) Partner (for mail) Edward E. Ashley, Cons. Engr., 10 East 40th St., New York, and 141-49 181st St., Springfield, L. I., N. Y.

BOND, Horace A. (*M* 1930) Prof. Engr., 152 Washington Ave., and (for mail) 12 Ramsey Pl., Albany, N. Y.

BONTHRON, Robert C. (*A* 1935) Syndicate Repr., Air Cond. Dept., Westinghouse Electric & Manufacturing Co., 150 Broadway, New York, and (for mail) 44 Ingraham Blvd., Hempstead, L. I., N. Y.

BOOT, Arthur (*M* 1938) Mgr., Refrig. & Air Cond. Div. (for mail) Boot & Co., Inc., 115 W. Fulton St., and 928 Orchard Ave., Grand Rapids, Mich.

BOOTH, Charles A. (*M* 1917) Vice-Pres., Buffalo Forge Co., 490 Broadway, and (for mail) 142 Summit Ave., Buffalo, N. Y.

BORAK, Eugene (*M* 1937) Engr., Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, N. Y., and (for mail) 261 Manhattan Ave., Jersey City, N. J.

BORG, Elmer H. (*M* 1938) Partner (for mail) Proudfoot Rawson-Brooks & Borg, Archts., 815 Hubbell Bldg., and 3101 Easton Blvd., Des Moines, Ia.

BORKAT, Philip (*J* 1936) Asst. Engr., Specifications Section, Bureau of Yards & Docks, U. S. Navy Dept., Washington, D. C., and (for mail) 905 Carroll Ave., Takoma Park, Md.

BORLING, John R. (*A* 1934) Engr.-Custodian (for mail) Chicago Board of Education, 214 N. Lavergne Ave., and 1000 N. Waller Ave., Chicago, Ill.

BORNEMANN, Walter A. (*M* 1924; *J* 1923) Sales Engr. (for mail) Carrier Corp., 12 South 12th St., Philadelphia, and 123 W. Wharton Rd., Glenside, Pa.

BORNSTEIN, William (*A* 1937) Partner (for mail) Wm. Bornstein & Son, 720 New Jersey Ave. N. W., Washington, D. C., and 222 Chestnut Ave., Takoma Park, Md.

BORTON, A. Robert (*J* 1939) Dist. Mgr. (for mail) John J. Nesbitt, Inc., 720 Empire Bldg., Pittsburgh, and 316 Breading Ave., Ben Avon, Pittsburgh 2, Pa.

BOTELHO, Nanto Junqueira (*A* 1937) Chief Engr. and Mgr., Ceibrasil Representacoes Ltda., Rua General Camara, 64-7° andar, Rio de Janeiro, Brazil, South America.

BOTTUM, Edward W. (*J* 1938) Chief Engr. (for mail) J. L. Skuttle Co., 1015 Franklin St., and 1601 Clark St., Detroit, Mich.

BOUEY, Angus J. (*A* 1937; *J* 1930) Sales (for mail) The B. F. Sturtevant Co., 681 Market St., and 4810 Fulton St., San Francisco, Calif.

BOUILLON, Lincoln (*M* 1933) Consulting Engr. (for mail) 426-1411 Fourth Ave. Bldg., and 2211-32nd Ave. S., Seattle, Wash.

BOWEN, John C. (*A* 1938) Sales Mgr., Midwest Engineering Co., 201-203 State St., La Crosse, Wis.

BOWERMAN, Everett L. (*A* 1937) Mgr. Air Cond. Dept., Bennett & Wright, Ltd., 72 Queen St. E., and (for mail) 274 Belsize Dr., Toronto, Ont., Canada.

BOWERS, Arthur F. (*A* 1919) Pres. (for mail) Industrial Heating & Engineering Co., 828 N. Broadway, and 2853 N. Hackett Ave., Milwaukee, Wis.

BOWES, Warren H. (*A* 1940) Air Cond. Engr. (for mail) Canadian General Electric Co., 214 King St., W. and 85 Pears Ave., Toronto, Ont., Canada.

BOWLES, Edmund N. (*A* 1937) Northwest Air Cond. Supvr. (for mail) Westinghouse Electric & Manufacturing Co., 20 N. Wacker Dr., and 6043 N. Paulina St., Chicago, Ill.

BOWLES, Potter (*A* 1928) Pres. (for mail) Hoffman Specialty Co., Inc., 77 Bedford St., Stamford, and Box 61, New Canaan, Conn.

BOXALL, Frederick (*M* 1937) Export-Air Cond. & Refrig. Engr., Worthington Pump & Machinery Corp., Worthington Ave., Harrison, and (for mail), 36 Kenwood Ave., Verona, N. J.

BOYAR, Sidney L. (*J* 1938) Asst. Buyer-Stokers & Controls, Sears Roebuck & Co., 925 S. Homan Ave., Chicago, Ill., and (for mail), 711 W. Chicago Ave., E. Chicago, Ind.

BOYD, Spencer W. (*M* 1937; *J* 1931) Consulting Engr. (for mail) Newcomb & Boyd, Cons. Engrs., Trust Co. of Georgia Bldg., and 1505 Fairview Rd., Atlanta, Ga.

BOYD, T. Dudley (*M* 1937) Sales Engr. (for mail) Johnson Service Co., 1113 Race St., and 3332 N. Sterling Way, Cincinnati, O.

BOYDEN, Davis S.* (*Life Member*; *M* 1909) (*Presidential Member*) (Pres. 1937; 1st Vice-Pres. 1936; Treas., 1933-34; Council, 1917, 1930-38) Consultant, Box 386, Shirley, Mass.

BOYKER, Robert Owen (*J* 1935) Contractor, Mac Boyker & Son (for mail) 220 First Ave., and 100 Kennebeck Ave., Kent, Wash.

BOYLE, John R. (*M* 1936) Traveling Sales Mgr., Westerlin & Campbell Co., 1113 Cornelia Ave., and (for mail) 6858 Osceola Ave., Chicago, Ill.

BOZEMAN, Richard (*M* 1936; *J* 1929) Production Supt., United Clay Products Co., 931 Investment Bldg., Washington, D. C., and (for mail) 1706 N. Uhle St., Arlington, Va.

BRAATZ, Chester J.* (*M* 1930) Sales Mgr., Temperature Control Div., Barber-Colman Co., and (for mail) 1819 Clinton St., Rockford, Ill.

BRACKEN, John H. (*M* 1927) Mgr. Industrial Uses Dept. (for mail) The Celotex Corp., 919 N. Michigan Ave., and 455 Oakdale Ave., Chicago, Ill.

BRADFIELD, William W. (*M* 1926) Mech. Engr. (for mail) 341 Michigan Trust Bldg., and 1352 Franklin St. S. E., Grand Rapids, Mich.

BRADFORD, Gilmore G. (*M* 1936) Mgr., Frigidaire Div., General Motors China Ltd., 201 Rte. Cardinal Mercier, Shanghai, China.

BRADLEY, Eugene P. (*M* 1906) Pres. (for mail) Hester-Bradley Co., 2835 Washington Blvd., and No. 4 Yale Ave., University City, St. Louis Co., Mo.

BRANDT, Allen D. (*M* 1940) P. A. Sanitary Engr. (R), U. S. Public Health Service, National Institute of Health, and (for mail) 137 N. Chelsea Lane, Bethesda, Md.

BRANDT, Ernst H., Jr. (*M* 1928) Pres. (for mail) Reliance Engineering Co., Inc., P. O. Box 1292, and 1101 Providence Rd., Charlotte, N. C.

BRANIFF, Paul R. (*A* 1939) Secy.-Treas. (for mail) Braniff Engineering Co., 817 N. Broadway, and 2004 Northwest 16th, Oklahoma City, Okla.

BRATT, Hero D. (*M* 1937) Sales Engr., Warren Webster & Co., 228 Ottawa Ave. S. W., and (for mail) 2259 Stafford Ave. S. W., Grand Rapids, Mich.

BRAUER, Roy (*M* 1926) Dist. Mgr. (for mail) The Trane Co., Pittsburgh, Pa.

BRAUN, Charles R., Jr. (*S* 1939) Student (for mail) Carnegie Institute of Technology, 4903 Forbes St., Pittsburgh, Pa., and 845 Thomas Rd., Columbus, O.

BRAUN, John J. (*M* 1932) Factory Mgr., The United States Playing Card Co., Cincinnati, and (for mail) 4305 Floral Ave., Norwood, O.

BRAUN, Louis T. (*M* 1921) Executive Secy. (for mail) Chicago Master Steamfitters Assn., 228 N. LaSalle St., and 1548 Pratt Blvd., Chicago, Ill.

BRAYMAN, Albert I. (*J* 1937) Draftsman and Estimator, Edward Brayman, Htg. Contractor, 81 Chamber St., Boston, and (for mail) 340 Boulevard, Revere, Mass.

BREDESEN, Bernhard P. (*A* 1931) Engr. (for mail) Reese & Bredesen, 403 Essex Bldg., and 3623 Knox Ave. N., Minneapolis, Minn.

BRENEMAN, Robert B. (*A* 1931; *J* 1927) Br. Mgr. (for mail) Armstrong Cork Co., 37 N. Third St., Columbus, and Rte. 2, Westerville, O.

BREX, Irving E. (*A* 1939) Asst. Mgr., Brex & Bieler Div., Excelsior Steel Furnace Co., 45th St. & First Ave., and (for mail) 7200 Ridge Blvd., Brooklyn, N. Y.

BREYER, Frederick (*S* 1940) Student (for mail) Carnegie Institute of Technology, 4921 Forbes St., Pittsburgh, Pa., and 4219 Richton, Detroit, Mich.

BRIDE, William T. (*M* 1928; *J* 1925) (for mail) Bride, Grimes & Co., P. O. Box 777, Lawrence, and 28 Albion St., Methuen, Mass.

BRIGHAM, Clare M. (*M* 1935) Vice-Pres. in Charge of Sales (for mail) C. A. Dunham Co., 450 E. Ohio St., Chicago, and 420 Maple Ave., Winnetka, Ill.

BRINKER, Harry A. (*M* 1934) 2521 University Ave., Kalamazoo, Mich.

BRINTON, Joseph W. (*M* 1920) Dist. Mgr. (for mail) American Blower Corp., 1003 Statler Bldg., Boston, and 42 Gleason St., West Medford, Mass.

BRISSENDEN, Carrol W. (*J* 1939) Htg. Engr. (for mail) Portland General Electric Co., 621 S. W. Alder St., and 2735 S. E. 61st Ave., Portland, Ore.

BRISSETTE, Leo A. (*M* 1930) Treas. (for mail) Trask Heating Co., 4 Merrimac St., Boston, and 168 Florence St., Melrose, Mass.

BRITTAIN, Alfred, Jr. (*M* 1938) Engr., Weathermakers (Canada) Ltd., 593 Adelaide St., and (for mail) 138 Wheeler Ave., Toronto, Ont., Canada.

BROCHA, John F. (*M* 1936) Buyer of Plbg. and Htg., Montgomery Ward & Co., 619 W. Chicago Ave., and (for mail) 5475 Hirsch St., Chicago, Ill.

BROCKINTON, C. E. (*A* 1937) Sales Engr. (for mail) Advanced Refrigeration, Inc., 350 Peachtree St., and 756 Elkmont Dr. N. E., Atlanta, Ga.

BRODERICK, Edwin L.* (*M* 1933) Research Asst. in Mech. Engrg. (for mail) University of Illinois, 213 Mech. Engrg. Lab., Urbana, and 909 S. First St., Champaign, Ill.

BRODNAX, George H., Jr. (*M* 1938) Sales Engr. (for mail) Georgia Power Co., Electric Bldg., and 1564 Westwood Ave. S. W., Atlanta, Ga.

BROKAW, George K. (*J* 1939; *S* 1938) Office Engr., Clyde E. Bentley, Cons. Engr., 216 Pine St., San Francisco, and (for mail) 2400 Haste St., Berkeley, Calif.

BRONSON, Carlos E.* (*M* 1919) Chief Mech. Engr., Kewanee Boiler Corp., Kewanee, Ill.

BROOKE, Irving E. (*M* 1938) Consulting Engr. (for mail) 189 W. Madison St., Chicago, and 830 Keystone Ave., River Forest, Ill.

BROOM, Benjamin A. (*M* 1914) Sales Promotion Engr., Weil-McLain Co., 641 W. Lake St., and (for mail) 1534 Fargo Ave., Chicago, Ill.

BROOME, Joseph H. (*A* 1936) Sales Engr., Minneapolis-Honeywell Regulator Co., 604 Central Ave., East Orange, and (for mail) 134 Cooper Ave., Montclair, N. J.

BROWN, Alfred P. (*M* 1927) Vice-Pres. (for mail) Reynolds Corp., 1400 Wabansia Ave., Chicago, and 439 Maple St., Winnetka, Ill.

BROWN, Aubrey I.* (*M* 1923) Prof. of Htg. and Vtg. (for mail) Ohio State University, and 169 Richards Rd., Columbus, O.

BROWN, David (*M* 1936) Owner (for mail) 67 Cooper Square, and 54 West 174th St., New York, N. Y.

BROWN, Foskett* (*M* 1926) Pres. (for mail) Gray & Dudley Co., 222 Third Ave. N., and 2314 West End Ave., Nashville, Tenn.

BROWN, H. Junius (*J* 1940) Engr., 64 Norwood Ave., Clifton, S. I., N. Y.

BROWN, John S. (*J* 1937) Test Engr. (for mail) Frigidaire Div., General Motors Sales Corp., Taylor St. Plant, and 35 E. Norman Ave., Dayton, O.

BROWN, Leland S., Jr. (*S* 1940) Student, Catholic University, and (for mail) 15 Bryant St. N. W., Washington, D. C.

BROWN, Mack David (*M* 1938; *J* 1936) Engr., (for mail) Northup & O'Brien, Archts., 602-03 Reynolds Bldg., and 2914 Bon Air Ave., Winston-Salem, N. C.

BROWN, Marvin L. (*M* 1939) Vice-Pres., Dallas Air Conditioning Co., Inc., 3500 Commerce St., and (for mail) 3461 Potomac St., Dallas, Tex.

BROWN, Maurice W. (*J* 1938) Sales Engr. (for mail) American Blower Corp., 619 Mercantile Bldg., and 2523 W. Tenth St., Dallas, Tex.

BROWN, Sterling D. (*J* 1939) Br. Mgr. (for mail) The Trane Co., West Sprague Ave., Spokane, Wash.

BROWN, Tom (*M* 1930) Mgr., Koolshade Div. (for mail) Avery Engineering Co., 1906 Euclid Ave., Cleveland, and 3796 Lowell Rd., Cleveland Heights, O.

BROWN, William H. (*A* 1923) Mgr., Brown Bros., Inc., 3015 North 22nd St., Milwaukee, Wis.

BROWN, W. Maynard (*A* 1930) Warren Webster & Co., 17th and Federal Sts., Camden, N. J.

BROWNE, Alfred L. (*M* 1923) Illinois Engineering Co., 253 Highland Rd., South Orange, N. J.

BRUNDAGE, F. Ward (*J* 1940) Mech. Engr., Stewart-Kingscott Co., 208 Elm St., and (for mail) 809 W. Lovell St., Kalamazoo, Mich.

BRUNETT, Adrian L. (*M* 1923) Mech. Engr., U. S. Supervising Architects Office, Procurement Bldg., Washington, D. C., and (for mail) P. O. Box 36, Rockville, Md.

BRUNNER, Emanuel G. (*A* 1940) Burner Sales, Dome Oil Co., Inc., and (for mail) 707-20th St. N. W., Washington, D. C.

BRUST, Otto (*M* 1930) Consulting Engr., Wilmersdorferstrasse 95, Berlin-Charlottenburg 4, Germany.

BRYANT, Alice Gertrude*, A.B., M.D., F.A.C.S., AS-E.E. (*Life Member*; *M* 1921) Otolaryngologist Physician and Surgeon, 405 Marlborough St., Boston, Mass.

BRYANT, Percy J. (*M* 1915) Chief Engr. (for mail) Prudential Insurance Co., 763 Broad St., Newark, and 754 Belvidere Ave., Westfield, N. J.

BUCK, David T. (*M* 1940; *A* 1936) Pres. (for mail) Buck Engineering Co., 37-41 Marcy St., and 116 W. Main St., Freehold, N. J.

BUCK, Lucien (*M* 1928) Engr., Proctor & Schwartz, Inc., Seventh St. & Tabor Rd., Philadelphia, and (for mail) Pardee Lane, Wyncote, Pa.

BUCKERIDGE, Victor L. (*A* 1938) Partner (for mail) H. Buckeridge & Son, 15108 Kercheval Ave., Grosse Pointe, and 1307 Brys Dr., Grosse Pointe Woods, Mich.

BUCKLEY, Duane J. (*S* 1939) Student, University of Illinois, Urbana, Ill., and (for mail) 421 S. Fountain, Wichita, Kan.

BUCKLEY, Malcomb L. (*A* 1939) Estimator (for mail) Phillips-Getschow Co., 32 W. Hubbard St., and 4542 Beacon St., Chicago, Ill.

BUENGER, Albert* (*M* 1920; *J* 1917) (Council, 1934-37) Mech. Engr. (for mail) A. M. Kinney, Inc., Cons. Engrs., 1301 Enquirer Bldg., and 3171 Portsmouth Ave., Cincinnati, O.

BUENSOD, Alfred C. (*M* 1918) Pres., Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., and (for mail) 33 Fifth Ave., New York, N. Y.

BULLER, Charles R. (*A* 1938) Chief Engr., Oil Burner Div., The Heil Co., 29th & Montana Ave., and (for mail) 2650 S. Shore Drive, Milwaukee, Wis.

BULLOCK, Howard H. (*A* 1933) Commercial Engr. (for mail) General Electric Co., 212 No. Vignes St., Los Angeles, and 2442 Cudahy St., Huntington Park, Calif.

BURCH, Laurence A. (*M* 1934) Sales Mgr., R. L. Deppmann Co., 5853 Hamilton Ave., Detroit, and (for mail) 78 Amherst Rd., Pleasant Ridge, Mich.

BURGES, Joseph H. M. (*J* 1939) 20 Orchard St., Bloomfield, N. J.

BURKE, James J. (*M* 1939; *A* 1937; *J* 1930) Engr. in charge of Air Cond. & Refrig., American Viscose Corp., Delaware Trust Bldg., Wilmington, Del.

BURKHART, Elder M. (*A* 1940; *J* 1935) Sales Engr., Overly Manufacturing Co., Greensburg, Pa., and (for mail) 642 D St. N. E., Washington, D. C.

BURNAM, C. M., Jr. (*M* 1938; *A* 1937) Engrg. Editor (for mail) Heating, Piping and Air Conditioning, 6 N. Michigan Ave., and 10565 S. Hale Ave., Chicago, Ill.

BURNETT, Earle S. (*M* 1920) Senior Mech. Engr., Petroleum and Natural Gas Div., U. S. Bureau of Mines, Amarillo Helium Plant, P. O. Box 2250, and (for mail) 4223 West 11th Ave., Amarillo, Tex.

BURNS, Edward J. (*M* 1923) Harris Bros. Plumbing Co., 217 W. Lake St., and (for mail) 4716 Aldrich Ave. S., Minneapolis, Minn.

BURNS, Frank G. (*M* 1940) Dealer Coordinator (for mail) New Orleans Public Service, Inc., 317 Baronne St., and 8515 Pritchard Pl., New Orleans, La.

BURNS, Harold J. (*A* 1941; *J* 1939) Asst. Htg. Engr., Washington Gas Light Co., 411 Tenth Street, Washington, D. C., and (for mail) 105 Allen Rd., Yorktowne Village, Md.

BURNS, John R. (*J* 1936; *S* 1933) Htg. Dept., Crane Co., 279 Madison Ave., New York, N. Y., and (for mail) 504 N. Main St., Wallingford, Conn.

BURR, Griffith C. (*M* 1937) Air Cond. Engr., Air Conditioning Corp., 306 N. Elm St., and (for mail) 808 Cypress St., Greensboro, N. C.

BURRITT, Charles G. (*A* 1916) Office Mgr. (for mail) Johnson Service Co., 922 Second Ave. S. and Leamington Hotel, Minneapolis, Minn.

BURTON, W. Russell (*A* 1939) Sales Engr., H. J. Sandberg, 500 N. E. Union Ave., and (for mail) 2816 N. E. 19th St., Portland, Ore.

BUSHNELL, Carl D. (*A* 1921) Pres. (for mail) Bushnell Machinery Co., 311 Ross St., Pittsburgh, and 94 Pilgrim Rd., Rosslyn Farms, Carnegie, Pa.

BUSSE, Herbert (*M* 1938) Chief Engr., Fisher Bldg. Div., Fisher & Co., 417 Fisher Bldg., and (for mail) 16760 Greenview Rd., Detroit, Mich.

BUTLER, Peter D. (*M* 1922) Sales, U. S. Radiator Corp., Detroit, Mich, and (for mail) 127 Edgewater Rd., Cliffside Park, N. J.

BUTT, Roderick E. W. (see Special Service Roll, p. 73).

BUZZARD, Francis H. (*M* 1939) Asst., Charles S. Leopold, Cons. Engr., 213 S. Broad St., Philadelphia, Pa., and (for mail) 624 Wood Lane, Haddonfield, N. J.

BYRD, Tom I. (*A* 1936) Mgr., Bldgs. Market Dept. (for mail) The American Rolling Mill Co., 703 Curtis St., and 2403 Fleming Rd., Middletown, O.

BYRNE, Joseph J. (*A* 1939) Htg. Engr. (for mail) Mueller Furnace Sales Co., 4069 N. E. Union Ave., and 6414 N. E. Rodney Ave., Portland, Ore.

BYSOM, Leslie L. (*M* 1915) Mech. Engr., Design Section, Puget Sound Navy Yard, Public Wks. Dept., and (for mail) 1214 Eighth St., Bremerton, Wash.

C

CADY, Edward F. (*J* 1937) Mech. Engr., The Austin Co., 16112 Euclid Ave., Cleveland, and (for mail) 2196 Rexwood Rd., Cleveland Heights, O.

CAIN, William J. (*S* 1940) 9111 Delphine Ave., Overland, Mo.

CALDWELL, Arthur C. (*M* 1930) Engr. and Estimator, P. Gormly Co., 155 N. Tenth St., and (for mail) 550 South 48th St., Philadelphia, Pa.

CALEB, David (*M* 1923) Engr. (for mail) Kansas City Power & Light Co., 1330 Baltimore Ave., and 141 Spruce St., Kansas City, Mo.

CALL, Joseph (*M* 1938; *J* 1936) Mgr., Air Cond. Div., Elliott-Lewis Co., 2518 N. Broad St., Philadelphia, and (for mail) 50 Fairfield Rd., Brookline Park, Del. Co., Pa.

CALLAHAN, Peter J. (*M* 1934) Inspecting Engr., Central Hanover Bank & Trust Co., 60 Broadway, New York, and (for mail) 4057 Amboy Rd., Great Kills, S. I., N. Y.

CALNAN, Edward J. (*M* 1941) Power Engr. (for mail) Ontario Paper Co., Ltd., Thorold, and 208 Russell Ave., St. Catharines, Ont., Canada.

CALVER, Robert W. (see Special Service Roll, p. 73).

CAMERON, Robert T. (*J* 1941; *S* 1938) Sales Engr., Crane Co., 1007 W. Bay St., and (for mail) 1516 Lorimier Rd., Jacksonville, Fla.

CAMPAU, William R. (*M* 1940) Secy. & Gen. Mgr. (for mail) Kendall Heating Co., 64 S. W. Front Ave., and 4418 N. E. 11th, Portland, Ore.

CAMPBELL, Alfred Q., Jr. (*A* 1940; *J* 1933) Engr. (for mail) E. K. Campbell Heating Co., P. O. Box 365, and Robin Rd., Nashville, Tenn.

CAMPBELL, Andy O. (*A* 1941; *J* 1939) Engr. & Draftsman (for mail) Oklahoma Gas & Electric Co., Third & Harvey Sts., and 2749 N. W. 21st St., Oklahoma City, Okla.

CAMPBELL, Bowen (*M* 1938) Chief Engr. (for mail) Campbell Htg. Co., 3127 Dean Ave., and 2404 East 29th St., Des Moines, Ia.

CAMPBELL, Everett K.* (*M* 1920) (Council, 1931-33; 1939-40) Pres. (for mail) E. K. Campbell Heating Co., 2445 Charlotte St., and 3717 Harrison, Kansas City, Mo.

CAMPBELL, E. K., Jr. (*M* 1938; *J* 1930) Secy., E. K. Campbell Heating Co., 2445 Charlotte St., Kansas City, and (for mail) 5351 F Gladstone, Normandy, Mo.

CAMPBELL, George Summers (*A* 1941; *J* 1937) Consulting Engr., Geo. S. Campbell, Mech. Engr., 1100-17th Ave. S., Nashville, Tenn.

CAMPBELL, George W. (*J* 1939) Engr., T. H. Urdahl, Cons. Engr., 726 Jackson Pl. N. W., and (for mail) 1311-30th St. N. W., Washington, D. C.

CAMPBELL, Robert E. (*A* 1940; *J* 1934) Chief Engr., General Cooling and Heating Corp., 120 E. Forsyth St., and (for mail) 1521 Catherines Ct., Jacksonville, Fla.

CAMPBELL, Roger P. (*J* 1939) Engr. (for mail) E. K. Campbell Heating Co., 2445 Charlotte St., and 3717 Harrison Blvd., Kansas City, Mo.

CAMPBELL, Thomas F. (*M* 1928) (for mail) T. F. Campbell Co., 1013 Penn Ave., and R. D. No. 1, Wilkinsburg, Pa.

CANDEE, Bertram C. (*M* 1933) Partner, Beman & Candee, 374 Delaware Ave., Buffalo, and (for mail) 19 Tremont Ave., Kenmore, N. Y.

CAPLE, Ira (*J* 1941; *S* 1938) Engr., Super Radiator Corp., and (for mail) 715 University Ave. S. E., Minneapolis, Minn.

CARBONE, James H. (*M* 1937) Htg.-Vtg. Inspector, City of New York, New York, and (for mail) 121-13 198th St., St. Albans, L. I., N. Y.

CAREY, Paul C. (*M* 1930) Consulting Engr. and Member of Firm (for mail) Runyon & Carey, 33 Fulton St., Newark, and 31 Claremont Dr., Maplewood, N. J.

CARLE, William E. (*M* 1926) Pres. (for mail) Carle-Boehling Co., Inc., 1641 W. Broad St., and 4015 W. Franklin St., Richmond, Va.

CARLOCK, Marion F. (*M* 1936) Dist. Mgr., American Foundry & Furnace Co., 7008 Amherst, University City, Mo.

CARLSON, C. O. (*A* 1937) Owner (for mail) C. O. Carlson Heating Co., 1627 Washington Ave. N., and 1806 Thomas Ave. N., Minneapolis, Minn.

CARLSON, Everett E. (*M* 1932; *A* 1929) Br. Mgr. (for mail) The Powers Regulator Co., 1010 Louderman Bldg., and 6652 Washington Ave., St. Louis, Mo.

CARNAHAN, John H. (*A* 1940; *J* 1937) Sales Dept., Oklahoma Gas & Electric Co., 321 N. Harvey St., and (for mail) 3116 Northwest 26th St., Oklahoma City, Okla.

CARNEY, Edward J. (*A* 1939) Partner (for mail) John C. Kohler Co., 554 North 16th St., and 1020 North 64th St., Philadelphia, Pa.

CARON, Hector (*A* 1938) Mgr. & Owner, Hector Caron, 324 Lincoln Highway, and (for mail) 421 S. Third St., Rochelle, Ill.

CARPENTER, Randolph H. (*M* 1921) (Council, 1930-35) Mgr. New York Office (for mail) Nash Engineering Co., Graybar Bldg., 420 Lexington Ave., New York, and 20 Jefferson Ave., White Plains, N. Y.

CARR, Maurice L.* (*M* 1931) Dir., Pittsburgh Testing Lab., Stevenson & Locust Sts., Pittsburgh, Pa.

CARRIER, Earl G. (*M* 1936; *J* 1929) Br. Mgr., Carrier Corp., 704 Statler Bldg., Boston, and (for mail) 68 High St., Winchester, Mass.

CARRIER, Willis Haviland* (*M* 1913) (*Presidential Member*) (Pres., 1931; 1st Vice-Pres., 1930; 2nd Vice-Pres, 1929; Council, 1923-32) Chairman of the Board (for mail) Carrier Corp., 302 S. Geddes St., and 2570 Valley Drive, Syracuse, N. Y.

CARROLL, Edgar E. (*A* 1939) Owner (for mail) Kleenair Furnace Co., 5329 N. E. Sandy Blvd., and 2434 N. E. 43rd Ave., Portland, Ore.

CARROLL, William M. (*J* 1938) Sales Engr. (for mail) Tom Dolan Htg. Co., 614 W. Grand, and 908 East Dr., Oklahoma City, Okla.

CARTER, Alexander W. (*M* 1940; *J* 1936) Htg. Engr. (for mail) Monarch Brass Manufacturing Co., Ltd., 71 Browns Ave., and 117 Elmer Ave., Toronto, Ont., Canada.

CARTER, Doctor (*M* 1934) Consulting Engr., 162 Avenue Parade, Accrington, Lancs., England.

CARTER, John H.* (*M* 1936) Mgr. Refrig. Dept. (for mail) Kupferle-Hicks Heating Co., 3974 Delmar Blvd., St. Louis, and 341 Spring Ave., Webster Groves, Mo.

CARY, Edward B. (*M* 1935) Lt. Comdr., Asst. to Public Works Officer (for mail) U. S. Naval Training Station, Great Lakes, and 412 Douglas Ave., Waukegan, Ill.

CASE, Delbert Vernon (*M* 1937) Mgr. & Owner, Case Engineering Co.—Lipman Refrigeration Sales Co., 1728 Grand Ave., Kansas City, and (for mail) Route No. 1, Hickman Mills, Mo.

CASE, Walter G. (*A* 1930) Mgr., Ideal Boilers & Radiators, Ltd., Ideal House, Great Marlborough St., London, W.1, and (for mail) 66 The Ridgeway, Kenton, Harrow, Middlesex, England.

CASEY, Byron L. (*M* 1921) Sales Engr. (for mail) Ilg Electric Ventilating Co., 222 N. LaSalle St., Chicago, and 404 Vine Ave., Park Ridge, Ill.

CASKEY, Luther H., Jr. (*S* 1938) Design Engr., L. H. Caskey, 513 N. Queen St., Martinsburg, W. Va.

CASPERD, Henry W. H. (*A* 1938; *J* 1930) Engr., Carrier Co., Ltd., 24 Buckingham Gate, London and (for mail) 21 Robin Hood Lane, Sutton, Surrey, England.

CASSELL, John D.* (*Life Member*; *M* 1913) (Council, 1930-35) Retired, 2008 Walnut St., Philadelphia, Pa.

CASSELL, William L. (*M* 1936) Owner (for mail) William L. Cassell, Mech. Engr., 912 Baltimore Ave., Kansas City, and R. F. D. No. 6, Independence, Mo.

CHALMERS, Charles H. (*M* 1925) Gen. Mgr. (for mail) Chalmers Oil Burner Co., 1234 Central Ave., and 523 Seventh St. S. E., Minneapolis, Minn.

CHAMBERS, Fred W. (*M* 1936) Pres. (for mail) F. W. Chambers & Co., Ltd., 96 Bloor St. W., and 55 Glengowan Rd., Toronto, Ont., Canada.

CHAMPLIN, Robert C. (*A* 1938) Mgr. Air Cond. Engrg. Dept. (for mail) Timken Silent Automatic Div., 100-400 Clark Ave., and 13640 Mendota Ave., Detroit, Mich.

CHAPIN, C. Graham (*M* 1933) Treas. (for mail) Hopson & Chapin Co., 231 State St., and 66 Faire Harbour Pl., New London, Conn.

CHAPIN, Harvey G. (*M* 1935) Sales Engr. (for mail) Westerlin & Campbell Co., 1113 Cornelia Ave., and 8352 Maryland Ave., Chicago, Ill.

CHAPMAN, William A., Jr. (*M* 1936) Commercial & Air Cond. Sales Planning (for mail) Frigidaire Div., General Motors Sales Corp., 300 Taylor St., and 525 Daytona Pkwy., Dayton, O.

CHARLES, Paul L. (*M* 1938) Mgr. (for mail) Walsh & Charles, 406 Tribune Bldg., and 145 Ash St., Winnipeg, Man., Canada.

CHARLET, Louis W. (*M* 1934) Gen. Sales Mgr., Kewanee Boiler Corp., and (for mail) 442 S. Tremont, Kewanee, Ill.

CHASE, Arthur M., Jr. (*M* 1938) Sales Engr. (for mail) York Ice Machinery Corp., 2201 Texas Ave., and 3333 Ozark St., Houston, Tex.

CHASE, Chauncey L. (*M* 1931) Partner (for mail) Edward E. Ashley, Cons. Engr., 10 East 40th St., New York, and 8829 Ft. Hamilton Pkwy., Brooklyn, N. Y.

CHASE, L. Richard (*M* 1938; *J* 1931) Vice-Pres. & Gen. Mgr. (for mail) Transport Clearing House, Inc., 111 W. Jackson Blvd., Chicago, and 555 Hinman, Evanston, Ill.

CHASE, Peter S. (*A* 1940) Owner (for mail) Chase Co., 936 Oak St., and 1167 Ferry St., Eugene, Ore.

CHASE, Roger E. (*A* 1939) Pres. (for mail) R. E. Chase & Co., Tacoma Bldg., and 117 N. Tacoma Ave., Tacoma, Wash.

CHEESEMAN, Evans W. (*J* 1937; *S* 1934) 1st Lt. Corps of Engrs. Res., and (for mail) 1503 Willow St., Coffeyville, Kans.

CHENEVERT, J. Georges (*M* 1938) Consulting Engr. (for mail) Arthur Surveyer & Co., Rm. 1203, 1010 St. Catherine St. W., Montreal, and 536 Outremont Ave., Outremont, Que., Canada.

CHENOWETH, Dale M. (*J* 1938; *S* 1936) Asst. Gen. Supt., S. Braintree Factory, Armstrong Cork Co., S. Braintree, and (for mail) 156 River St., Braintree, Mass.

CHERNE, Realto E. (*M* 1938; *J* 1929) Br. Chief Engr., Carrier Corp., and (for mail) 167 Ridgeway Ave., Syracuse, N. Y.

CHERRY, Lester A.* (*M* 1921) (for mail) Cherry, Cushing and Preble, Cons. Engrs., 271 Delaware Ave., Buffalo, and 151 Euclid Ave., Kenmore, N. Y.

CHESTER, Frank L. (*A* 1940) Mgr. (for mail) W. G. Chester & Son, 179 Bannatyne Ave., and 219 Kingston Row, Winnipeg, Man., Canada.

CHESTER, Thomas* (*M* 1917) 230 Fifth Ave., New York, N. Y.

CHEYNEY, Charles C. (*A* 1913) Asst. Sales Mgr. (for mail) Buffalo Forge Co., 490 Broadway, and 255 Lincoln Pkwy., Buffalo, N. Y.

CHILDS, Lewis A. (*M* 1938) Dist. Sales Mgr. (for mail) Clarage Fan Co., 520 Commercial Trust Bldg., and 1320 Foulkrod St., Philadelphia, Pa.

CHRISTENSON, Harry (*A* 1931) Co-partner (for mail) Hunter-Prell Co., 15-19 E. Jackson St., and 121 Sunset Blvd., Battle Creek, Mich.

CHRISTMAN, William F. (*A* 1931) Engr. (for mail) Kroeschell Engineering Co., 215 W. Ontario St., and 5649 Artesian Ave., Chicago, Ill.

CHRISTOPHERSEN, Andrew E. (*M* 1935) Engr.-Custodian (for mail) Board of Education, Spalding School, 1628 Washington Blvd., and 2923 N. Kilpatrick Ave., Chicago, Ill.

CHURCH, Herbert J. (*M* 1922) Mgr. (for mail) Darling Brothers, Ltd., 137 Wellington St. W., Toronto, and 358 Main St. N., Weston, Ont., Canada.

CITRON, Daniel J. (*S* 1938) 2055 Ryer Ave., New York, N. Y.

CLAR, Robert, Jr. (*A* 1938) Sales Engr., U. S. Radiator Corp., 127 Campbell Ave., Detroit, and (for mail) 6898 Livernois, Rochester, Mich.

CLARE, Fulton W. (*M* 1927) 935 Plymouth Rd. N. E., Atlanta, Ga.

CLARK, Adrian N. (*A* 1939) Staff Engr., Good Housekeeping Institute, Eighth Ave. & 57th St., New York, and (for mail) 86 Chase Rd., Manhasset, L. I., N. Y.

CLARK, Albert C. (*A* 1939) Engr., Mueller Furnace Sales Co., 4069 N. E. Union Ave., and (for mail) 3742 N. E. 68th Ave., Portland, Ore.

CLARK, E. Harold (*M* 1922) Mfrs. Agent, 600 Michigan Theatre Bldg., and (for mail) 2539 Lakewood, Detroit, Mich.

CLARK, Lynn W. (*A* 1938) Engr. & Salesman (for mail) Hall-Neal Furnace Co., 1324 N. Capitol Ave., and 737 West 32nd St., Indianapolis, Ind.

CLARK, Robert L. (*A* 1918) Pres. & Treas., The Clark Asbestos Co., 1893 East 55th St., Cleveland, and (for mail) 927 Caledonia Ave., Cleveland Heights, O.

CLARKSON, John R. (*J* 1941; *S* 1938) 2832 Burd, St. Louis, Mo.

CLAUSEN, Arnold H. (*M* 1939) Owner & Mgr. (for mail) Clausen Engineering Co., 307 Wall St., and 404 E. Howell St., Seattle, Wash.

CLAY, Wharton (*M* 1939; *A* 1938) Secy. (for mail) National Mineral Wool Association, 1270 Sixth Ave., New York, and 127 S. Broadway, Nyack, N. Y.

CLEGG, Carl (*M* 1922) Dist. Mgr. (for mail) American Blower Corp., 311 Mutual Bldg., and 3413 Gillham Rd., Kansas City, Mo.

CLEMENS, Joseph D. (*S* 1940) Graduate Student (for mail) Purdue University, 201 Quincy St. W., Lafayette, Ind., and Lincoln Ave., St. Joseph, Mich.

CLIFTON, John Arthur (*A* 1938) Mgr. (for mail) Renown Plumbing Supplies, Ltd., 236 Parliament St., and 369 Belsize Dr., Toronto, Ont., Canada.

CLO, Harry E. (*J* 1939) Sales Engr. (for mail) American Air Filter Co., Inc., Rm. 1310, 228 N. LaSalle St., Chicago, and 630 Library Pl., Evanston, Ill.

CLOSE, Paul D.* (*M* 1928) Tech. Secy. (for mail) Insulation Board Institute, 111 W. Washington St., Chicago, and 757 Maclean Ave., Kenilworth, Ill.

CLOSE, Robert (*M* 1938) Chief Air Cond. Engr., National Broadcasting Co., 30 Rockefeller Plaza, New York, N. Y., and (for mail) 185 Glenwood Ave., Leonia, N. J.

COCHRAN, Lex H. (*M* 1934) Sales Mgr., Western Div. (for mail) American Blower Corp., 625 Market St., and 130 Camino Del Mar, San Francisco, Calif.

COCKINS, William W. (*A* 1941; *J* 1937) Sales Engr. (for mail) The Trane Co., 1129 Folsom St., San Francisco, and 1700 Madera St., Berkeley, Calif.

CODY, Henry C. (*M* 1936) Sales Engr., Pierce Butler Radiator Corp., 15th & Glenwood Ave., and (for mail) 7336 North 21st St., Philadelphia, Pa.

COGHLAN, Sherman F. (*A* 1937) Mech. Engr., Metropolitan Water District of So. Calif., 306 W. Third St., Los Angeles, and (for mail) 414 Ninth St., Santa Monica, Calif.

COHAGEN, Chandler C. (*M* 1919) Archt. (for mail) 211 Hedden Bldg., Box 2100, and 245 Avenue G, Billings, Mont.

COHEN, Philip (*M* 1932) Dist. Mgr. (for mail) B. F. Sturtevant Co., 401 E. Ohio Gas Bldg., and 7100 Euclid Ave., Cleveland, O.

COLBY, John H. (*J* 1939) Sales Engr. (for mail) Johnson Service Co., 20 Winchester St., Boston, and 25 Jefferson Rd., Wellesley Hills, Mass.

COLCLOUGH, Otho T. (*A* 1933) Custodian, American Legation, and (for mail) 726 Parkdale Ave., Ottawa, Ont., Canada.

COLE, C. Boynton (*M* 1940; *J* 1937) Owner, Boynton Cole Contracting Engr., Htg. & Vtg., 1873 Piedmont Rd., and (for mail) 1843 Flagler Ave. N. E., Atlanta, Ga.

COLE, Grant E. (*A* 1925) Vice-Pres. & Mgr. (for mail) Trane Co. of Canada, Ltd., 4 Mowat Ave., and 112 Tyndall Ave., Toronto, Ont., Canada.

COLEMAN, John B. (*M* 1920) Chief Engr. (for mail) Grinnell Co., Inc., 275 W. Exchange St., and 237 Cole Ave., Providence, R. I.

COLFORD, John (*A* 1937) Pres., John Colford, Ltd., 2007 Guy St., Montreal, and (for mail) 51 Upper Bellevue Ave., Westmount, Que., Canada.

COLLE, Samuel S. (*A* 1938) Engr. (for mail) Air Conditioning Engineering Co., 361 Youville Sq., and 4968 Fulton St., Montreal, Que., Canada.

COLLIER, William I. (*M* 1921) Pres. (for mail) W. I. Collier & Co., 522 Park Ave., Baltimore, and Ellicott City, Md.

COLLINS, John F. S., Jr. (*M* 1933) (Council, 1940) Secy.-Treas. (for mail) National District Heating Assn., 1231 Grant Bldg., and 827 N. Euclid Ave., E. E., Pittsburgh, Pa.

COLMAN, Robert C. (*A* 1940) Vice-Pres., McQuay, Inc., 1600 Broadway N. E., Minneapolis, and (for mail) 102 Exeter Pl., St. Paul, Minn.

COLMENARES, Gaspar Vizoso (*A* 1938) (for mail) Castel-Vizo, S. A., Obrapia 407, and Calle 10 entre 5a y 3a, Miramar, Havana, Cuba.

COMO, Jack A. (*M* 1939) Estimating Engr., Independent Plumbing Co., 172 Luckie St. N. W., and (for mail) 2865 Elliot Circle, Atlanta, Ga.

COMSTOCK, Glen M. (*A* 1926) Dist. Repr.-Engr. (for mail) L. J. Wing Manufacturing Co., 604 Chamber of Commerce Bldg., Pittsburgh, and 154 College Ave., Beaver, Pa.

CONATY, Bernard M. (*M* 1935) Sales Mgr. (for mail) American District Steam Co., North Tonawanda, and P. O. Box 342, Eden, N. Y.

CONE, William Edward (*A* 1941; *J* 1937) Air Cond. Engr., Shook & Fletcher Supply Co., 1814 First Ave. N., and (for mail) 1037 Tenth Ave. S., Birmingham, Ala.

CONKLIN, Robert Murray (*J* 1940) Engr. (for mail) The Fred D. Pfening Co., 1075 W. Fifth Ave., and 1445 W. Second Ave., Columbus, O.

CONNELL, Richard F. (*M* 1916) Mgr. Capitol Testing Laboratory (for mail) U. S. Radiator Corp., 1056 National Bank Bldg., and 2970 Burlingame, Detroit, Mich.

CONNER, Raymond M. (*M* 1931) Dir. Testing Labs. (for mail) American Gas Association, 1032 East 62nd St., Cleveland, and 271 East 216th St., Euclid, O.

CONNORS, Edward C. (*A* 1940) Engr. Custodian Chicago Board of Education, 1000 Grand Ave., and (for mail) 4850 Quincy St., Chicago, Ill.

CONRAD, Roy (*M* 1935) Sales Engr., Carrier Corp., 1500 S. Santa Fe, Los Angeles, Calif., and (for mail) 3416 Colfax "B", Denver, Colo.

CONSTANT, Earl S. (*J* 1935) Engr., Buffalo Forge Co., 490 Broadway, and (for mail) 149 Highland Ave., Buffalo, N. Y.

CONVERSE, Thornton John (*M* 1941) Engr. (for mail) Office of Douglas Orr, Archt., 96 Grove St., New Haven, and Flying Point, Stony Creek, Conn.

COOK, Benjamin F. (*M* 1920) Consulting Engr., 114 W. Tenth St. Bldg., Kansas City, and (for mail) 1720 Overton Ave., Independence, Mo.

COOK, Henry Dale (*A* 1938) Sales Engr. (for mail) General Controls Co., 450 E. Ohio St., Chicago, Ill., and 73 East 10th St., Holland, Mich.

COOK, Ralph P. (*M* 1930) Asst. Supt. Engrg. & Maintenance Dept., in charge of Engrg. Div. (for mail) Eastman Kodak Co., Kodak Park, and 663 Seneca Pkwy., Rochester, N. Y.

COOKE, Thomas C. (*A* 1937) Htg. & Air Cond. Engr. (for mail) Tomlinson Co., Inc., 400-402 E. Peabody St., P. O. Box 217, and 1118½ Eighth St., Durham, N. C.

COOLEY, Edgerton C. (*M* 1938) Owner (for mail) E. C. Cooley Co., 625 Market St., San Francisco, and Box 789 B, Route 1, Los Altos, Calif.

COOMBE, James (*A* 1932) Vice-Pres. (for mail) William Powell Co., 2525 Spring Grove Ave., and 2363 Grandin Rd., Cincinnati, O.

COON, Thurlow E. (*M* 1916) Pres. (for mail) The Coon-DeVisser Co., Inc., 2051 W. Lafayette, and 826 Edison Ave., Detroit, Mich.

COOPER, Dale S. (*M* 1938; *A* 1937) Consulting Engr., 216 E. Cowan Dr., Houston, Tex.

COOPER, Donald E. (*J* 1939) Partner, D. E. Cooper & Son, 540 Hood St., Salem, Ore.

COOPER, John W. (*M* 1932; *A* 1925; *J* 1921) Repr. (for mail) Buffalo Forge Co., 1598 Arcade Bldg., St. Louis, and 612 Hawbrook Dr., Kirkwood, Mo.

COOPER, William B. (*J* 1937) Application Engr. Home Htg. Sales, Westinghouse Electric & Manufacturing Co., 653 Page Blvd., Springfield, Mass.

COOPERMAN, Edward (*S* 1940) Student, Carnegie Institute of Technology, and (for mail) 3120 Avalon St., Pittsburgh, Pa.

COPPERUD, Edmund R. (*J* 1933) Asst. Mgr., Minneapolis Plumbing Co., 1420 Nicollet Ave., and (for mail) 17 West 25th St., Minneapolis, Minn.

CORNWALL, Charles C. (*J* 1935) Research Engr., The Bahnson Co., 1001 S. Marshall St., and (for mail) 473 Carolina Circle, Winston-Salem, N. C.

CORNWALL, George I. (*M* 1919) Sales Engr., Burnham Boiler Corp., 701 Spring St., Elizabeth, N. J.

CORRAO, Joseph (*A* 1936; *J* 1933) Engr., Dept. of Works, Engrg. Dept., City Hall, and (for mail) 854-31st Ave., San Francisco, Calif.

CORRIGAN, James A. (*A* 1940; *J* 1935; *S* 1930) Engr. (for mail) Corrigan Co., 2501 St. Louis Ave., and 6130 McPherson Ave., St. Louis, Mo.

COST, George W. (*J* 1939; *S* 1938) Jr. Sales Engr., Cornelius Engineering Corp., 628 Forbes Ave., Wilkinsburg, and (for mail) 5556 Forbes St., Pittsburgh, Pa.

COTT, William B. (*M* 1940) Sales Engr. Air Cond., Doermann-Roehrer Co., 450 E. Pearl St., and (for mail) 3001 Bellewood Ave., Cincinnati, O.

COTTRELL, Walter H. (*A* 1940) Gen. Mgr. (for mail) Home Owners Heating Equipment Co., 2919 Nicollet Ave., Minneapolis, and 3148 Webster Ave., St. Louis Park, Minn.

COVER, E. B. (*M* 1937) Sales Engr., York Ice Machinery Corp., 115 South 11th St., St. Louis, Mo., and (for mail) 3252 Waverly, East St. Louis, Ill.

COVER, Richard R. (*A* 1936) Engr., 2011 N. Utah St., Arlington, Va.

COWARD, Charles W. (*M* 1935) Pres. & Treas. (for mail) Coward Engineering Co., 411 Cooper St., Camden, and 815 Lincoln Ave., Palmyra, N. J.

COX, Harrison F. (*A* 1930) Htg. & Air Cond., 243 Carroll St., Paterson, N. J.

COX, Samuel F. (*M* 1939) Tech. Dir. Double Glazing Div. (for mail) Pittsburgh Plate Glass Co., 2200 Grant Bldg., and 6049 Bunkerhill St., Pittsburgh, Pa.

COX, Thomas M., Jr. (*A* 1941; *J* 1937) Engr., Carrier Corp., and (for mail) 178 Maplewood Ave., Syracuse, N. Y.

COX, Vernon G. (*A* 1939) Dist. Sales Mgr. (for mail) Century Electric Co., 514 Mercantile Bldg., 810 Main St., and 207 Yarmouth St., Dallas, Tex.

COX, William W. (*Life Member*; *M* 1923) Pres. & Mgr. (for mail) Heating Service Co., Inc., 326 Columbia St., and 6232-31st Ave. N. E., Seattle, Wash.

CRAIG, Joseph A. (*J* 1940) Sales, Trane Co., 850 Cromwell Ave., St. Paul, and (for mail) 3736-16th Ave. S., Minneapolis, Minn.

CRANE, Robert S. (*M* 1938) Dist. Engr. (for mail) Air Cond. Frigidaire Div., General Motors Sales Corp., 4 Cummins Station, and 3514 Harding Rd., Nashville, Tenn.

CRARY, James O. (*A* 1940) Mgr. Commercial Air Cond. Dept. (for mail) Frigidaire Div., General Motors Sales Corp., 4436 Toulouse St., and 4430 S. Johnson St., New Orleans, La.

CRAWFORD, Arthur C. (*A* 1938) Commercial Engr., Potomac Electric Power Co., Tenth & E Sts. N. W., and (for mail) 429 Butternut St. N. W., Washington, D. C.

CRAWFORD, John H., Jr. (*A* 1936; *J* 1930) Air Cond. Engr., Hitchen Co., 441 Lexington Ave., New York, N. Y., and (for mail) 289 Reynolds Terr., Orange, N. J.

CRESSY, L. Villere (*M* 1940) Partner (for mail) L. Villere Cressy & Lewis S. Alcus, Cons. Engrs., 916 Union St., and 3217 DeSoto St., New Orleans, La.

CRIQUI, Albert A.* (*M* 1919) Chief Engr. Htg. & Vtg. Dept., Buffalo Forge Co., 490 Broadway, Buffalo, and (for mail) 39 St. Johns Ave., Kenmore, N. Y.

CROLEY, Jack G. (*J* 1940) Htg. Engr. (for mail) Savannah Gas Co., 114 Barnard St., and 215 E. Charlton St., Savannah, Ga.

CROMBIE, James (*A* 1939) Sales Engr., American Radiator & Standard Sanitary Corp., Elyria, and (for mail) 3901 Oak St., Mariemont, Cincinnati, O.

CRONE, Charles E. (*M* 1922) Pres. (for mail) Charles E. Crone Co., 1656 N. Ogden Ave., and 1320 N. State St., Chicago, Ill.

CRONE, Thomas E. (*Life Member*; *M* 1920) Retired, c/o Mrs. Carol Brown, Apt. 6-B, 3705-80th St., Jackson Heights, L. I., N. Y.

CRONEY, P. Alfred (*M* 1938) Chief of Mech. Section, U. S. Housing Authority, Interior Bldg. N., Washington, D. C., and (for mail) 112 Granville Dr., Silver Spring, Md.

CROPPER, Robert O. (*M* 1938) Head Operating Engr., Refrigeration Plant & Htg. Equip., War Dept., c/o Quartermaster, Ft. Knox, and (for mail) Vine Grove, Ky.

CROSBY, Edward L. (*M* 1936) Pres. (for mail) Henry Adams, Inc., 1015-1023 Calvert Bldg., and 5323 Belleville Ave., Baltimore, Md.

CROSS, Freeman G. (*M* 1936) Sales Mgr. Controls Div. (for mail) Fulton Sylphon Co., and 31 Nokomis Circle, Knoxville, Tenn.

CROSS, Robert C.* (*M* 1937) Chief of Htg. Div., Schwitzer-Cummins Co., 1125-1229 Massachusetts Ave., Indianapolis, Ind.

CROSS, Robert E. (*M* 1938; *A* 1931) Dist Mgr., Minneapolis-Honeywell Regulator Co., 271 Columbus Ave., and (for mail) 68 Kimberly Ave., Springfield, Mass.

CROUT, Marvin M. (*M* 1930; *A* 1938) Br. Mgr. (for mail) York Ice Machinery Corp., 412 Houston St., and 2392 Hurst Dr., Atlanta, Ga.

CRUMP, Alvin L. (*M* 1937) Sales Engr. (for mail) Powers Regulator Co., 2720 Greenview Ave., Chicago, and 2701 Payne St., Evanston, Ill.

CUCCI, Victor J. (*M* 1930) Consulting Engr. (for mail) 30 Church St., New York, and 451-55th St., Brooklyn, N. Y.

CULBERT, William P. (*A* 1929) Partner (for mail) Culbert-Whitby Co., 2019 Rittenhouse Sq., Philadelphia, and 929 Alexander Ave., Drexel Hill, Pa.

CULLEN, Augustine G. (*M* 1939; *A* 1936) Owner, Cullen Co., 20 L St. S. W., Washington, D. C.

CULLIN, William W. (*M* 1938) Chief Engr. Home Insulation Div., Johns-Manville Sales Corp., 22 East 40th St., New York, and (for mail) 35 Wildwood Ave., Mt. Vernon, N. Y.

CUMMING, Ford J. (*M* 1936) Pres. (for mail) Beecher-Cumming, Inc., 820 Second Ave. S., Minneapolis, and 120 Interlachen Rd., Hopkins, Minn.

CUMMING, Robert W. (*M* 1928) Engr. & Sales Executive, Sarco Co., Inc., 183 Madison Ave., New York, and (for mail) 81 Alkamont Ave., Scarsdale, N. Y.

CUMMINGS, Carl H. (*A* 1927; *J* 1926) Pres. & Treas. (for mail) Industrial Appliance Co. of New England, 110 Arlington St., Boston, and 41 Edgehill Rd., Chestnut Hill, Mass.

CUMMINGS, G. J. (*M* 1923) Vice-Pres. (for mail) The Scott Co., 113 Tenth St., and 113 Trestle Glen Rd., Oakland, Calif.

CUMMINGS, Robert J. (*J* 1940) Engr. & Estimator (for mail) Franck & Fric, 9334 Kinsman Rd., and 11314 Glenboro Dr., Cleveland, O.

CUMMINS, George H. (*M* 1919) Dist. Mgr., Aerofin Corp., 918 United Artists Bldg., and (for mail) 16210 Ashton Rd., Detroit, Mich.

CUMMISKEY, Jerome F. (*A* 1940) Sales (for mail) Minneapolis-Honeywell Regulator Co., 2405 N. Maryland, and 4433 N. Cramer, Milwaukee, Wis.

CUMNOCK, H. (*A* 1938) Pres., Little Rock Refrigeration Co., 417 W. Capitol Ave., Little Rock, Ark.

CUNNINGHAM, John S. (*A* 1941; *J* 1937; *S* 1935) Htg. Engr., Dowagiac Steel Furnace Co., and (for mail) 311 N. Front St., Dowagiac Mich.

CUNNINGHAM, Thomas M. (*M* 1931; *J* 1930) Production Mgr. (for mail) Carrier Corp., 7-122 Merchandise Mart, Chicago, Ill.

CURRY, Roger F. (*J* 1940; *S* 1938) Jr. Engr., H. Curry Sheet Metal Works, 3142 Sutton Blvd., Maplewood, Mo.

CURTICE, Jean M. (*A* 1936) Htg. Engr. & Dist. Mgr., Citizens Utilities Co., 15 W. Fourth St., La Junta, Colo.

CURTIS, Herbert F. (*A* 1934) Chief Engr. (for mail) Henry Furnace & Foundry Co., 3471 East 49th St., Cleveland, and 59 Fourth Ave., Berea, O.

CUSHING, Charles F. (*M* 1938) Mgr. Air Cond. Sales (for mail) Bryant Heater Co., 17825 St. Clair Ave., Cleveland, and 2204 Edgerton Rd., Cleveland Heights, O.

CUSHING, R. C. (*A* 1940) Sales Engr. (for mail) Minneapolis-Honeywell Regulator Co., 1136 Howard St., San Francisco, and 1925 San Antonio Ave., Berkeley, Calif.

CUTLER, Joseph A. (*M* 1916) (Council 1920-26) Pres. (for mail) Johnson Service Co., 507 E. Michigan St., and 4811 N. Lake Dr., Milwaukee, Wis.

D

DABBS, John T. (*A* 1940) Chief Engr. (for mail) Highland Technical School, 1141 N. Highland Ave., Los Angeles, and 945-A W. Ninth St., San Pedro, Calif.

DADDARIO, Frank T. (*J* 1939) Air Cond. Engr., (for mail) Carolina Engineering Co., 220 Trust Bldg., and 104 Briar-Cliff Rd., Durham, N. C.

DAFTER, Edwin H. (*M* 1938) Sales Engr. (for mail) Carrier Corp., 12 South 12th St., Philadelphia, and 117 Crosshill Rd., Overbrook Hills, Pa.

DAHLGREN, Gustave E. (*A* 1940) Insulation Mgr. (for mail) Thorkelsson Ltd., 1331 Spruce St., and 561 Sherburn St., Winnipeg, Man., Canada.

DAHLSTROM, Godfrey A. (*A* 1927) Htg. Sales Engr., American Radiator & Standard Sanitary Corp., 312 S. Third St., and (for mail) 3721-47th Ave. S., Minneapolis, Minn.

DAITSH, Abe (*J* 1938) Estimating Engr. (for mail) Henderson-Smart (Pty.) Ltd., 143-145 Annan House, 86 Commissioner St., and 106/7 Annan House, 86 Commissioner St., Johannesburg, South Africa.

DALY, Robert E. (*M* 1931) Dir. of Engrg. (for mail) American Radiator & Standard Sanitary Corp., 40 West 40th St., New York, and 270 Bronxville Rd., Bronxville, N. Y.

D'AMBLY, A. Ernest (*M* 1924; *J* 1921) Owner (for mail) A. Ernest D'Ambly, 2101 Architects Bldg., Philadelphia, and 242 E. Montgomery Ave., Ardmore, Pa.

DANIEL, William E. (*A* 1941; *J* 1939) Partner (for mail) E. Ashby & Co., Jamaica Wharf, 20 Upper Ground, Blackfriars, and 9 Dudley House, Westmoreland St., London, W. 1, England.

DANIELSON, Wilmot A.* (*M* 1935) Col., Constructing Quartermaster, Quarry Heights, Canal Zone, Panama.

DARLING, Arthur B. (*A* 1929) Asst. Sales Mgr. (for mail) Darling Brothers, Ltd., 140 Prince St., and 4326 Sherbrooke St. W., Montreal, Que., Canada.

DARLINGTON, Allan P. (*M* 1930) Mgr. Power Apparatus (for mail) American Blower Corp., 6000 Russell St., and 5200 Haverhill, Detroit, Mich.

DARTS, John A. (*M* 1919) Kewanee Boiler Co., Inc., 101 Park Ave., New York, N. Y.

DASING, Emil (*M* 1937) Design Engr., Sears Roebuck & Co., 925 S. Homan Ave., and (for mail) 4729 N. Talman Ave., Chicago, Ill.

DAUBER, Oscar W. (*M* 1937) Consulting Engr. (for mail) 224 S. Michigan Ave., Chicago, and 366 Winnetka Ave., Winnetka, Ill.

DAUCH, Emil O. (*M* 1921) Pres. (for mail) McCormick Plumbing Supply Co., 1675 Bagley Ave., Detroit, and 729 Bedford Rd., Grosse Pointe Park, Mich.

DAVEY, Geoffrey I. (*M* 1937) Consulting Engr. (for mail) Haskins, Davey & A. Gordon Gutteridge, 60 Hunter St., and "Netherby" Bangalla St., Warrawee, Sydney, N. S. W., Australia.

DAVIDSON, John C. (*M* 1940; *J* 1936) Jr. Htg. & Air Cond. Engr., Dept. of Bldgs., 213 City Hall, and (for mail) 4708 Isabel Ave., Minneapolis, Minn.

DAVIDSON, L. Clifford (*M* 1927) Assoc. Dist. Mgr. (for mail) Buffalo Forge Co., 220 South 16th St., Philadelphia, and 322 Winding Way, Merion, Pa.

DAVIDSON, Philip L. (*M* 1924; *J* 1921) Consulting Engr. (for mail) 1600 Walnut St., Philadelphia, and 12 Curwen Rd., Villanova, Pa.

DAVIES, George W. (*M* 1918) Mgr. (for mail) G. W. Davies & Co., 19 Maclaggan St., Dunedin, C. 1, and P. O. Box 390, Dunedin, N. 2, and Colinswood, Macandrew Bay, New Zealand.

DAVIES, Reginald H. (*M* 1939) Gas Htg. Adviser, The Colonial Gas Association, Ltd., 360 Collins St., Melbourne, and (for mail) "Harelands," 5 Willsmere Rd., Kew, Victoria, Australia.

DAVIS, Arthur C.* (*M* 1920) Retired, 73 Preston St., Ridgefield Park, N. J.

DAVIS, Arthur F. (*M* 1934) Pres. (for mail) Johnson & Davis Plumbing & Heating Co., 2235 Arapahoe St., and 1901 Ivanhoe St., Denver, Colo.

DAVIS, Bert C. (*Life Member*; *M* 1904) (Council, 1917) Pres. (for mail) American Warming & Ventilating Co., 317-319 Pennsylvania Ave., and 603 W. Church St., Elmira, N. Y.

DAVIS, Calvin R. (*M* 1927) Br. Mgr. (for mail) Johnson Service Co., 2328 Locust St., and 7534 Westmoreland Dr., St. Louis, Mo.

DAVIS, Charles (*M* 1938) Engr. (for mail) Rathe Heating Corp., 700 Elton Ave., and 281 Wadsworth Ave., New York, N. Y.

DAVIS, Donald W., Jr. (*J* 1939) 832 Empire Bldg., Milwaukee, Wis.

DAVIS, Edward J. (*J* 1938) Sales Engr. (for mail) Gurney Foundry Co., Ltd., 4 Junction Rd., Toronto, and Lakeview, Ont., Canada.

DAVIS, George C. (*M* 1939; *J* 1936) Vice-Pres., Northern Public Service Corp., Ltd., 307 Power Bldg., and (for mail) 366 Ash St., Winnipeg, Man., Canada.

DAVIS, George L., Jr. (*A* 1938) Estimator, R. L. Spitzley Heating Co., 1200 W. Fort St., Detroit, and (for mail) 1300 Wayburn St., Grosse Pointe Park, Mich.

DAVIS, Joseph (*M* 1927; *A* 1926) Owner, Joseph Davis, Engr. & Contractor (for mail) 70 W. Chippewa St., and 166 Huntington Ave., Buffalo, N. Y.

DAVIS, Keith T. (*M* 1937) Chief Engr. (for mail) L. J. Mueller Furnace Co., 2005 W. Oklahoma, and 1500 E. Marion St., Milwaukee, Wis.

DAVIS, Otis E. (*M* 1929; *A* 1925) Sales Engr. (for mail) Hoffman Specialty Co., Box 98, and 1402 Third Ave., Scottsbluff, Nebr.

DAVIS, Robert J. (*M* 1939; *A* 1933) Mech. Engr., Leichnitz-Johnson Co., 110 N. First St., Yakima, Wash., and (for mail) 328 N. Shaver St., Portland, Ore.

DAVIS, Rowland G. (*A* 1921) Sales Repr., 887 Nela View Rd., Cleveland Heights, O.

DAVISON, Robert L. (*M* 1934) Dir. of Housing Research (for mail) John B. Pierce Foundation, 40 West 40th St., New York, and East Northport, L. I., N. Y.

DAWSON, Eugene F. (*M* 1934) Assoc. Prof. Mech. Engrg. (for mail) University of Oklahoma, and 229 E. Frank St., Norman, Okla.

DAWSON, Thomas L. (*M* 1930) Pres. (for mail) Thomas L. Dawson Co., 2035 Washington St., Kansas City, Mo., and Shawnee Mission Rd., Rosedale Station, Kansas City, Kan.

DAY, Harold C. (*A* 1934) Sales Office Mgr., American Radiator & Standard Sanitary Corp., 1807 Elmwood Ave., Buffalo, N. Y.

DAY, Irying M. (*A* 1936) Sales Engr. (for mail) 709 Mills Bldg., Washington, D. C., and 405 Cumberland Ave., Chevy Chase, Md.

DAY, Vincent S.* (*M* 1924) Asst. to Vice-Pres. in charge of Marketing (for mail) Carrier Corp., 302 S. Geddes St., and 316 Highland Ave., Syracuse, N. Y.

DAYNES, Joseph Henry (*M* 1938) Air Cond. Engr. (for mail) Canadian General Elec. Co., Ltd., 212-214 King St. W., and 25 Elvina Gardens, Toronto, Ont., Canada.

DEAN, Carl H. (*M* 1936) Air Cond. Engr. (for mail) Oklahoma Natural Gas Co., P. O. Box 871, and 109 East 26th Pl., Tulsa, Okla.

DEAN, Charles L. (*M* 1932) Asst. Prof. Mech. Engrg., University of Wisconsin, 305 University Extension Bldg., and (for mail) 102 Grand Ave., Madison, Wis.

DEAN, Frank J., Jr. (*J* 1935; *S* 1934) Pres., Dean-Hagny Corp., 14th & Magee St., and (for mail) 6028 Walnut St., Kansas City, Mo.

DEAN, Marshall H. (*J* 1938; *S* 1936) Chief Engr., Hotel President, 14th & Baltimore, and (for mail) 1030 West 55th St., Kansas City, Mo.

DeBERARD, Philip E. (*A* 1939) Pres. (for mail) Conditioned Air Systems, Inc., 1209 Washington St., and 1220 Greenwood Ave., Wilmette, Ill.

DEE, Leo H. (*J* 1937) Engr., L. P. Graner, Cons. Engr., 40 East 49th St., and (for mail) Shelton Hotel, Lexington Ave. at 48th St., New York, N. Y.

DEEVES, Edward W. (*J* 1940) Partner (for mail) Fred Deeves & Sons, 1711-17th Ave. W., and 2409-33rd St. W., Calgary, Alta., Canada.

DEGLER, Howard E.* (*M* 1938) Prof., Mech. Engrg., University of Texas, Austin, Tex.

DeLAND, Charles W. (*M* 1924; *J* 1923) Secy.-Treas. (for mail) C. W. Johnson, Inc., 211 N. Desplaines St., and 2021 Estes Ave., Chicago, Ill.

DELANY, John V. (*J* 1941; *S* 1938) Draftsman, Sylvania Industrial Corp., and (for mail) 1414 Prince Edward St., Fredericksburg, Va.

DeLAUREAL, William David (*J* 1940) Air Cond. Sales Engr., Fairbanks Morse & Co., and (for mail) 6330 St. Charles Ave., New Orleans, La.

DELAVAN, Nelson B. (*M* 1938) Prop. (for mail) Delavan Engineering Co., 414-12th St., and 338-42nd St., Des Moines, Ia.

DELL'ORTO, Luciano (*A* 1940; *J* 1938) Engr. Refrigerating Branch, Ing. Giuseppe Dell'Orto, 18 Via Merano, Milano (139), Italy.

DEMAREST, Richard T. (*J* 1938) Sales Promotion Dept. (for mail) Fitzgibbons Boiler Co., Inc., 101 Park Ave., and 11 Marble Hill Ave., New York, N. Y.

DEMETER, Julius (*A* 1939) Htg. & Air Cond. Engr., Julio Donoso D, Calle Lirios 375, Santiago, Chile.

DEMING, Roy E. (*A* 1939) Htg. Engr., Premier Furnace Co., and (for mail) 107 Jay St., Dowagiac, Mich.

DEMPSEY, Stephen J. (*A* 1938) Owner, Stephen J. Dempsey Co., 79 Harvard St., P. O. Box 714, Battle Creek, Mich.

DENHAM, Howard S. (*M* 1939) Consulting Engr., Cleverdon, Varney & Pike, 46 Cornhill St., Boston, and (for mail) 80 Dexter St., Malden, Mass.

DENNY, Harold R. (*A* 1934) Eastern Merchandise Mgr. (for mail) American Blower Corp., 50 West 40th St., New York, N. Y., and 429 Edgewood Ave., Westfield, N. J.

DEPPMANN, Ray L. (*A* 1937) Pres. (for mail) R. L. Deppmann Co., 5853 Hamilton Ave., and 13201 Cloverlawn Ave., Detroit, Mich.

DERER, Bernard (*A* 1940) Designer & Estimator —Sheet Metal Construction for Vtg. & Air Cond., 754 East 23rd St., Brooklyn, N. Y.

DeROO, William C. (*A* 1939) Research & Designing Engr. (for mail) Hart & Cooley Manufacturing Co., and 567 Central Ave., Holland, Mich.

DeSALES, Monteiro, Jr. (*M* 1939) Chief Engr. (for mail) Isnard & Co., Rua de Lavradie 67 1°, and Rua Senador Vergueire 193 2°, Rio de Janeiro, Brazil.

DeSOMMA, A. Edward (*J* 1937) Htg. & Vtg. Engr., George G. Sharp, 30 Church St., New York, and (for mail) 2052 Homecrest Ave., Brooklyn, N. Y.

Des REIS, John F. (*M* 1936) Regional Mgr.—Latin America, Carrier Corp., and (for mail) 409 Wendell Terrace, Syracuse, N. Y.

DETERLING, William C. (*A* 1937) Sales Agent (for mail) General Electric Co., 570 Lexington Ave., New York, and 32 W. Milton St., Freeport, L. I., N. Y.

DEVER, Henry F. (*M* 1936; *A* 1935) Vice-Pres., Minneapolis-Honeywell Regulator Co., and (for mail) 4609 Edina Blvd., Minneapolis, Minn.

DeVILBISS, Parker T. (*A* 1937) Engr. (for mail) R. E. Griffith Theatres, Inc., 7th Fl., Tower Petroleum Bldg., Dallas, Tex., and Hobbs, N. M.

DEVLIN, John (*M* 1940) Partner (for mail) Devlin Bros., 1003 Maritime Bldg., and 706 S. Carrollton Ave., New Orleans, La.

DEVORE, Angus B. (*A* 1937) Sales Engr. (for mail) James A. Messer Co., Inc., 1206 K St. N. W., Washington, D. C., and 2016 Queens Chapel Rd., (Avondale), Hyattsville, Md.

DEWEY, Ritchie P. (*M* 1934) Mgr. Temp. Control & Uni-Flo Depts. (for mail) Barber-Colman Co., and 2301 Oxford St., Rockford, Ill.

DeWITT, Earl S. (*A* 1936) Branch Mgr. (for mail) American Blower Corp., 438 Woodward Bldg., Washington, D. C., and 3224 Oliver St. N. W., Chevy Chase, Md.

DIAMOND, David D. (*J* 1937) Partner (for mail) Diamond-Neil Heating & Air Conditioning Co., 99 N. Snelling Ave., and 441 Fairview Ave. N., St. Paul, Minn.

DIBBLE, Samuel E.* (*M* 1917) (*Presidential Member*) (Pres., 1925; 1st Vice-Pres., 1924; 2nd Vice-Pres., 1923; Council, 1921-26) Supt., Patton School, Elizabethtown, Pa.

DICK, Andrew V. (*A* 1941; *J* 1935) Partner, National Heating & Insulation Co., 91 N. Pearl St., and (for mail) 305 Northern Blvd., Albany, N. Y.

DRIEMEYER, Ray C. (*J* 1937) Engr., Airtherm Mfg. Co., 700 S. Spring Ave., and (for mail) 5410 Vernon Ave., St. Louis, Mo.

DRINKER, Philip* (*M* 1922) Prof. of Industrial Hygiene (for mail) Harvard University, 55 Shattuck St., Boston, and Newton Center, Mass.

DRISCOLL, William H.* (*M* 1904) (*Presidential Member*) (Pres., 1926; 1st Vice-Pres., 1925; 2nd Vice-Pres., 1924; Treas., 1923; Council, 1918-27) Vice-Pres. (for mail) Carrier Corp., Syracuse, N. Y., and 50 Glenwood Ave., Jersey City, N. J.

DRUM, Leo J., Jr. (*J* 1939) Sales Engr. (for mail) York Ice Machinery Corp., P. O. Box 182, and 1448 Milner Crescent, Birmingham, Ala.

DuBOIS, Louis J. (*M* 1931) Air Cond. Engr., York Ice Machinery Corp., 117 South 11th St., St. Louis, and (for mail) 7451 Bland Dr., Clayton, Mo.

DUBRY, Ernest E. (*M* 1924) Asst. Supt., Central Htg., Detroit Edison Co., 2000 Second Ave., and (for mail) 9116 Dexter Blvd., Detroit, Mich.

DuCHATEAU, Manuel F. (*J* 1938) Mgr. Htg. Dept. (for mail) Crane Co., Washington St. Viaduct, and 737 Barnett, Apt. A-4, Atlanta, Ga.

DUDLEY, William H., Jr. (*A* 1940) Dist. Mgr., The Trane Co., and (for mail) 2831 Audubon St., New Orleans, La.

DUFAULT, Felix H. (*A* 1936) Mgr. Furnace Div., Que. & Maritime Provinces (for mail) General Steel Wares, Ltd., 2355 Delisle St., and 5115 Bordeaux St., Apt. 6, Montreal, Que., Canada.

DUGAN, Thomas M. (*M* 1920) Sanitary-Htg. Engr., National Tube Co., Fourth Ave. & Locust St., and (for mail) 1308 Freemont St., McKeesport, Pa.

DUITCH, Paul R. (*J* 1940) Sales Engr. (for mail) Delavan Engineering Co., 414-12th St., and 1406 Burlington Terrace, Des Moines, Ia.

DULLE, Willferd L. (*J* 1936) Asst. Secy., E. E. Souther Iron Co., 1952 Kienlen Ave., St. Louis, and (for mail) 2910 Lincoln Ave., Normandy, Mo.

DUNCAN, William A. (*A* 1930) Mgr. Process Service (for mail) Dominion Oxygen Co., Ltd., 159 Bay St., and 71 Jackson Ave., The Kingsway, Toronto, Ont., Canada.

DUNHAM, Clayton A.* (*M* 1911) Pres. (for mail) C. A. Dunham Co., 450 E. Ohio St., Chicago, and 150 Maple Hill Rd., Glencoe, Ill.

DUNLAP, A. Lee (*M* 1940) Assoc. Prof. Mech. Engrg. (for mail) Tulane University, and 1318 Nashville Ave., New Orleans, La.

DUNNE, Russell Van Dyke (*M* 1937) Chief Engr. International Div. (for mail) Carrier Corp., S. Geddes St., and 216 Robineau Rd., Syracuse, N. Y.

DUPLANT, Jean L. (*A* 1940) (for mail) Westinghouse Electric Co. of India, 294 A Bazargate St., Bombay, India, and 137-43 219th St., Springfield Gardens, L. I., N. Y.

DUTCHER, Harvey S. (*A* 1938) Air & Refrigeration, Inc., 11 West 42nd St., New York, and (for mail) 3420 Clarendon Rd., Brooklyn, N. Y.

DWYER, Thomas F. (*M* 1923) Chief of Htg. & Vtg. Div. (for mail) Board of Education, 49 Flatbush Ave. Ext., Brooklyn, and 82 Iris Ave., Floral Park, L. I., N. Y.

DYER, Wilfrid S. (*A* 1939) Partner (for mail) H. W. Dyer & Son, and 92 Byron St., Battle Creek, Mich.

DYKES, James B. (*A* 1939; *J* 1936) Vice-Pres. (for mail) T. A. Morrison & Co., Ltd., 1070 Bleury St., and 3156 Maplewood Ave., Apt. 15, Montreal, Que., Canada.

E

EADIE, John G. (*M* 1909) Consulting Engr., Eadie, Freund & Campbell Co., 110 West 40th St., New York, N. Y.

EAGLETON, Sterling P. (*M* 1936) Chief Engr. & Bldg. Supt. (for mail) National Gallery of Art, Seventh & Constitution Ave., and (for mail) 3522 S St. N. W., Washington, D. C.

EARL, Warren (*A* 1936) Vice-Pres. & Chief Engr., Esko Manufacturing Corp., 3409-11 McKinney, and (for mail) 515 Fargo, Houston, Tex.

EARLE, Frederic E. (*M* 1937) Owner (for mail) Frederic E. Earle Co., 898 Norman St., Bridgeport, and 1536 Main St., Stratford, Conn.

EASTMAN, Carl B. (*M* 1932; *J* 1929) Sales Engr. (for mail) C. A. Dunham Co., 1500 Walnut St., Philadelphia, and 530 Brookview Lane, Upper Darby P. O., Pa.

EASTWOOD, E. O. (*M* 1921) (2nd Vice-Pres., 1940; Council, 1937-40) Prof. of Mech. Engrg., Dir. Aero. Engrg. (for mail) University of Washington, and 4702-12th Ave. N. E., Seattle, Wash.

EASTWOOD, Harry F. (*M* 1925) Mgr. Anthracite Equipment Exhibit, Anthracite Industries, Inc., Chrysler Bldg., and 101 Park Ave., New York, and (for mail) 157 Frankel Blvd., Merrick, L. I., N. Y.

EATON, Byron K. (*M* 1920) Htg. & Air Cond. Sales Mgr., Major Appliance Co., 2558 Farnam St., and (for mail) 817 South 38th St., Omaha, Nebr.

EATON, William G. M. (*A* 1934) Sales Engr., Pease Foundry Co., Ltd., 227 Victoria St., and (for mail) 300 Wellesley St., Toronto, Ont., Canada.

EBERT, William A. (*M* 1920) Partner (for mail) Ebert Air Conditioning, 1026 W. Ashby Pl., and 2151 W. Kings Highway, San Antonio, Tex.

EDER, James (*J* 1940) (for mail) Walker & Eder, Inc., 37 West 39th St., and 29 Washington Sq., New York, N. Y.

EDGE, Alfred J. (*M* 1938) Engr. in charge Htg. & Air Cond. (for mail) John F. Reynolds, Cons. Engr., Rm. 316, Duval Bldg., and 2864 Olgo Pl., Jacksonville, Fla.

EDWARDS, Arthur W. (*M* 1936) Dist. Mgr., The Trane Co., 626 Broadway, and (for mail) 3423 Paxton Ave., Cincinnati, O.

EDWARDS, Don J. (*A* 1933) Vice-Pres. (for mail) General Heat & Appliance Co., 596 Commonwealth Ave., Boston, and 8 Devon Terrace, Newton, Mass.

EDWARDS, Junius D.* (*M* 1936) Asst. Dir. of Research (for mail) Aluminum Research Laboratories, Aluminum Co. of America, P. O. Box 772, New Kensington, and 536 Sixth St., Oakmont, Pa.

EDWARDS, Paul A. (*M* 1919) Pres. (for mail) G. F. Higgins Co., 608 Wabash Bldg., Pittsburgh, and 3074 Pinehurst Ave., Dormont, Pa.

EGGLESTON, Herbert L. (*M* 1938) Mgr. Natural Gas & Refining Depts., Gilmore Oil Co., 2423 East 28th St., Los Angeles, and (for mail) 1017 Cumberland Rd., Glendale, Calif.

EHLERS, Jacobus (*A* 1939; *J* 1937) Resident Engr. (for mail) Carrier Engrg. South Africa, Ltd., P. O. Box 3013 and Berea, Camps Bay, Cape Town, South Africa.

EHRLICH, M. William* (*M* 1916) Chief Engr., Commodore Heaters Corp., 11 West 42nd St., New York, N. Y., and (for mail) 56 Ridge Rd., Lyndhurst, N. J.

EICHER, Hubert C. (*M* 1922) Chief, School Plant Div., Dept. of Public Instruction, State Capitol, and (for mail) 207 North 30th St., Harrisburg, Pa.

EISELE, Dudley E. (*A* 1938) Owner (for mail) Eisele Engineering Co., 427 W. College Ave., and 1735 N. Morrison St., Appleton, Wis.

EISS, Robert M. (*M* 1933; *J* 1930) Mech. Engr., Kimberly-Clark Corp., and (for mail) Rte. 1, Adella Beach, Neenah, Wis.

EKINGS, Robert M., Jr. (*M* 1938) Air Cond. Engr., General Electric Co., 5 Lawrence St., Bloomfield, and (for mail) 233 Prospect St., Apt. 5D, East Orange, N. J.

EKLUND, Gaylord Paul (*S* 1940) Student (for mail) University of Minnesota, Pioneer Hall, Minneapolis, Minn., and 612 Bohm St., Rockford, Ill.

EKLUND, Karl G. (*M* 1938) Consulting Engr. (for mail) Karl G. Eklunds Ingeniorsbyra, Brunkebergstorg 15, and Storangen, Stockholm, Sweden.

ELIZARDI, Ralph (*J* 1940) Design Engr., Leo S. Weil & Walter B. Moses, 425 S. Peters St., and (for mail) 2222 Dublin St., New Orleans, La.

ELLINGWOOD, Elliott L. (*M* 1909) Consulting Engr. (for mail) 124 W. Fourth St., Rm. 700, Los Angeles, and 210 S. Los Robles Ave., Pasadena, Calif.

ELLIOT, Edwin (*M* 1929) (for mail) Edwin Elliot & Co., 560 North 16th St., and 403 W. Price St., Germantown, Philadelphia, Pa.

ELLIOTT, Irwin (*A* 1937) Chief Engr., Universal Oven Co., 271 Broadway, New York, and (for mail) 103 Penfield Ave., Croton, N. Y.

ELLIOTT, Louis (*M* 1932) Consulting Mech. Engr., Ebasco Services, Inc., 2 Rector St., Rm. 1530, New York, N. Y.

ELLIOTT, Norton B. (*A* 1934) Sales Engr., American Blower Corp., 632 Fisher Bldg., Detroit, Mich.

ELLIS, Fred E. (*M* 1923) Sales Mgr. (for mail) Imperial Iron Corp., Ltd., 30 Jefferson Ave., and 9 Princeton Rd., Toronto, Ont., Canada.

ELLIS, Frederic R. (*M* 1913) Buerkel & Co., Inc., 18-24 Union Park St., Boston, and (for mail) 131 Beacon St., Hyde Park, Mass.

ELLIS, Gershom P. (*M* 1935) Dist. Mgr., Combustion Engineering Co., Inc., and (for mail) 1118 Delta Ave., Cincinnati, O.

ELLIS, George W. (*J* 1940) Engr., Acme Heating & Ventilating Co., 4224 S. Lowe Ave., and (for mail) 4004 S. Artesian, Chicago, Ill.

ELLIS, Harry W. (*Life Member*; *M* 1923; *A* 1909) Chairman of the Board, Johnson Service Co., 507 E. Michigan St., and (for mail) 2317 E. Wyoming Pl., Milwaukee, Wis.

ELWOOD, Willis H. (*M* 1936) Br. Mgr., Holland Furnace Co., 209 King St., Ithaca, N. Y.

ELY, Roland S. (*S* 1940) Student, Michigan State College, and (for mail) 525 Charles St., East Lansing, Mich.

EMANUELS, Mason (*J* 1939) Sales Engr., Pacific Scientific Co., 25 Stillman St., San Francisco, and (for mail) 2516 Stockbridge Dr., Oakland, Calif.

EMERSON, Ralph R. (*M* 1922) Pres., Emerson Swan Goodyer Co., 712 Beacon St., Boston, and (for mail) 44 Whitney Rd., Newtonville, Mass.

EMMERT, Luther D. (*M* 1919) Sales Engr. (for mail) Buffalo Forge Co., 20 N. Wacker Dr., Chicago, and 1740 Hinman Ave., Evanston, Ill.

ENDERS, Clarence E. (*A* 1938) Mgr. & Engr. (for mail) Electrol Oil Burner Co., 417 S. E. Clay, and 1813 Southeast 60th Ave., Portland, Ore.

ENGDAHL, Richard B.* (*J* 1938) Special Research Asst. (for mail) University of Illinois, 102 Mech. Engrg. Lab., and 1108 W. Stoughton, Urbana, Ill.

ENGLE, Alfred (*A* 1923) Secy. (for mail) Jenkins Bros., 80 White St., New York, and 1 Edgewood Rd., Scarsdale, N. Y.

ENGLISH, Harrold (*M* 1935; *A* 1930) Pres. (for mail) English & Lauer, Inc., 1978 S. Los Angeles St., and 515 S. Norton, Los Angeles, Calif.

ENSIGN, Willis A. (*M* 1935) Vice-Pres., Frontier Engineering Corp., 986 Ellicott Sq. Bldg., Buffalo, and (for mail) Shadagee Rd., Eden, N. Y.

ERICKSON, Harry H. (*A* 1929) Sales Engr. (for mail) Haynes Selling Co., Inc., 1124 Spring Garden St., Philadelphia, and 25 Eagle School Rd., Strafford, Pa.

ERICSSON, Eric B. (*M* 1933) Engr.-Custodian, Board of Education, and (for mail) 605 West 116th St., Chicago, Ill.

ERIKSON, Harald A. (*M* 1939) Vice-Pres., A. B. Svenska Flaktfabriken, Kungsgatan 16-18, Stockholm 7, and (for mail) Nockebyvagen 61, Nockeby, Sweden.

ERISMAN, Percival H., Jr. (*M* 1936) Vice-Pres. (for mail) Washington Refrigeration Co., 1733 14th St. N. W., Washington, D. C., and 4 Waltonway Rd., Belle Haven, Alexandria, Va.

ESCHENBACH, Samuel P. (*J* 1935) 1st Lt. 62nd C. A. (A. A.), Fort Totten, S. I., N. Y.

ESPENSCHIED, Frederic F. (*M* 1940) Sales Engr., 410 Hill Bldg., and (for mail) 3373 Stuyvesant Pl. N. W., Washington, D. C.

ESSEX, Jesse L. (*M* 1940) Chief Insulation & Paint Div., Armco International Corp., Middletown, O.

ESTEP, Leslie G. (*M* 1936) Chief Engr., United Wall Paper Factories, Inc., Chicago, and (for mail) 314 N. Kensington Ave., La Grange Park, Ill.

ESTES, Edwin C. (*A* 1936) Mech. Draftsman (for mail) Northern Pacific Ry., General Office, St. Paul, and Victoria Rd., Mendota, Minn.

EUTSLER, Eugene Ernest (*J* 1938) Sales Engr., Buffalo Forge Co., 1801 Tower Petroleum Bldg., Dallas, Tex.

EVANS, Bruce L. (*M* 1938; *A* 1937) Design Engr. & Service Mgr. (for mail) Oil Heat, Inc., 3804 W. Pine Blvd., and 6334 Waterman Ave., St. Louis, Mo.

EVANS, Edwin C. (*M* 1919) Sales Engr. (for mail) B. F. Sturtevant Co., 607 Eckel Bldg., and 307 Montgomery St., Syracuse, N. Y.

EVANS, Richard W. (*M* 1940) (for mail) Conditioned Air Equipment Co., 2455 University Ave., St. Paul, and 1912 Emerson Ave. S., Minneapolis, Minn.

EVANS, William A. (*M* 1918) Dist. Mgr. (for mail) Aerofin Corp., 1121 Fidelity Bldg., Cleveland, and 3642 Winchell Rd., Shaker Heights, Cleveland, O.

EVELETH, Charles F.* (*M* 1911) Wilbur Watson & Associates, Cons. Engrs., 4614 Prospect Ave., and (for mail) 2030 East 115th St., Cleveland, O.

EVEREST, R. Harry (*M* 1935) Engrg. & Sales, Sheldons, Ltd., Galt, and (for mail) 235 Waterloo St., Preston, Ont., Canada.

EVERETTS, John, Jr.* (*M* 1938; *A* 1935; *J* 1929) Frigidaire Div., General Motors Sales Corp., 300 Taylor St., and (for mail) 110 Oak Knoll Dr., Dayton, O.

EWENS, Frank G.* (*M* 1937) Htg. Engr., Defense Industries, Ltd., 1155 Beaver Hall Sq., and (for mail) Apt. 14, 4800 Cote Des Neiges, Montreal, Que., Canada.

EZZ-EL-DIN, Kamal (*A* 1941; *J* 1938) 78 Helwan St., Mounirah, Cairo, Egypt.

F

FABER, Oscar (*M* 1934) Hayes Court, Hayes Lane, Kenley, Surrey, England.

FABLING, Walter D. (*A* 1937) Sales Mgr. (for mail) Sterling Electric Motors, Inc., 5401 Telegraph Rd., Los Angeles, and 1950 Del Mar Ave., San Marino, Calif.

FAGIN, Daniel J. (*M* 1932) Chief Sales Engr. (for mail) Laclede Gas Light Co., 1017 Olive St., and 5846 Lindenwood Ave., St. Louis, Mo.

FAHNESTOCK, Maurice K.* (*M* 1927) Research Assoc. Prof. in Mech. Engrg. (for mail) University of Illinois, 214 Mech. Engrg. Lab., and 702 W. Vermont St., Urbana, Ill.

FAILE, Edward H. (*M* 1934) Consulting Engr. (for mail) 608 Fifth Ave., New York, N. Y., and R. F. D. 1, Westport, Conn.

FALK, David S. (*J* 1937) Sales Engr., The Trane Co., 8316 Woodward Ave., and (for mail) 20 East Euclid, Apt. 415, Detroit, Mich.

FALTENBACHER, Harry J. (*M* 1930) Owner, Harry J. Faltenbacher, Inc., 235 E. Wister St., Philadelphia, Pa.

FALVEY, John D. (*M* 1922) Consulting Engr. (for mail) 316 N. Eighth St., St. Louis, and 6636 Pershing Ave., University City, Mo.

FAMILETTI, A. Robert (*M* 1938; *J* 1930) Assoc. Mech. Engr., Industrial Dept., Navy Yard, and (for mail) 2230 Tasker St., Philadelphia, Pa.

FARBER, Louis M.* (*A* 1940; *J* 1936) Engr. (for mail) Natkin & Co., 1800 Baltimore, and 3714 Flora Ave., Kansas City, Mo.

FARLEY, W. F. (*M* 1930) Sales Repr., American Radiator & Standard Sanitary Corp., 50 West 40th St., New York, and (for mail) 28 Elm St., New Rochelle, N. Y.

FARNES, Bert W. (*A* 1938) Vice-Pres. & Gen. Mgr. (for mail) Control Equipment Co., 304 Selling Bldg., and 3565 N.E. Hollyrood Court, Portland, Ore.

FARNHAM, Roswell (*M* 1920) (Council, 1927-33) Dist. Mgr. Engrg. Sales (for mail) Buffalo Forge Co., P. O. Box 985, and 5 Clarendon Pl., Buffalo, N. Y.

FARRAR, Cecil W. (*M* 1920; *A* 1918) (Treas., 1930; Council, 1930) Vice-Pres., W. A. Case & Son Mfg. Co., 31 Main St., and (for mail) 29 Oakland Pl., Buffalo, N. Y.

FARRINGTON, S. Edward (*M* 1940) Designer (for mail) Moody & Hutchison, 1701 Architects Bldg., and 3123 Rawle St., Philadelphia, Pa.

FARROW, Ernest E. (*A* 1938) Pres., E. E. Farrow, Inc., 2808 Inwood Rd., and (for mail) 1518 Kings Highway, Dallas, Tex.

FARROW, Hollis L. (*J* 1937) Stokol Stokers Engr., Sprague, Breed, Stevens & Newhall, Inc., 135 Broad St., and (for mail) 910 Lynnfield St., Lynn, Mass.

FATZ, Joseph L. (*M* 1935) Htg. & Vtg. Engr., Board of Education, 228 N. LaSalle St., Rm. 536, and (for mail) 5914 W. North Ave., Chicago, Ill.

FAUST, Frank H.* (*M* 1936; *J* 1930) Air Cond. & Commercial Refrigerating Dept. (for mail) General Electric Co., 5 Lawrence St., Bloomfield, and 239 Vreeland Ave., Nutley, N. J.

FAXON, Harold C. (*M* 1937) Engr. Appliance Section (for mail) Borneo Co., Ltd., Mercantile Bank Bldg., and 73 Grange Rd., Singapore, Straits Settlements.

FEAR, S. Lorne (*M* 1938) Asst. Mech. Engr. (for mail) Hydro Electric Power Commission of Ontario, 620 University Ave., Toronto, and (for mail) 18 Vesta Dr., Toronto 10, Ont., Canada.

FEBREY, Ernest J. (*Life Member*; *M* 1903) Pres. (for mail) E. J. Febrey & Co., Inc., 616 New York Ave. N. W., and 2331 Cathedral Ave. N. W., Washington, D. C.

FEDDERS, Melvin P. (*M* 1938) Chief Test Engr., Minneapolis-Honeywell Regulator Co., 2747 Fourth Ave. S., and (for mail) 5212 W. Nokomis Pkwy., Minneapolis, Minn.

FEDER, Nathan (*J* 1938) Proprietor (for mail) East Side Sheet Metal Co., 403 East 74th St., and 3525 Decatur Ave., New York, N. Y.

FEEHAN, J. B. (*Life Member*; *M* 1923) Pres.-Treas. (for mail) John B. Feehan, Inc., 58 Spring St., Lynn, and 4 Long View Dr., Marblehead, Mass.

FEELY, Frank J. (*M* 1935; *A* 1929) Mgr. of Sales, Taylor Supply Co., 700 Monroe Ave., Detroit, and (for mail) 950 Trombley Rd., Grosse Pointe Park, Mich.

FEHLIG, John B. (*Life Member*; *M* 1918) Mgr. (for mail) Excelsior Furnace Co. Div., 528 Delaware St., and 2927 Brooklyn Ave., Kansas City, Mo.

FEHLIG, John B., Jr. (*A* 1941) Sales Mgr., Excelsior Htg. Supply Div., 528 Delaware, and (for mail) 1194 East 65th St., Kansas City, Mo.

FEINBERG, Emanuel (*J* 1937) Gen. Mgr. (for mail) Thermalair Engineering Co., 439 Penobscot Bldg., and 3359 Cortland Ave., Detroit, Mich.

FEIRN, William H. (*M* 1938) Engr., C. A. Hooper Co., 453 W. Gilman St., and (for mail) Shorewood Hills, Madison, Wis.

FELDERMANN, William (*A* 1937) Pres. (for mail) Walton Laboratories, Inc., 1186 Grove St., Irvington, and 357 Irving Ave., South Orange, N. J.

FELDMAN, A. M.* (*Life Member*; *M* 1903) Consulting Engr., 320 Central Park West, New York, N. Y.

FELDSTEIN, Harold (*J* 1938) 1807 Eighth St., Brownwood, Tex.

FELS, Arthur B. (*M* 1919) Pres. (for mail) The Fels Co., 42 Union St., Portland, and Yarmouth, Me.

FELTWELL, Robert H. (*Life Member*; *M* 1905) Htg. Engr., U. S. Radiator Corp., 2321 Fourth St. N. E., and (for mail) 1370 Oak St. N. W., Washington, D. C.

FENNER, N. Paul (*A* 1928) Dist. Office Mgr. (for mail) Hoffman Specialty Co., 130 N. Wells St., Chicago, and 168 Avon Rd., Elmhurst, Ill.

FENSTERMAKER, Sidney E. (*M* 1909) Partner (for mail) S. E. Fenstermaker & Co., 937 Architects & Builders Bldg., Indianapolis, and Carmel, Ind.

FERDERBER, Dr. Murray B.* (*M* 1938) Fellow of Dept. of Industrial Hygiene, University of Pittsburgh Medical School, and (for mail) 5722 Fifth Ave., Pittsburgh, Pa.

FERGESTAD, Marvin L. (*M* 1938; *J* 1935) Sales Engr., Pacific Lumber Co., 35 E. Wacker Dr., Chicago, Ill., and (for mail) 4013 Lincoln Place Dr., Des Moines, Ia.

FERGUSON, Ralph R. (*M* 1934; *A* 1927; *J* 1925) Mgr. Air Cond. Dept., American Blower Corp., 50 West 40th St., New York, N. Y., and (for mail) 160 Prospect St., East Orange, N. J.

FERRARINI, Joseph (*A* 1939; *J* 1937) Confidential Staff Asst., Washington Gas Light Co., 411 Tenth St. N. W., Washington, D. C., and (for mail) 1728 Queen's Lane, Apt. 180, Colonial Village, Arlington, Va.

FIDELIUS, Walter R. (*M* 1936) Sales, Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, and (for mail) 135 Amersfort Pl., Brooklyn, N. Y.

FIEDLER, Harry W. (*M* 1923) Owner (for mail) Air Conditioning Utilities Co., 8 West 40th St., New York, and 16 Dobbs Terrace, Scarsdale, N. Y.

FIFE, G. Donald (*M* 1937; *A* 1931; *J* 1929) Air Cond. Engr., Architect of the Capitol, and (for mail) 211 Delaware Ave., Washington, D. C.

FIGGIS, Thomas G. (*A* 1937; *J* 1936) Tech. Sales Engr., J. & E. Hall, Ltd., Dartford Ironworks, Kent, England.

FINERAN, Edward V. (*A* 1940; *J* 1935) Mgr. Industrial Dept., Washington Gas Light Co., 411 Tenth St. N. W., Washington, D. C., and (for mail) 305 Edgewood Ave., Silver Spring, Md.

FINNEY, Brandon (*M* 1937) Tech. Engr., Southern Water Heater Corp., 1900 N. Alameda, Compton, and (for mail) 721 Via de La Paz, Pacific Palisades, Calif.

FINNIGAN, William T. (*M* 1939) Mgr. and Engr. (for mail) Finnigan Bros., 1421 Southeast 20th Ave., and 1634 Southeast 29th Ave., Portland, Ore.

FIRESTONE, Maurice T. (*M* 1939) Mgr. of Dealers Northeastern Dist. (for mail) Carrier Corp., Chrysler Bldg., New York, N. Y., and 1365 Rittenhouse St. N. W., Washington, D. C.

FISCHER, Frank P. (*A* 1940) Prop. (for mail) Frank P. Fischer Engineering Co., 412 Dryades St., and 1820 Audubon St., New Orleans, La.

FISCHER, Lawrence W. (*J* 1937) Production Mgr., Anemostat Corp. of America, 10 East 39th St., New York, and (for mail) 91 Islip Ave., Islip, L. I., N. Y.

FISHER, John T. (*J* 1936) Chief Engr. (for mail) United Equipment & Supply Co., 1812 M St. N. W., and 1709-19th St. N. W., Washington, D. C.

FITTS, Joseph C. (*M* 1930) Secy., Heating, Piping & Air Conditioning Contractors National Association, 1250 Sixth Ave., New York, N. Y., and (for mail) 215 Kenilworth Rd., Ridgewood, N. J.

FITZ, Jean C. (*M* 1924) Engr. & Estimator, J. L. Murphy, Inc., 340 East 44th St., New York, and (for mail) 405 Webster Ave., New Rochelle, N. Y.

FITZGERALD, Matthew J. (*M* 1934) Secy.-Treas., Standard Asbestos Manufacturing Co., 820 W. Lake St., Chicago, and (for mail) 1117 N. Linden Ave., Oak Park, Ill.

FITZ GERALD, William E. (*J* 1936; *S* 1935) Pres. & Gen. Mgr., Fitz Gerald Plumbing & Heating Co., Inc., 939-41 Louisiana Ave., and (for mail) 1905 Gilbert St., Shreveport, La.

FITZSIMONS, J. Patrick (*A* 1940; *J* 1934; *S* 1932) Mgr. Air Cond. Dept. (for mail) Trane Co. of Canada, Ltd., 4 Mowat St., and 927 St. Clair Ave. W., Toronto, Ont., Canada.

FLANAGAN, James B. (*A* 1939) Sales Mgr. (for mail) Warden-King, Ltd., 2104 Bennett Ave., and 4244 Westhill Ave., Montreal, Que., Canada.

FLARSHEIM, Clarence A. (*A* 1940; *J* 1933) P. O. Box 56, and (for mail) 3720 Holmes St., Kansas City, Mo.

FLEAK, William D. (*A* 1938) Lab. Engr., Industrial Training Institute, 2141 Lawrence Ave., and (for mail) 4535 N. Mozart St., Chicago, Ill.

FLEISHER, Walter L.* (*M* 1914) (1st Vice-Pres., 1940; 2nd Vice-Pres., 1939; Council, 1936-40) Consulting Engr. (for mail) 11 West 42nd St., New York, and Saw Mill Farm, New City, N. Y.

FLINK, Carl H. (*M* 1923) Mech. Engr. (for mail) American Radiator & Standard Sanitary Corp., 675 Bronx River Road, Yonkers, and 111 Magnolia Ave., Mt. Vernon, N. Y.

FLINT, Coll T. (*M* 1919) Sales Mgr. (for mail) H. B. Smith Co., Inc., 640 Main St., Cambridge, and 8 Searle Ave., Brookline, Mass.

FLORETH, John J. (*M* 1939) Mgr., Air Cond. Div. Westerlin & Campbell Co., 1113 Cornelia, and (for mail) 5718 N. Richmond Ave., Chicago, Ill.

FLUCKEY, Kenneth N. (*J* 1940) Lab. Asst., Washington Gas Light Co., 411 Tenth St. N. W., and (for mail) 417 W. Clifton Terrace Apts., Washington, D. C.

FOERSTNER, George C. (*A* 1938) Mgr., Amana Society, Amana, Ia.

FOGG, Joseph H. (*J* 1940) Sales Engr. (for mail) B. F. Sturtevant Co., 713 Mills Bldg., Washington, D. C., and 214 E. Bellefonte Ave., Alexandria, Va.

FOLEY, Daniel F. (*M* 1939; *A* 1937) Sales Engr., Purch. Div., U. S. Supply Co., 1315 West 12th St., Kansas City, Mo., and (for mail) 122 Spruce St., Leavenworth, Kan.

FOLEY, John J. (*A* 1938) Pres. (for mail) Weathermakers (Canada) Ltd., 593 Adelaide St. W., and 46 Castle Knock Rd., Toronto, Ont., Canada.

FOLEY, J. Lester (*M* 1938) Precipitron Specialist, Westinghouse Electric & Manufacturing Co., 306 Fourth Ave., Pittsburgh, Pa., and (for mail) 3567 Riedham Rd., Shaker Heights, O.

FOLSOM, Rolfe A. (*M* 1938) Vice-Pres. (for mail) W. R. Ames Co., 150 Hooper St., San Francisco, and 2411 Easton Dr., Burlingame, Calif.

FOOTE, Earle E. (*M* 1936) Gen. Supt., Consumers Central Heating Co., 108 East 11th St., and (for mail) 3412 North 28th St., Tacoma, Wash.

FOOTE, James H., Jr. (*S* 1940) Student, Michigan State College, East Lansing, and (for mail) Okemos, Mich.

FORBES, Homer B., Jr. (*J* 1941; *S* 1938) Sales Engr. (for mail) Niagara Blower Co, 407 S. Dearborn St., and 3218 Balmoral Ave., Chicago, Ill.

FORDERBRUGGEN, Kevin J. (*A* 1941; *J* 1938) Engr. (for mail) Minnesota Valley Natural Gas Co., 222 S. Front St., and Ben Pay Hotel, Mankota, Minn.

FORFAR, Donald M. (*M* 1917) Mech. Engr., Grinnell Co., Inc., 240 Seventh Ave. S., and (for mail) 4817 Emerson Ave. S., Minneapolis, Minn.

FORRESTER, Norman J. (*A* 1936) Mgr. Contract Div., Garth Co., 750 Belair Ave., and (for mail) 4800 Westmore Ave., Montreal, Que., Canada.

FORSBERG, William (*M* 1919) Secy. & Supt. (for mail) The Hopson & Chapin Manufacturing Co., 231 State St., New London, and Quaker Hill, Conn.

FORSLUND, Oliver A. (*M* 1936) Gen. Mgr., Forslund Pump & Machinery Co., 1717-19 Main St., and (for mail) 108th St. & State Line, Kansas City, Mo.

FOSS, Edwin R. (*A* 1936) Dist. Mgr. (for mail) The Powers Regulator Co., 407 Bona Allen Bldg., and 257 Bolling Rd. N. E., Atlanta, Ga.

FOSTER, Charles (*M* 1923) Consulting Engr. (for mail) 316 Medical Arts Bldg., and 2831 E. First St., Duluth, Minn.

FOSTER, James M. (*M* 1930; *A* 1920) Owner (for mail) J. M. Foster, 4485 Olive St., St. Louis, and 7021 Lindell Ave., University City, Mo.

FOSTER, John G. (*J* 1938) Sales Engr. (for mail) Air Conditioning Utilities Co., 8 West 40th St., and 2635 Sedgwick Ave., New York, N. Y.

FOSTER, Philip H. (see Special Service Roll, p. 73).

FOULDS, P. A. L. (*M* 1916) Partner (for mail) Hubbard, Rickerd & Blakeley, Cons. Engrs., 110 State St., Boston, and 72 Whitin Ave., Revere, Mass.

FOWLES, Harry H. (*A* 1940; *J* 1934) Htg. Engr. (for mail) Carman-Thompson Co., 12-14 Lincoln St., Lewiston, and 176 Summer St., Auburn, Me.

FOX, Ernest (*M* 1935) Asst. Engr. (for mail) C. A. Dunham Co., Ltd., 1523 Davenport Rd., and 409 Glenholme Ave., Toronto, Ont., Canada.

FOX, John H. (see Special Service Roll, p. 73).

FOX, William K. (*A* 1939) Supt., Northwest Stove Works, Inc., 2345 S. E. Gladstone St., and (for mail) 6112 N. E. Prescott St., Portland, Ore.

FRANCK, Peter (*J* 1938) Secy., Tiltz Air Conditioning Corp., 230 Park Ave., New York, and (for mail) 3311A 69th St., Jackson Heights, L. I. ,N.Y.

FRANK, John M. (*M* 1918; *A* 1912) Pres. (for mail) Ilg Electric Ventilating Co., 2850 N. Crawford Ave., Chicago, and 1152 Chatfield Rd., Hubbard Woods, Ill.

FRANK, Olive E.* (*M* 1919) Pres., Frank Heaters, Inc., 150 Railroad Ave., and (for mail) 288 Graham Ave., Paterson, N. J.

FRANKEL, Gilbert S. (*M* 1926) Mgr., Federal & Marine Dept. (for mail) Buffalo Forge Co.—Buffalo Pumps, Inc., 640 Woodward Bldg., and 3601 Connecticut Ave., Washington, D. C.

FRANKLIN, Ralph S. (*M* 1919) Pres.-Treas. (for mail) Albert B. Franklin, Inc., 38 Chauncy St., Boston, and 320 Grove St., Melrose, Mass.

FRANKLIN, Sam H., Jr. (*A* 1938) Prop. (for mail) S. H. Franklin, Jr., 921 Main St., and 204 Colonial Court, Lynchburg, Va.

FRASER, James J. (*A* 1936) Director (for mail) Honeywell Brown, Ltd., Wadsworth Rd., Perivale, Greenford, Middlesex, England.

FRAZIER, J. Earl (*A* 1936) Vice-Pres. & Treas. (for mail) Frazier-Simplex, Inc., 436 East Beau St., and 7 Wilmont Ave., Washington, Pa.

FREDERICK, Holmes W. (*M* 1937) Asst. Htg. Engr., Cornell University, Morrill Hall, and (for mail) 103 Harvard Place, Ithaca, N. Y.

FREDERICK, Walter L. (*A* 1937) Pres. (for mail) Bryant Air Conditioning Corp., 1626 K St. N. W., and 15 Abingdon Rd., Friendship Station, Washington, D. C.

FREEMAN, Alfred W. (*J* 1940; *S* 1939) Gen. Mgr., Freeman Co., 119 Greenpoint Ave., Brooklyn, and (for mail) 31-05 88th St., Jackson Heights, L. I., N. Y.

FREEMAN, Edwin M. (*A* 1937) Vice-Pres. & Sales (for mail) Canadian Asbestos Co., 316-322 Youville Sq., and 66 Courcelette, Montreal, Que., Canada.

FREEMAN, J. Albert (*A* 1940; *J* 1938) Engr. (for mail) Western Engineering Co., 1623 Southeast 11th Ave., and Rte. 8, Box 2001, Portland, Ore.

FREITAG, Frederic G. (*M* 1932) 9 Harrison St., Mt. Vernon, N. Y.

FRENCH, Donald (*M* 1926) Vice-Pres. in Charge of Engrg. (for mail) Carrier Corp., S. Geddes St., and 618 Rugby Rd., Syracuse, N. Y.

FRENTZEL, Herman C. (*M* 1936) Chief Engr., The Heil Co., 3000 W. Montana St., and (for mail) 4363 N. Wildwood Ave., Milwaukee, Wis.

FRIEDLER, Joseph J., Jr. (*M* 1940) So. Dist. Mgr. (for mail) Ilg Electric Ventilating Co., 304 Natchez Bldg., and 5411 Fontainbleau Dr., New Orleans, La.

FRIEDLINE, James M. (*J* 1937) Sales Engr., Fred Keating Coal Co., 910 Walnut St., and (for mail) 3021 Ingersoll, Des Moines, Ia.

FRIEDMAN, Arthur (*A* 1936) (for mail) Air Controls, Inc., 1933 West 114th St., Cleveland, and 15700 S. Moreland Blvd., Shaker Heights, O.

FRIEDMAN, D. Harry, Jr. (*M* 1936) Air Cond. & Industrial Engr. (for mail) Peoples Water and Gas Co., 15th & Washington Ave., and 1620 Pennsylvania Ave., Apt. 201, Miami Beach, Fla.

FRIEDMAN, Ferdinand J.* (*M* 1921) Consulting Engr. (for mail) McDougall & Friedman, 1221 Osborne St., Montreal, Que., Canada, and 31 Union Square, New York, N. Y.

FRIEDMAN, Milton (*A* 1939; *J* 1935; *S* 1933) H. F. Klawuhn Genl. Contractor, 34-24 82nd St., Jackson Heights, L. I., and (for mail) 333 West End Ave., Apt. 14C, New York, N. Y.

FRIMET, Maurice (*A* 1941; *J* 1936) Engr., S. I. Heating & Air Conditioning Co., 418 Castleton Ave., New Brighton, and (for mail) 15 Mundy Ave., West Brighton, S. I., N. Y.

FRITZ, Charles V. (*J* 1936; *S* 1933) Designer and Estimator, Charles F. Fritz, 67 W. Merrick Rd., and (for mail) 26 Cottage Court, Freeport, L. I., N. Y.

FROELICH, H. Allen (*A* 1939) Gen. Mgr., The Glen'Aire Cooler Co., 133½ W. Seventh St., and (for mail) Box 404, Junction City, Kan.

FUKUI, Kunitaro (*M* 1926) Auditor, Oriental Carrier Engineering Co., Ltd., Toyo Menka Bldg., Koraibashi-Higashi-ku, Osaka, Japan.

FULLER, Charles A. (*M* 1913) Partner (for mail) Slocum & Fuller, Cons. Engrs., 18 East 41st St., New York, and 501 E. 5th St., Mt. Vernon, N. Y.

FULLER, Elbridge W. (*M* 1938) S. W. Zone Mgr., Commercial & Air Conditioning (for mail) Frigidaire Div., General Motors Sales Corp., Taylor St., and 308 Central Ave., Dayton, O.

FUNCK, Elmer H. (*M* 1939; *A* 1928; *J* 1926) Sales Engr. (for mail) Johnson Fan & Blower Corp., 1319 W. Lake St., and 4545 N. Hamilton Ave., Chicago, Ill.

FUNK, Donald S. (*A* 1940) Pres. (for mail) Airmode Mfg. Co., 325 W. Huron St., Chicago, and 530 Washington Blvd., Oak Park, Ill.

G

GABLE, H. Raymond (*A* 1939) Design Engr., Union Steel Products Co., 500 N. Berrien St., and (for mail) 303 Ingham St. S., Albion, Mich.

GAIR, Kenneth B. (*J* 1939) Sales Engr. (for mail) B. F. Sturtevant Co., 812 Michigan Bldg., Detroit, and 188 West Cambourne, Ferndale, Mich.

GALLAGHER, Frank H. (*A* 1938) Asst. Engr., Board of Public Education, and (for mail) 2727 Strachan Ave. (16), Pittsburgh, Pa.

GALLIGAN, Andrew B. (*M* 1921) Pres. (for mail) Galligan Bros., Inc., 716 South 51st St., and 5987 Woodbine Ave., Philadelphia, Pa.

GAMBLE, Cary B. (*M* 1939; *A* 1935) Engr., Leo S. Weil & Walter B. Moses, 425 S. Peters St., and (for mail) 4235 S. Carrollton Ave., New Orleans, La.

GAMMILL, Oscar E., Jr. (*M* 1940; *A* 1937; *J* 1930) Sales Engr. (for mail) Carrier Corp., 1413 Hibernia Bank Bldg., and 5515 Magnolia St., New Orleans, La.

GANGE, Frank B. (*M* 1937) Managing Dir., Gordon & Co., Ltd., 185 Yuen Ming Yuen Rd., Shanghai, China.

GANNON, Russell R. (*M* 1939) Pres. (for mail) Russell R. Gannon Co., Gwynne Bldg., and 1824 Fairfax Ave., Cincinnati, O.

GANT, H. P.* (*M* 1915) (*Presidential Member*) (Pres., 1923; 1st Vice-Pres., 1922; 2nd Vice-Pres., 1921; Council, 1918-24) R. D. No. 1, Glenmoore, Pa.

GARBER, William E., Jr. (*J* 1938) Sales Mgr., Farquar Heating Service Co., 3406 E. Tenth St., Indianapolis, and (for mail) Rural Route 1, Fairland, Ind.

GARDNER, C. Rollins (*A* 1937) Vice-Pres. (for mail) Martyn Brothers, Inc., 911 Camp St., Dallas, and 4417 E. Lancaster, Ft. Worth, Tex.

GARDNER, William (*A* 1921) Pres., Garden City Fan Co., 332 S. Michigan Ave., and (for mail) 7836 Loomis Blvd., Chicago, Ill.

GARNEAU, Leo (*M* 1938; *J* 1930) Sales Engr. (for mail) C. A. Dunham Co., Ltd., 931 Dominion Square Bldg., Montreal, and 34 Coolbreeze Ave., Lakeside, Que., Canada.

GAULEY, Ernest R. (*A* 1935) Pres. & Gen. Mgr. (for mail) Age Publications, Ltd., 31 Willcocks St., and 156 Dewhurst Blvd., Toronto, Ont., Canada.

GAULT, George W. (*J* 1937; *S* 1934) Corps of Engrs., U. S. A., Ft. Belvoir, Va.

GAUSE, H. Chester (*M* 1937) Power Sales Engr. (for mail) Alabama Power Co., 600 North 18th St., and 905 South 38th St., Birmingham, Ala.

GAUSEWITZ, William H. (*A* 1937) Pres. (for mail) Yale Engineers, Inc., 428 Stinson Blvd., and 1321 W. Minnehaha Pkwy., Minneapolis, Minn.

GAUSMAN, Carl E. (*M* 1923) Partner, Gausman & Moore, 1026 First Natl. Bank Bldg., and (for mail) 2360 Chilcombe Ave., St. Paul, Minn.

GAWTHROP, Fred H. (*M* 1919) Pres., Gawthrop & Bro. Co., 705 Orange St., and (for mail) 2211 Shallcross Ave., Wilmington, Del.

GAYLORD, Frank H. (*M* 1921) Western Sales Mgr. (for mail) Hoffman Specialty Co., Inc., 130 N. Wells St., Chicago, and 362 N. York St., Elmhurst, Ill.

GAYMAN, Paul D. (*M* 1938) Dist. Mgr. (for mail) Johnson Service Co., 2142 East 19th St., Cleveland, and 20875 Endsley, Rocky River, O.

GAYNER, James (*M* 1937) Mech. Engr., G. M. Simonson, Cons. Engr., 74 New Montgomery St., San Francisco, and (for mail) 327 Magnolia Ave., Piedmont, Calif.

GEBEL, Kurt M. (*S* 1940) Sales, Devonshire Artic Chemical Co., 727 Atlantic Ave., Boston, Mass., and (for mail) 229-120th St., Rockaway Beach, L. I., N. Y.

GEE, William W. (*A* 1940; *J* 1938) Engr. & Contractor, William Gee, Jr. & Co., 133 Geary St., San Francisco, and (for mail) 605 Burlingame Ave., Burlingame, Calif.

GEHRS, William (*A* 1939) Br. Mgr. (for mail) Johnson Service Co., 1312 N. W. Raleigh St., and 3801 S. E. Woodward St., Portland, Ore.

GEIGER, Irvin H. (*M* 1919) Registered Prof. Engr. & Mfrs. Repr. (for mail) 319 Telegraph Bldg., and 240 Maclay St., Harrisburg, Pa.

GEIGER, Raymond L. (*M* 1939) Engr., War Dept., Quartermaster Corps, and (for mail) 6501-14th St. N. W., Washington, D. C.

GELTZ, Ralph W. (*J* 1936) Air Cond. Engr., York Ice Machinery Corp., 2700 Washington Ave., and (for mail) 14213 Glenside Rd., Cleveland, O.

GENRE, E. John (*A* 1938) 502 W. Roosevelt, Phoenix, Ariz.

GERMAIN, Oscar (*M* 1935) Htg. Expert, 1343 St. Louis Blvd., Three Rivers, Que., Canada.

GERRISH, Grenville B. (*A* 1936; *J* 1930) Sales Repr., Fitzgibbons Boiler Co., Inc., 31 Main St., Cambridge, and (for mail) 26 Standish Rd., Melrose, Mass.

GERRISH, Harry E. (*M* 1910) (Council, 1919) Partner (for mail) Morgan-Gerrish Co., 307 Essex Bldg., and 4534 Fremont Ave. S., Minneapolis, Minn.

GERSTENBERGER, Edgar J. (*A* 1938) Sales Engr., Glendale Supply Co., 1819 W. Glendale Ave., and (for mail) 3824 North 17th St., Milwaukee, Wis.

GETSCHOW, Roy M. (*M* 1919) Pres. & Treas. (for mail) Phillips-Getschow Co., 32 W. Hubbard St., Chicago, and 122 Woodstock, Kenilworth, Ill.

GHILARDI, Fernand (*M* 1937) Chief Engr., Herdt et Charton, Inc., 34 Rue Godot de Mauroy, Paris, and (for mail) 12 Rue Gabrielle d'Estrees, Vanves (Seine), France.

GHOSE, Khagendra N. (*A* 1938) Consulting Engr., 17 State St., New York, N. Y., and (for mail) 39 Ramkanta Bose St., Bagh Bazar, Calcutta, India.

GHOSH, Bidhu B. (*J* 1939) Air Cond. Engr. (for mail) Messrs. Refrigerators (India) Ltd., 13 C, Russell St., and Galstaun Mansions, Calcutta, British India.

GIANNINI, Mario C. (*M* 1935) Asst. Prof. of Mech. Engrg. (for mail) New York University, University Heights, and 72 Park Terrace W., New York, N. Y.

GIBBONS, Michael J. (*M* 1914) Owner, M. J. Gibbons Supply Co., 601 E. Monument Ave., and (for mail) 22 Oxford Ave., Dayton, O.

GIBBS, Edward W. (*M* 1919) Pres. (for mail) The Smith-Gibbs Co., 201 S. Main St., and 39 President Ave., Providence, R. I.

GIESECKE, Frederick Ernest* (*M* 1913) (*Presidential Member*) (Pres., 1940; 1st Vice-Pres., 1939; 2nd Vice-Pres., 1938; Council, 1932-40) Professor Emeritus, Htg., Vtg. & Air Cond., A. & M. College of Texas, College Station, Tex.

GIFFORD, Clarence A. (*A* 1934) Sales, American Radiator & Standard Sanitary Corp., 1807 Elmwood, Buffalo, and (for mail) 78 Roycroft Blvd., Snyder, N. Y.

GIFFORD, Edmund W. (*M* 1938; *J* 1929) Office Mgr., Himelblau Byfield & Co., 611 N. Broadway, Milwaukee, and (for mail) 7935 Warren Ave., Wauwatosa, Wis.

GIFFORD, Robert L. (*Life Member*; *M* 1908) Pres., Illinois Engineering Co., Cor. 21st St. & Racine Ave., Chicago, Ill., and (for mail) 1231 S. El Molino Ave., Pasadena, Calif.

GIGUERE, George H. (*M* 1920) Mech. Engr., Smith, Hinchman & Grylls, 800 Marquette Bldg., and (for mail) 17205 Fairport, Detroit, Mich.

GILBERT, Leslie S. (*M* 1937) Owner (for mail) Gilbert Engineering Co., 1305 Liberty Bank Bldg., and 3713 Southwestern Blvd., Dallas, Tex.

GILBERT, Thomas (*A* 1940) Asst. Sales Mgr., Empire Brass Manufacturing Co., and (for mail) 40 Alma St., London, Ont., Canada.

GILFRIN, George F. (*M* 1932) Climas Artificiales, S. A. (for mail) Edificio "La Nacional" 902, and Esplanada No. 715 Lomas de Chapultepec, Mexico, D. F.

GILLE, Hadar B. (*M* 1930) Consulting Engr. (for mail) Hugo Theorells Ingeniorsbyra, Skoldungagatan 4, Stockholm, and Svanhildsvagen 19, Nockeby, Sweden.

GILLETT, Merriman C. (*M* 1916) Dist. Sales Mgr., Hoffman Specialty Co., Inc., and (for mail) 6600 Rising Sun Ave., Philadelphia, Pa.

GILLHAM, Walter E. (*M* 1917) (Treas., 1926-29; Council, 1924-29) Consulting Engr. (for mail) 337 Law Bldg., and 3427 Bellefontain, Kansas City, Mo.

GILMAN, Franklin W. (*M* 1935) Plant Engr., Fleetwings, Inc., Bristol, and (for mail) 514 W. Coulter St., Philadelphia, Pa.

GILMORE, John L. (*A* 1938) Owner, John L. Gilmore Htg.-Vtg., 1602 Kay Ave., and (for mail) 1604 Union St., Brunswick, Ga.

GILMORE, Louis A. (*A* 1940; *J* 1935; *S* 1930) Vice-Pres. (for mail) John Gilmore & Co., 115 South 11th St., and 5906 McPherson Ave., St. Louis, Mo.

GINI, Aldo (*M* 1933) via Correggio 18, Milan, Italy

GINN, Tony M. (*M* 1935) General Mgr., Tony M. Ginn Co., 214-24 Fifth St. S., Great Falls, Mont.

GITTERMAN, Henry (*A* 1937) Precipitron Specialist (for mail) Westinghouse Electric & Manufacturing Co., 150 Broadway, New York, and Yorktown Heights, N. Y.

GITTLESON, Harold (*A* 1936) Sales Mgr., Lariviere, Inc., 3715 St. Lawrence Blvd., Montreal, and (for mail) 1125 Lajoie Ave., Outremont, Que., Canada.

GIVIN, Albert W. (*A* 1925) Vice-Pres. (for mail) The Gurney Foundry Co., Ltd., 4 Junction Rd., and 219 St. Clair Ave. W., Toronto, Ont., Canada.

GJERTSEN, George (*S* 1940) Social Security Board, and (for mail) 4509 Arabia Ave., Baltimore, Md.

GLASS, William (*M* 1934) Mgr. (for mail) Partridge-Halliday, Ltd., 144 Lombard St., Winnipeg, and 190 Braemar Ave., Norwood, Man., Canada.

GLEASON, Gilbert H. (*M* 1923) Partner (for mail) Gilbert Howe Gleason & Co., 28 St. Botolph St., Boston, and 10 Edgehill Rd., Winchester, Mass.

GODFREY, Joseph E. (*J* 1938) Engr., Frigidaire Div., General Motors Sales Corp., and (for mail) 1500 Ridgeway Rd., Dayton, O.

GOEHLER, Elmer E. (*A* 1939) Pres. (for mail) Vortex Manufacturing Co., 687 N. Tillamook St., and 2932 N. E. 37th Ave., Portland, Ore.

GOELZ, Arnold H. (*M* 1931) Pres. (for mail) Kroeschell Engineering Co., 215 W. Ontario St., Chicago, and 827 Greenwood Ave., Wilmette, Ill.

GOENAGA, Roger C. (*M* 1931) Tech. Director, Ateliers Ventil (for mail) 109 Cours Gambetta, Lyon, and 33 Avenue Valioud-Ste-Foy-les-Lyon, Rhone, France.

GOERG, Bernard (*M* 1928) American Radiator & Standard Sanitary Corp., 675 Bronx River Rd., Yonkers, N. Y.

GOERGENS, Albert G. (*A* 1938) Asst. Engr., U. S. War Dept., Munitions Bldg., Washington, D. C., and (for mail) 817 Chalfonte Dr., Alexandria, Va.

GOFF, John A. (*M* 1939) Dean (for mail) Towne Scientific School, University of Pennsylvania, Philadelphia, and 511 Cambridge Rd., Bala-Cynwyd, Pa.

GOLDBERG, Moses (*A* 1934) Pres., Electric Motors Corp., 168 Centre St., New York, and (for mail) 885 E. Eighth St., Brooklyn, N. Y.

GOLDMANN, Philipp (*S* 1940) Student, Catholic University, and (for mail) 3536-11th St. N. W., Washington, D. C.

GOLDSMITH, Elliot (*J* 1939) Engr. (for mail) Anemostat Corp. of America, 10 East 39th St., New York, and 102-03 65th Rd., Forest Hills, L. I., N. Y.

GOLDSMITH, F. Willius (*M* 1936) Pres. (for mail) The W. Clasmann Co., 324 E. Wisconsin Ave., and 629 E. Day Ave., Milwaukee, Wis.

GOLL, Willard A. (*A* 1937) Dist. Mgr., Lennox Furnace Co., Marshalltown, Ia., and (for mail) 1457 Washington St., Denver, Colo.

GOMBERS, Henry B. (*Life Member*; *A* 1901) Secy. Emeritus, Heating, Piping and Air Conditioning Contractors National Association, 1256 Sixth Ave., New York, N. Y., and (for mail) 160 Halsted St., East Orange, N. J.

GONZALEZ, Rafael A. (*M* 1936) Mgr., Application Engrg. Dept. (for mail) Airtemp Div., Chrysler Corp., 1119 Leo St., and 2909 Fairmont Ave., Dayton, O.

GOOD, Charles S. (*J* 1941; *S* 1939) A. C. Good & Sons, 620 Rebecca Ave., Pittsburgh (21), Pa.

GOODMAN, Clifford E. (*M* 1940) Mgr. (for mail) Trane Co. of Canada, Ltd., 236 Roy Bldg., and 405 Oxford St., Halifax, N. S., Canada.

GOODRAM, William E. (*M* 1939; *A* 1936) Partner, Goodram Bros., 88 King St. W., Hamilton, and (for mail) R. R. 2, Freeman, Ont., Canada.

GOODRICH, Charles F. (*M* 1919) Andrews & Goodrich, Inc., Boston, and (for mail) 336 Adams St., Dorchester, Mass.

GOODWIN, Eugene W. (*M* 1936) Principal Mech. Engr., Public Bldgs. Administration, Washington, D. C., and (for mail) 7024 Hampden Lane, Bethesda, Md.

GOODWIN, Samuel L. (*M* 1924) Consulting Engr., 1560 Broadway, New York, N. Y., and (for mail) 247 Madison Ave., Hasbrouck Heights, N. J.

GORDON, Colin W. (*A* 1938) Vice-Pres. & Supt., A. G. Baird, Ltd., 286 Lisgar St., and (for mail) 962 Shaw St., Toronto, Ont., Canada.

GORDON, Edward B., Jr. (*Life Member*; *M* 1908) Pres., Pillsbury Engineering Co., 1200 Second Ave. S., and (for mail) 2450 West 24th St., Minneapolis, Minn.

GORDON, Peter B. (*A* 1938; *J* 1935) Treas. (for mail) Wolff & Munier, Inc., 222 East 41st St., New York, N. Y., and 35 Park Ave., Bloomfield, N. J.

GORGEN, Roy E. (*A* 1940) Owner (for mail) Roy E. Gorgen Co., 205 Wesley Temple Bldg., Minneapolis, and 2901 Raleigh Ave., St. Louis Park, Minn.

GORNSTON, Michael H. (*A* 1923) Custodian-Engr. (for mail) Board of Education, Thomas Jefferson High School, Brooklyn, and 90-11-149th St., Jamaica, L. I., N. Y.

GOSS, Matthew H. (*M* 1921) Partner (for mail) M. H. Goss Co., 3409 Ludden St., and 1476 Seyburn Ave., Detroit, Mich.

GOSSETT, Earl J. (*M* 1923) Pres. (for mail) Bell & Gossett Co., 3000 Wallace St., Chicago, and 314 Woodland Ave., Winnetka, Ill.

GOTHARD, William W. (*A* 1936) Editorial Dir. (for mail) Domestic Engineering, 1900 Prairie Ave., Chicago, and 1027 Arlington Ave., La Grange, Ill.

GOTSCHALL, Harry C. (*M* 1935) Instructor in Air Cond., Lane Technical High School, 2501 W. Addison St., and (for mail) 2953 Eastwood Ave., Chicago, Ill.

GOTTWALD, C. (*A* 1916) Pres. (for mail) The Ric-wiL Co., 1563 Union Commerce Bldg., Cleveland, and 2225 Stillman Rd., Cleveland Heights, O.

GOULDING, William (*A* 1933) Air Cond. Engr., World Broadcasting System, Inc., 711 Fifth Ave., New York, and (for mail) 42 Highview Ave., Tuckahoe, N. Y.

GOUNDIE, Joseph K. (*M* 1938) Sales Engr., Fritch Coal Co., 116 River St., Bethlehem, and (for mail) 1426 Walnut St., Allentown, Pa.

GOWDY, Allen C. (*A* 1941; *J* 1939) Htg. Engr., The Huffman-Wolfe Co., 669 N. High St., and (for mail) 1007 Bryden Rd., Columbus, O.

GRABER, Ernst (*J* 1936) Engr., Minneapolis-Honeywell Regulator Co., 801 Second Ave., New York, and (for mail) 222 Hollywood Ave., Douglaston, L. I., N. Y.

GRABMAN, Henry B. (*S* 1938) Student Engr., Crane Co., 836 S. Michigan Ave., and (for mail) 4906 S. Ellis Ave., Chicago, Ill.

GRAHAM, F. D. (*M* 1940) Br. Mgr. (for mail) York Ice Machinery Corp., 222 Union Bldg., and 4204 S. Broad, New Orleans, La.

GRAHAM, J. Barrie (*S* 1939) Student, Carnegie Institute of Technology, Pittsburgh, and (for mail) 193½ Meade St., Wilkinsburg, Pa.

GRAHAM, John J. (*J* 1940) J. J. Graham, Htg., Vtg. & Air Cond., 216 Lippincott Ave., Riverside, N. J.

GRAHAM, John M. (*A* 1937; *J* 1936) Br. Office Mgr. (for mail) B. F. Sturtevant Co., 938 Spitzer Bldg., and 2308 Robinwood, Toledo, O.

GRAHAM, William D. (*M* 1929; *A* 1925; *J* 1923) Mgr., Unit Heater Dept., Carrier Corp., S. Geddes St., and (for mail) 129 Circle Rd., Syracuse, N. Y.

GRANDIA, Willem M. (*A* 1940) In charge Air Cond. and Refrig. Div. (for mail) Pacific Commercial Co., and University Apts., No. 211, Manila, P. I.

GRANKE, Arnold A. (*M* 1939) Sales Engr., Speakman Co., and (for mail) 28 West 41st St., Wilmington, Del.

GRANSTON, Ray O. (*A* 1939; *J* 1935; *S* 1930) Secy. (for mail) University Plumbing & Heating Co., 3939 University Way, and 4333 Ninth Ave. N. E., Seattle, Wash.

GRANT, Walter A. (*A* 1933; *J* 1929) Dist. Chief Engr., Carrier Corp., 12 South 12th St., Philadelphia, and (for mail) 414 Haverford Ave., Narberth, Pa.

GRANT, Walter H., Jr. (*M* 1940) Sales Engr. (for mail) 208-209 Vincent Bldg., and 5632 Elysian Fields Ave., New Orleans, La.

GRAVES, Vernon (*A* 1941; *J* 1940) Engr., Frank A. Leon Co., 901 Girard St. N. E., and (for mail) 1904-17th St. S. E., Washington, D. C.

GRAVES, Willard B. (*Life Member*; *M* 1906) Pres. (for mail) W. B. Graves Heating Co., 162 N. Desplaines St., and 5920 Addison St., Chicago, Ill.

GRAY, Earle W. (*M* 1938; *A* 1934) Div. Commercial Mgr. (for mail) Oklahoma Gas & Electric Co., Third and Harvey Sts., and 2125 Northwest 18th St., Oklahoma City, Okla.

GRAY, Everett W. (*M* 1936) Dist. Repr. (for mail) The Trane Co., 1900 Euclid Ave., Cleveland, and 17545 Madison Ave., Lakewood, O.

GRAY, George A. (*M* 1924) Br. Mgr. (for mail) C. A. Dunham Co., Ltd., 404 Plaza Bldg., and 114 Belmont Ave., Ottawa, Ont., Canada.

GRAY, Hamilton E. (*J* 1940) Secy.-Sales Engr., Gray Engineering Co., Box 264, and (for mail) Box 1223, High Point, N. C.

GRAY, John W. (*M* 1938) The Gray Heating Co., 614 N. Water St., Bay City, Mich.

GRAY, William C. (*J* 1940) Air Cond. Engr., Gulf Mobile & Ohio Railroad, and (for mail) 601 Virginia Ave., Bogalusa, La.

GRAY, William E. (*M* 1922) Pres. (for mail) Gray Engineering Co., Box 264, and 718 W. Farriss, High Point, N. C.

GREEN, Everett W. (*J* 1938) Asst. Mgr. (for mail) Green Furnace & Plumbing Co., 2747 North 48th St., and 5100 Leighton Ave., Lincoln, Nebr.

GREEN, Sydney H. (*J* 1939) Engr. (for mail) Dallas Air Conditioning Co., Inc., 3500 Commerce St., and 2609 Routh St., Dallas, Tex.

GREEN, William C. (*Life Member*; *M* 1906) Dist. Repr. (for mail) Warren Webster & Co., 704 Race St., Rm. 602, and 244 Erkenbrecher Ave., Cincinnati, O.

GREENBURG, Dr. Leonard* (*M* 1932) Executive Dir., Div. of Industrial Hygiene (for mail) N. Y. State Dept. of Labor, 80 Centre St., and 173 West 78th St., New York, N. Y.

GREENLAND, Sidney F. (*M* 1934) Htg. & Vtg. Engr., Gee, Walker & Slater, Ltd., 3 Fitzmaurice Pl., London, W. 1, and (for mail) 20 Cedarville Gardens, Steatham Common, London, S. W. 16, England.

GREGERSON, George (*A* 1941; *J* 1939) Instructor, Hinsdale Township Schools, 317 W. Maple, and (for mail) 242 E. Ogden Ave., Hinsdale, Ill.

GREGG, Stephen L. (*A* 1939; *J* 1936) Sales Engr. (for mail) Potomac Electric Power Co., Tenth and E Sts. N. W., Washington, D. C., and 4828 Edgemoor Lane, Bethesda, Md.

GREINER, George E., Jr. (*J* 1938; *S* 1935) Engr., Wayne Crouse, Inc., 4647 Centre Ave., and (for mail) 5515 Claybourne St., Pittsburgh, Pa.

GRIESS, Philip G. (*M* 1937) Mech. Engr., Voorhees, Walker, Foley & Smith, 101 Park Ave., New York, N. Y., and (for mail) 189 Walnut Ave., Bogota, N. J.

GRIEST, Kermit C. (*A* 1940; *J* 1936) Htg. Engr., Frank-Limbach Co., 1722 E. Ohio St., and (for mail) 437 Bausman St., Pittsburgh, Pa.

GRIEWISCH, Alfred H. (*A* 1938) Pres. (for mail) Bayley Heating Supply Co., 2045 W. St. Paul Ave., and 2557 North 47th St., Milwaukee, Wis.

GRIFFIN, Charles J. (*M* 1940) Engr.-Custodian, Chicago Board of Education, Barnard School, 10354 Charles St., and (for mail) 8231 S. Loomis Blvd., Chicago, Ill.

GRIFFITH, Claude A. (*A* 1938) Contractor, Heating & Air Conditioning Service, 632 Hill Top Dr., Cumberland, Md.

GRIFFITH, Herbert T. (*M* 1938) Designing Engr. (for mail) Lincoln Bouillon, Cons. Engr., 1411 Fourth Ave. Bldg., and 1909 Third Ave. W., Seattle, Wash.

GRIFFITH, Joseph B. (*J* 1938) Sales Engr., Drayer & Hanson, Inc., 738 E. Pico Blvd., Los Angeles, and (for mail) 529 N. Gerona Ave., San Gabriel, Calif.

GRIMES, Fenner M. (*J* 1935) Asst. Engr., War Dept., O. Q. M. G., Constr. Div., Ft. Myer, and (for mail) 849 S. Ivy St., Arlington, Va.

GRITSCHKE, Elmer R. (*M* 1940) Consulting Engr. (for mail) E. R. Gritschke, Cons. Engrs., 123 W. Madison St., Chicago, and 1432 Gregory Ave., Wilmette, Ill.

GRITZAN, L. LeRoy (*M* 1939) Vice-Pres. and Gen. Mgr. (for mail) United Equipment & Supply Co., 1812 M St. N. W., Washington, D. C., and 208 Edgewood Ave., Silver Spring, Md.

GROOT, Harry W. (*M* 1937) Chief Engr., The Home Comfortable, Inc., 912 Baxter Ave., and (for mail) 3728 N. Western Pkwy., Louisville, Ky.

GROSS, Lyman C. (*M* 1931) Sales Engr., Minneapolis-Honeywell Regulator Co., 2727 Fourth Ave. S., and (for mail) 5324 Oaklawn Ave., Linden Hills Station, Minneapolis, Minn.

GROSSENBACHER, Henry E. (*A* 1938) Pres. (for mail) Grossenbacher Furnace Co., Inc., 9410-16 W. Milton Ave., St. Louis, and 9741 Lackland Rd., Overland, St. Louis Co., Mo.

GROSSMAN, F. Arthur (*J* 1938; *S* 1937) Engr., Servel, Inc., and (for mail) 1161 E. Illinois St., Evansville, Ind.

GROSSMANN, Harry A. (*M* 1931) Owner, H. A. Grossmann Co., 3138 Cass Ave., and (for mail) 3122 Geyer Ave., St. Louis, Mo.

GROVES, Samuel A. (*A* 1940; *J* 1935) Sales Supvr., American Radiator & Standard Sanitary Corp., 32-04 Northern Blvd., Long Island City, and (for mail) 36 Cross St., Bronxville, N. Y.

GUEST, Perry L., Jr. (*A* 1939) Pres. (for mail) P. L. Guest Sales Co., 504 Piedmont Bldg., and 716 Dover Rd., Greensboro, N. C.

GUEST, Ross B. (*A* 1940) Estimator & Engr. (for mail) Guest & Viviano Sheet Metal Works, Inc., 827 Dryades St., and 1537 Conery St., New Orleans, La.

GUILBERT, Stanley R. (*A* 1940) Air Cond. Engr., The Riester & Thesmacher Co., 1526 West 25th St., Cleveland, and (for mail) R. F. D. 3, Chagrin Falls, O.

GULER, George D. (*A* 1937) Eastern Modutrol Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 801 Second Ave. at 43rd St., New York, and 6 McBride Ave., White Plains, N. Y.

GUMAER, P. Wilcox (*M* 1937) Industrial Hygiene Engr. (for mail) The Barrett Co., 40 Rector St., New York, N. Y., and 25 Garden St., West Englewood, N. J.

GURNEY, E. Holt (*M* 1929) (*Presidential Member*) (Pres., 1938; 1st Vice-Pres., 1937; 2nd Vice-Pres., 1936; Council, 1931-39) Pres. (for mail) The Gurney Foundry Co., Ltd., 4 Junction Rd., and 347 Walmer Rd., Toronto, Ont., Canada.

GURNEY, Edward R. (*A* 1940; *J* 1937) Asst. to Plant Supt. (for mail) Gurney Foundry Co., Ltd., 4 Junction Rd., and 50 Eastbourne Ave., Toronto, Ont., Canada.

GUSTAFSON, Carl A. (*M* 1938) Sales Engr., (for mail) The Powers Regulator Co., 2720 Greenview Ave., and 6231 N. Fairfield Ave., Chicago, Ill.

GUTKNECHT, Fritz (*M* 1940) Engr. (for mail) Blattmann Weeser Sheet Metal Works, Inc., 1001 Toulouse St., and 5607 Woodlawn Pl., New Orleans, La.

H

HAAS, Samuel L. (*M* 1923) Pres. & Treas. (for mail) Advance Heating & Air Conditioning Corp., 117 N. Desplaines St., and 4300 Lake Shore Dr., Chicago, Ill.

HACH, Edward C. (*M* 1939) Asst. Chief Engr., Sales Dept., Standard Air Conditioning, Inc., Second St. & Beechwood Ave., New Rochelle, N. Y., and (for mail) 261 Seneca Pl., Westfield, N. J.

HACKETT, Frank C. (*A* 1940) Resident Mgr., Bell & Gossett Co., 4855 North 16th St., Arlington, Va.

HADEN, G. Nelson (*M* 1934; *A* 1928; *J* 1922) Chairman and Managing Dir. (for mail) G. N. Haden & Sons, Ltd., 19-29 Woburn Pl., London, W. C. 1, and 36 Wildwood Rd., London, N. W. 11, England.

HADEN, William N. (*Life Member*; *M* 1902) Retired Chairman, G. N. Haden & Sons, Ltd., 19-29 Woburn Pl., London, W. C. 1, and (for mail) Arnolds Hill, Trowbridge, Wilts., England.

HADJISKY, Joseph N. (*M* 1930) Consulting Engr., Htg. & Vtg., 744 Bates St., Birmingham, Mich.

HAERLE, Robert A. (*A* 1938) Design Engr., Bayley Blower Co., 1817 South 66th St., and (for mail) 1438 N. Humboldt Ave., Milwaukee, Wis.

HAGAN, William V. (*M* 1938; *A* 1933; *J* 1926) Secy., V. J. Hagan Co., and (for mail) 1811 Jones St., Sioux City, Ia.

HAGEDON, Charles H. (*M* 1919) Partner (for mail) S. E. Fenstermaker & Co., 937 Archt. & Builders Bldg., and 1107 West 58th St., Indianapolis, Ind.

HAHN, Roy F. (*A* 1941; *J* 1936) Air Cond. Engr. (for mail) Advance Refrigeration, Inc., 350 Peachtree St., and 274 Eighth St. N. E., Atlanta, Ga.

HAINES, John E. (*M* 1940) Mgr., Modutrol Div. (for mail) Minneapolis-Honeywell Regulator Co., and 2119 S. Humboldt Ave., Minneapolis, Minn.

HAINES, John J. (*M* 1915) Pres. (for mail) The Haines Co., 1931 W. Lake St., Chicago, and 623-17th Ave., Maywood, Ill.

HAITMANEK, Louis M. (*A* 1938) Sheet Metal Worker, 217 Rose St., Newark, N. J.

HAJEK, William J. (*M* 1932) Management Consultant (for mail) 372 W. Johnson St., Philadelphia, Pa.

HAKES, Leon M. (*M* 1932; *J* 1929) Resident Repr. (for mail) Warren Webster & Co., 210 Reynolds Arcade Bldg., and 144 Inglewood Dr., Rochester, N. Y.

HALE, Fred J. (*M* 1936) Mgr. (for mail) Empire Sheet Metal Works, Ltd., 1606 W. First Ave., and 3606 Point Grey Rd., Vancouver, B. C., Canada.

HALE, John F. (*Life Member*; *M* 1902) (*Presidential Member*) (Pres., 1913; 1st Vice-Pres., 1912; Board of Governors, 1908-10, 1912-13) (Council, 1914) Dist. Mgr. (for mail) Aerofin Corp., Rm. 544, 111 W. Washington St., Chicago, and 615 W. Elm Ave., La Grange, Ill.

HALEY, Harry S.* (*M* 1914) Consulting Engr. and Partner (for mail) Leland & Haley, 58 Sutter St., and 735-21st Ave., San Francisco, Calif.

HALEY, Robert T. (*A* 1938) Dealer Contact (for mail) Minneapolis Gas Light Co., Marquette at Eighth St., and 5024-12th Ave. S., Minneapolis, Minn.

HALL, Charles J. (*A* 1939) Htg. & Vtg. Engr., (for mail) American Radiator & Standard Sanitary Corp., 40 West 40th St., New York, N. Y., and 102 Dunster Rd., Boston, Mass.

HALL, Cortice H. (*M* 1927) Chief Engr., Stoker Div. (for mail) Fairbanks Morse & Co., and 1004 N. Main St., Three Rivers, Mich.

HALL, George (*A* 1937) Secy.-Treas. (for mail) Hyland, Hall & Co., 218 N. Bassett St., and 4201 Wanetah Trail, Madison, Wis.

HALL, John R. (*M* 1937; *J* 1932) Executive Engr., De Vilbiss Co., and (for mail) 2902 W. Central Ave., Toledo, O.

HALL, Mora S. (*M* 1934) (for mail) Paul Rosenthal & Associates, 600 Newton Pl. N. W., Washington, D. C., and 3504 Hobson St., Brentwood, Md.

HALL, Norman H. (*A* 1939) Supvr.-Engr. (for mail) East Ohio Gas Co., East 62nd St., N. of St. Clair, Cleveland, and 147 Beachview, Willobee, O.

HALL, Robert A. (*J* 1939) Grad. Student, Stanford University, and (for mail) 627 Forest Ave., Palo Alto, Calif.

HALL, Truman (*A* 1940) Sales Engr. (for mail) Dygert Dist. Co., 1 Ionia Ave., Grand Rapids, Mich.

HALLER, Arthur L. (*M* 1920) Pres. (for mail) Haller Appliance Sales Co., Inc., 2007 Olive St., St. Louis, and 7485 Drexel Dr., University City, Mo.

HAMACHER, K. F. (*M* 1938) Partner (for mail) Hamacher & Williams, 2540 W. Wells St., and 4387 S. Austin Ave., Milwaukee, Wis.

HAMIG, Louis L. (*A* 1940; *J* 1935) Engr., John D. Falvey, Cons. Engr., 316 N. Eighth St., and (for mail) 3514 Utah St., St. Louis, Mo.

HAMILTON, Howard S. (*A* 1940) Co-Partner, Air Correction Co., 1214 N. Astor St., Milwaukee, Wis.

HAMLET, F. Aylmer (*A* 1936) Br. Sales Office Mgr. (for mail) C. A. Dunham Co., Ltd., Rm. 931, Dominion Square Bldg., 1010 St. Catherine St. W., and 3550 Shuter St., Montreal, Que., Canada.

HAMLET, Thomas F. (*M* 1938) Sales Engr. (for mail) Taylor-Forbes, Ltd., 6550 Durocher Ave., and 34 Burton Ave., Westmount, Montreal, Que., Canada.

HAMLIN, James B., Jr. (*A* 1937) Htg. Engr., Crane Co., 14 W. Broad St., and (for mail) 1530 East 51st St., Savannah, Ga.

HAMPLE, Henry J. (*S* 1939) Student, Carnegie Institute of Technology, and (for mail) 4921 Forbes St., Pittsburgh, Pa.

HANBURGER, Fred W. (*M* 1930) Consulting Engr., 252 West 76th St., New York, N. Y.

HANLEIN, Joseph H. (*M* 1937) Mech. Engr. (for mail) Wilberding Co., Inc., 1822 Eye St. N. W., and 5420 Connecticut Ave. N. W., Washington, D. C.

HANLEY, Edward V. (*A* 1933) Pres. (for mail) S. V. Hanley Co., 1653 N. Farwell Ave., Milwaukee, and 844 E. Birch Ave., Whitefish Bay, Wis.

HANLEY, Thomas F., Jr. (*M* 1933) Pres. (for mail) Hanley & Co., 1503 S. Michigan Ave., Chicago, Ill.

HANNIGAN, William (*M* 1940) Bldg. Supt., Acacia Mutual Life Insurance Co., 51 Louisiana Ave., Washington, D. C., and (for mail) Route 2, Silver Spring, Md.

HANSLER, John E. (*M* 1937) Field Engr., Delco Appliance Div., General Motors Sales Corp., 391 Lyell Ave., Rochester, N. Y., and (for mail) 104 Nelson Pl., Westfield, N. J.

HANSON, Leon C. (*A* 1918) Htg. & Plbg. Contractor (for mail) Bjorkman Bros. Co., 712 Tenth St. S., and 4713 Townes Rd., Minneapolis, Minn.

HANSON, Leslie P. (*M* 1937; *A* 1936; *J* 1935; *S* 1933) Engr., U. S. Air Conditioning Corp., 2101 Kennedy N. E., and (for mail) 5027 Nokomis Ave. S., Minneapolis, Minn.

HANTHORN, Walter (*J* 1939) Engr. (for mail) Kleenair Furnace Co., 5329 N. E. Sandy Blvd., and 2946 Northeast 54th, Portland, Ore.

HARBERGER, G. L. (*A* 1939) Sales Mgr., Boiler Div., The Eastern Foundry Co., Boyertown, and (for mail) 855 N. Evans St., Pottstown, Pa.

HARBORDT, Otto E. (*A* 1936) Sales Mgr. (for mail) U. S. Supply Co., 1315 West 12th St., and 303 Brush Creek Blvd., Kansas City, Mo.

HARD, Amos L. (*A* 1938) Chief Engr., Thos. Emery Sons & Co., Carew Tower, and (for mail) 910 Kreis Lane, Cincinnati, O.

HARDEN, J. Clinton (*M* 1938) 106 Courtland St., Dowagiac, Mich.

HARDING, Edward R. (*M* 1936) N. C. State Sales Engr. (for mail) Kewanee Boiler Corp., P. O. Box 536, 704 Jefferson Bldg., Greensboro, and Guilford College, N. C.

HARDING, Louis A.* (*M* 1911) (*Presidential Member*) (Pres., 1930; 1st Vice-Pres., 1929; 2nd Vice-Pres., 1928; Council, 1922-31) Commissioner of Public Works, City Hall, and (for mail) 85 Cleveland Ave., Buffalo, N. Y.

HARDY, Frank L. (*A* 1941; *J* 1937) Secy. (for mail) Gulf-York Co., 2300 Third Ave. N., and 2619 Arlington Ave. S., Birmingham, Ala.

HARMONAY, William L. (*A* 1935) Treas. (for mail) M. J. Harmonay, Inc., 124 Elm St., Yonkers, and 3 Brooklands, Bronxville, N. Y.

HARRIGAN, Edward M. (*M* 1915) Gen. Mgr. (for mail) Harrigan & Reid Co., 1365 Bagley Ave., and 7450 LaSalle Blvd., Detroit, Mich.

HARRIGAN, Edward R. (*M* 1939; *J* 1930) (for mail) Harrigan & Reid Co., 1365 Bagley Ave., and 18688 Pennington Dr., Detroit, Mich.

HARRINGTON, Elliott* (*M* 1932; *A* 1930) Sales Mgr., Western Div., Air Cond. & Commercial Refrigeration Dept. (for mail) General Electric Co., Bloomfield, and 17 Wilson Terrace, Caldwell, N. J.

HARRIS, Albert M. (*M* 1938) Vice-Pres. & Gen. Mgr., Baker Ice Machine Co. of Texas, 509 E. Third St., and (for mail) 4415 Meadowbrook Dr., Ft. Worth, Tex.

HARRIS, Jesse B. (*M* 1918) Co-Partner (for mail) Rose & Harris Engineers, 416 Essex Bldg., and 3620 Colfax Ave. S., Minneapolis, Minn.

HARRISON, George G. (*M* 1937) Chief Engr. (for mail) S. T. Johnson Co., 940 Arlington Ave., Oakland, and Durant Hotel, Berkeley, Calif.

HARROWER, William C. (*A* 1937) Air Cond. Engr., Timken Silent Automatic Div., 100-400 Clark Ave., Detroit, and (for mail) 12561 Third Ave., Highland Park, Mich.

HART, F. Donald (*J* 1937) Air Cond. Engr. (for mail) E. I. DuPont de Nemours & Co., Buffalo, N. Y., and 623 Delaware Ave., Wilmington, Del.

HART, Harry M.* (*M* 1912) (*Presidential Member*) (Pres., 1916; 1st Vice-Pres., 1915; Council, 1914-17) Pres. (for mail) L. H. Prentice Co., 1048 Van Buren St., and 3730 Lakeshore Dr., Chicago, Ill.

HART, Stanley (*M* 1938) Vice-Pres., Tuttle & Bailey, Inc., New Britain, Conn.

HART, Theodore S. (*M* 1938) Engr. (for mail) Tuttle & Bailey, Inc., and 530 Lincoln St., New Britain, Conn.

HART-BAKER, Henry W. (*M* 1918) Prop. (for mail) Hart Engineering Co., 392 E. Seward Rd., and P. O. Box 1464, and 530 Embankment Bldgs., Shanghai, China.

HARTIN, William Rhett, Jr. (*J* 1935) Vice-Pres.-Secy., W. R. Hartin & Son, Inc., 2123 Green St., and (for mail) 2744 Trenholm Rd., Columbia, S. C.

HARTMAN, John M. (*M* 1927) Engr. (for mail) Kewanee Boiler Corp., and 618 Elliott St., Kewanee, Ill.

HARTON, A. J. (*A* 1935) Sales Engr., St. Joseph Railway, Light Heat & Power Co., 510 Francis St., and (for mail) 730 E. Hyde Park Ave., St. Joseph, Mo.

HARTSOOK, Granville S., Jr. (*A* 1939) Owner (for mail) G. S. Hartsook, Jr., Plbg. & Htg. Contr., P. O. Box 361, and 112 E. Fourth St., Front Royal, Va.

HENDRICKSON, W. B. (*A* 1940) Service Mgr., Highwood Coal Co., Inc., 5 Sheffield Ave., Englewood, N. J.

HENDRIKSEN, Leonard (*A* 1938) Prop., Hendriksen Sheet Metal & Heating Service, 1919 Vernon Ave., Flint, Mich.

HENION, Hudson D. (*A* 1923) Sales Mgr. (for mail) C. A. Dunham Co., Ltd., 1523 Davenport Rd., and 45 Ridge Dr., Toronto, Ont., Canada.

HENNESSY, William J. (*M* 1938) Personnel Dir., Green Colonial Furnace Co., 322 S. W. Third, and (for mail) 1238-47th St., Des Moines, Ia.

HENRY, Alexander S., Jr. (*M* 1930) 300 Central Park West, New York, N. Y.

HENRY, Ernest C. (*M* 1938) Mgr., E. C. Henry Co., 101 Salzburg Ave., and (for mail) 1115 Park Ave., Bay City, Mich.

HENSZEY, William P. (*A* 1940; *J* 1935) Engr., Carrier Corp., and (for mail) 275 Westwood Rd., Syracuse, N. Y.

HEPBURN, E. M. (*A* 1940) Br. Mgr. (for mail) Empire Brass Manufacturing Co., Ltd., 74 Princess St., and 1045 McMillan Ave., Winnipeg Man., Canada.

HERBERT, James S. (*J* 1940) Sales Engr. (for mail) Blue Ridge Glass Corp., and 177 W. Sevier St., Kingsport, Tenn.

HERBERT, Richard M. (*J* 1938) Engr. (for mail) Herbert Refrigeration, 311 E. Sixth, and 609 Baltimore, Waterloo, Ia.

HERING, Alfred (*M* 1935) Pres., Hering Heating Co., Inc., 203 East 88th St., and (for mail) 1830 Tenbroeck Ave., New York, N. Y.

HERLIHY, Jeremiah J. (*Life Member*; *M* 1914) 3751 Eddy St., Chicago, Ill.

HERMAN, Neil B. (*J* 1937; *S* 1936) Engr. (for mail) Riggs Distler & Co., Inc., 516 Fifth Ave., New York, N. Y., and 4217 Garfield Ave. S., Minneapolis, Minn.

HERO, George A., Jr. (*M* 1940) Partner, Airflow Co., 815 Baronne St., New Orleans, and (for mail) P. O. Box 84, Gretna, and Fort St. Leon, Plazuemines, Parish, La.

HERRE, Harold A. (*S* 1940) Apprentice in Plbg., Htg. & Vtg., Herre Bros., 7th & Emerald St., and (for mail) 205 Montrose St., Harrisburg, Pa.

HERRING, Edgar (*Life Member*; *M* 1919) Chairman and Governing Dir. (for mail) J. Jeffreys & Co., St. George's House, 195-203 Waterloo Rd., London, S. E. 1, and "Kenia," Keswick Rd., Putney, London, S. W., England.

HERSH, Franklin C. (*M* 1939; *A* 1933; *J* 1930) Specialty Engr., Pennsylvania Power & Light Co., 901 Hamilton St., and (for mail) 317 South 16th St., Allentown, Pa.

HERSHEY, Albert E.* (*M* 1940) Research Asst. Prof. (for mail) University of Illinois, and 408 E. Washington St., Urbana, Ill.

HERSKE, Arthur R. (*M* 1926) Pres., Herske & Timmis, Inc., 11 West 42nd St., New York, and (for mail) 101 Brookfield Rd., Mt. Vernon, N. Y.

HERTY, Frank B. (*M* 1933) The Gas Engine & Electric Co., Inc., 280 Meeting St., and (for mail) 294 Congress St., Charleston, S. C.

HERTZLER, John R.* (*M* 1936; *J* 1928) Gen. Sales Mgr. (for mail) York Ice Machinery Corp., and 863 S. George St., York, Pa.

HESS, Arthur J. (*M* 1937) Engr. (for mail) English & Lauer, Inc., 1978 S. Los Angeles St., and 2616 West 70th St., Los Angeles, Calif.

HESSELSCHWERDT, August L., Jr. (*M* 1940; *J* 1937) Asst. Prof. of Mech. Engrg., Wayne University, 4841 Cass Ave., and (for mail) 15722 Kentucky Ave., Detroit, Mich.

HESSLER, Lester W. (*M* 1936) Mgr., The Trane Co., 1835 N. Third St., and (for mail) 6034 N. Bayridge Ave., Milwaukee, Wis.

HESTER, Thomas J. (*M* 1919) Vice-Pres.-Treas. (for mail) Hester Bradley Co., 2835 Washington and 67 Aberdeen Pl., St. Louis, Mo.

HEWETT, John B. (*M* 1937; *A* 1935) Sales Mgr., Anemostat Corp. of America, 10 East 39th St., New York, and (for mail) Sussex Hall, Dobbs Ferry, N. Y.

HEYDON, Charles G. (*A* 1923) Mgr. Sales Western Div., Wright Austin Co., 315 Woodbridge St. W., and (for mail) 2681 Nebraska Blvd., Detroit, Mich.

HIBBS, Frank C. (*M* 1917) Htg. Engr., The H. B. Smith Co., Inc., 2209 Chestnut St., and (for mail) 846 North 65th St., Philadelphia, Pa.

HICKEY, Daniel W. (*A* 1931) Pres. (for mail) D. W. Hickey & Co., Inc., 1841 University Ave., and 1874 Highland Pkwy., St. Paul, Minn.

HICKMAN, Herbert V. (*A* 1938) Sales Engr. (for mail) Anemostat Corp. of America, 1129 Folsom St., San Francisco, and 433 San Diego Ave., Daly City, Calif.

HIERS, Charles R. (*M* 1929; *J* 1927) Sales Engr., Minneapolis-Honeywell Regulator Co., 801 Second Ave., New York, and (for mail) 19 Westminster Rd., Great Neck, L. I., N. Y.

HIGH, John M. (*M* 1940; *A* 1938) Mgr., Insulation Div. (for mail) The Ruberoid Co., 500 Fifth Ave., New York, N. Y., and 6115 Nassau Rd., Philadelphia, Pa.

HILDER, Frederick L. (*M* 1937) Chief Engr., Electric Furnace-Man, Inc., 780 East 138th St., New York, N. Y., and (for mail) 162 Trenton Ave., Clifton, N. J.

HILDRETH, Egbert S. (*A* 1936) Air Cond. Specialist, Indianapolis Power & Light Co., 17 N. Meridian St., Indianapolis, Ind.

HILL, Charles F. (*J* 1936) Carrier Air Cond. Dept. Mgr., United Engineers, Ltd., River Valley Rd., Singapore, Straits Settlements.

HILL, Edward, Jr. (*J* 1939) Warm Air Sales Engr., American Radiator & Standard Sanitary Corp., Fourth & Townsend Sts., and (for mail) 3355 Octavia St., San Francisco, Calif.

HILL, Dr. E. Vernon* (*M* 1914; *A* 1912) (*Presidential Member*) (Pres., 1920; 1st Vice-Pres., 1919; 2nd Vice-Pres., 1918; Council, 1915-21) 6826 Highland Ave., Chicago, Ill.

HILL, Fred M. (*M* 1930) Engr., 2104 S. Vermont Ave., and (for mail) 225 East Avenue 39, Los Angeles, Calif.

HILL, Harold G. (*J* 1938) Mgr. Furnace & Air Cond. Div., Gurney Foundry Co., Ltd., 4 Junction Rd., and (for mail) 7 Armadale Ave., Toronto, Ont., Canada.

HILL, Harold H. (*M* 1935) Dist. Mgr. (for mail) American Blower Corp., 1211 Commercial Bank Bldg., and 2303 Vail Ave., Charlotte, N. C.

HILL, Jared A. (*M* 1938) Htg. & Air Cond. Engr., Pacific Gas & Electric Co., 245 Market St., San Francisco, and (for mail) 715 Laurel Ave., Burlingame, Calif.

HILL, Vaughn H. (*J* 1938) Htg. Engr., Motor Wheel Corp., and (for mail) 2111 Colvin Court, Lansing, Mich.

HILL, Walter W. (*J* 1940) Engr., United Clay Products Co., 931 Investment Bldg., and (for mail) 3415 Fessenden St. N. W., Washington, D. C.

HILLS, Arthur H. (*M* 1924) Mgr. (for mail) Sarco Canada, Ltd., 85 Richmond St. W., and 100 Nealon Ave., Toronto, Ont., Canada.

HINES, Guy M. (*A* 1937) Chief Engr., Texas Agricultural & Mechanical College, Power Dept., and (for mail) P. O. Box 248, College Station, Tex.

HINES, John C. (*M* 1937) Pres. (for mail) Hines Engineering Co., 4503 Lincoln Ave., Chicago, and 6629 Ramona Ave., Lincolnwood, Ill.

HINNANT, Clarence H., Jr. (*J* 1938) Asst. Air Cond. Engr., American Viscose Corp., Wilmington, and (for mail) 602 Claymont Apt., Claymont, Del.

HINRICHSEN, Arthur F. (*M* 1928) Pres.-Treas. (for mail) A. F. Hinrichsen, Inc., 50 Church St., Rm. 1859, New York, N. Y., and Mountain Lakes, N. J.

HIRSCH, Martin H. (*M* 1938) The Trane Co., 250 East 43rd St., New York, and (for mail) 65-74 Saunders St., Forest Hills, L. I., N. Y.

HIRSCHMAN, William F. (*M* 1929) Pres. and Chief Engr., W. F. Hirschman Co., Inc., 1245 McKinley Parkway, and (for mail) 165 Le Brun Circle, Buffalo, N. Y.

HITCHCOCK, Paul C. (*M* 1931) Pres. (for mail) Hitchcock & Estabrook, Inc., 521 Sexton Bldg., and 5130 Harriet Ave., Minneapolis, Minn.

HITT, John C. (*A* 1936) Mgr. (for mail) Holland Furnace Co., 55-18th St., and Howard Pl., Wheeling, W. Va.

HOBBIE, Edward H. (*A* 1937) Mgr., Sales Promotion and Research (for mail) Mississippi Glass Co., 220 Fifth Ave., New York, N. Y., and Ridgedale Ave., Florham Park, N. J.

HOBBS, J. Clarence (*M* 1920) Vice-Pres. (for mail) Diamond Alkali Co., and 60 Wood St., Painesville, O.

HOBBS, William S. (*A* 1936) (for mail) P. O. Box 269, and 327 Park Ave., Swarthmore, Pa.

HOCHMAN, Eugene (*J* 1940; *S* 1938) Jr. Mech. Engr., U. S. Navy Yard, Boston, and (for mail) 286 Chestnut St., Chelsea, Mass.

HOCKENSMITH, Francis E. (*M* 1936) Chief Engr. (for mail) Lennox Furnace Co., Inc., 400 N. Midler Ave., and 454 Plymouth Dr., Syracuse, N. Y.

HODGE, William B. (*M* 1934) Vice-Pres. (for mail) Parks-Cramer Co., P. O. Box 23, and 2600 Roswell Ave., Charlotte, N. C.

HOEY, James K. (*A* 1938) Pres.-Mgr. (for mail) Crater Metal & Engineering, Inc., 142 N. Front St., and 119 Lincoln St., Medford, Ore.

HOFFMAN, Charles S. (*M* 1924) Pres. (for mail) Baker Smith & Co., Inc., 576 Greenwich St., and 108 East 38th St., New York, N. Y.

HOFFMAN, Harry (*M* 1939) Mgr. (for mail) Johnson Service Co., 105 Piedmont Bldg., Greensboro, and Route 1, Guilford College, N. C.

HOFFMANN, Angelo (*A* 1938) Vice-Pres. (for mail) Louis Hoffmann Co., 117 W. Pittsburgh Ave., and 4850 N. Oakland Ave., Milwaukee, Wis.

HOGAN, Edward L.* (*M* 1911) General Consulting Engr. (for mail) American Blower Corp., 6000 Russell St., and 700 Seward St., Detroit, Mich.

HOGUE, William M. (*A* 1935) Sales Engr. (for mail) U. S. Electrical Motors, Inc., 200 E. Slauson Ave., and 4839 Keniston Ave., Los Angeles, Calif.

HOLLAND, George R. (*J* 1941; *S* 1938) Student Engr., Abbott Lester & Co., Inc., 140 Cedar St., New York, and (for mail) 35-07-90th St. Jackson Heights, L. I., N. Y.

HOLLAND, Robert B. (*M* 1938) Sales Engr. (for mail) York Ice Machinery Corp., 1275 Folsom St., and 3820 Scott St., San Francisco, Calif.

HOLLAND, William T. (*A* 1940) Engr., Heating, Piping & Air Conditioning Contractors, Milwaukee Assn., and (for mail) 1820-75th St., Milwaukee, Wis.

HOLLISTER, Norman A.* (*M* 1933) 7101 Colonial Rd., Brooklyn, N. Y.

HOLMES, Arthur D. (*M* 1935) Vice-Pres. (for mail) Plumber Supply Co., 323 W. First, and 1848 East 18th St., Tulsa, Okla.

HOLMES, Paul B. (*A* 1936) Br. Mgr. (for mail) The National Radiator Co., 600 W St. N. E., and 4525 Fessenden St. N. W., Washington, D. C.

HOLMES, Richard E. (*A* 1938; *J* 1934) Air Cond. Design Engr., Westinghouse Electric & Manufacturing Co., 653 Page Blvd., and (for mail) 188 Bristol St., Springfield, Mass.

HOLSWORTH, Robert C. (*M* 1940) Consulting Mech. Engr.-Pres., Holsworth Equipment Co., P. O. Box 1981, Corpus Christi, Tex.

HOLT, James (*M* 1933) Assoc. Prof., Mech. Engrg. (for mail) Massachusetts Institute of Technology, Cambridge, and 1062 Massachusetts Ave., Lexington, Mass.

HOLT, Walter H. (*J* 1938) Sales Engr., Buffalo Forge Co., 512 Woodward Bldg., Washington, D. C.

HOLUBA, Henry Julian (*J* 1938) Sales Engr. (for mail) A-J Manufacturing Co., 2119 Washington St., and 505 West 31st St., Kansas City, Mo.

HOLZER, Rudolph J., Jr. (*J* 1940) Engr. & Estimator, Holzer Sheet Metal Works, 317 Burgundy St., and (for mail) 3738 Octavia St., New Orleans, La.

HONERKAMP, Fritz (*M* 1937) Chief Engr. (for mail) Anemostat Corp. of America, 10 East 39th St., New York, and 6712-50th Ave., Woodside, L. I., N. Y.

HOOK, Frank W. (*M* 1938) Br. Mgr. (for mail) Johnson Service Co., 814 Rialto Bldg., and 2444 Larkin St., San Francisco, Calif.

HOOVER, William L. (*J* 1940) Sales Engr., Clarage Fan Co., 723 Albee Bldg., and (for mail) 2242 Cathedral Ave., Washington, D. C.

HOPPE, Marcel F. (*M* 1938) Consulting Engr. (for mail) 1621 Connecticut Ave. N. W., Washington, D. C., and P. O. Box 531, Falls Church, Va.

HOPPER, Garnet H. (*M* 1923) Engr., Taylor-Forbes, Ltd., 1088 King St., W., and (for mail) 19 Brummel Ave., Toronto, Ont., Canada.

HOPPER, John S. (*M* 1938) Asst. Prof. Mech. Engrg. (for mail) A. & M. College of Texas, and 305 Kerry St., College Park, College Station, Tex.

HOPSON, William T. (*Life Member*; *M* 1915) Pres. (for mail) Hopson & Chapin Mfg. Co., 231 State St., and 217 Ocean Ave., New London, Conn.

HORCH, George E. (*M* 1939) Mech. Engr. (for mail) Wm. B. Ittner, Inc., Arch. & Engrs., 911 Locust St., St. Louis, and 436 Park Ave., Webster Groves, Mo.

HORNE, Herbert F. (*A* 1940) Sales Engr., Johns-Manville Co., 826 Woodward Bldg., Washington, D. C., and (for mail) 1721 Queens Lane, Arlington, Va.

HORNUNG, John C. (*Life Member*; *M* 1914) Retired—E. E., 854 Bluff St., Glencoe, Ill.

HOSHALL, Robert H. (*M* 1930) Associate (for mail) Thos. H. Allen, Cons. Engr., 65 McCall Pl., and 1844 Cowden Ave., Memphis, Tenn.

HOSKING, Homer L. (*M* 1930) Mgr. N. Y. Sales Office (for mail) Pacific Steel Boiler Div., 101 Park Ave., New York, and 208 Madison Rd., Scarsdale, N. Y.

HOSTERMAN, Charles O. (*M* 1924) Supt., The McMurrer Co., 303 Congress St., Boston, and (for mail) 25 Bateswell Rd., Dorchester, Mass.

HOTCHKISS, Charles H. B. (*M* 1927) Vice-Pres.-Director, Ameresco, Inc., 50 Church St., New York, N. Y.

HOUGHTEN, Ferry C.* (*M* 1921) Dir. (for mail) ASHVE Research Laboratory, U. S. Bureau of Mines, 4800 Forbes St., and 1136 Murray Hill Ave., Pittsburgh, Pa.

HOULIS, Louis D. (*M* 1935) Chief Engr., Master Baker Ovens, and (for mail) 836 Pedretti Rd., West Price Hill, Cincinnati, O.

HOULISTON, G. Baillie (*A* 1928) Secy. (for mail) The W. C. Green Co., 704 Race St., Cincinnati, O., and 33 Tremont Ave., Ft. Thomas, Ky.

HOWARD, Fenton L. (*M* 1937) Chief Engr. (for mail) Industrial Training Institute, Inc., 2154 Lawrence Ave., and 6619 N. Rockwell St., Chicago, Ill.

HOWATT, John* (*M* 1915) (*Presidential Member*) (Pres., 1935; 1st Vice-Pres., 1934; 2nd Vice-Pres., 1933; Council, 1927-36) Chief Engr. (for mail) Board of Education, 228 N. LaSalle St., Chicago, Ill.

HOWE, Willis W. (*M* 1936; *A* 1917) Sales Engr., Pacific Gas & Electric Co., San Rafael, and (for mail) 68 Central Ave., Sausalito, Calif.

HOWELL, Lloyd (*M* 1915) Engr., Industrial Dept. (for mail) Peoples Gas Light & Coke Co., 122 S. Michigan Ave., and 7605 Yates Ave., Chicago, Ill.

HOYT, Leroy W. (*M* 1930) Supvr., Plbg. & Htg., Dept. Pub. Works, State of Connecticut, Hartford, and (for mail) 45 Cumberland Rd., West Hartford, Conn.

HUBBARD, George W.* (*M* 1911) Consulting Engr. (for mail) 1406 Railway Exchange, Chicago, and 710 Bonnie Brae, River Forest, Ill.

HUBBARD, Nelson B. (*M* 1919) Engr., Hubbard & Wagschal, Engrs., 243 Congress St. W., and (for mail) 2985 Blaine Ave., Detroit, Mich.

HUBBUCH, Nicholas J., Jr. (*S* 1939) Student, University of Dayton, Dayton, O., and (for mail) 510 Breckenridge Lane, Louisville, Ky.

HUBER, Enrique (*M* 1938) Mech. Engr., Calle Marsella Num. 66, Mexico, D. F., Mexico.

HUCH, A. J. (*M* 1919) Br. Mgr. (for mail) American Radiator & Standard Sanitary Corp., 312 S. Third St., and 4037 Harriet Ave., Minneapolis, Minn.

HUCKER, Joseph H. (*M* 1921) Partner (for mail) Hucker-Pryibil Co., 1700 Walnut St., Philadelphia, and 715 Stanbridge St., Norristown, Pa.

HUDEPOHL, Louis F. (*M* 1936) Pres. (for mail) T. J. Conner, Inc., 3290 Spring Grove Ave., and 4395 Haight Ave., Cincinnati, O.

HUDSON, Robert A. (*M* 1934) Partner, Hunter & Hudson, Engrs., 41 Sutter St., San Francisco, and (for mail) Route 2, Box 51, Cordilleras Rd., Redwood City, Calif.

HUGGINS, L. Gale (*M* 1939) Asst. Mgr., Air Cond. Dept., Westinghouse Electric & Manufacturing Co., and (for mail) 176 Springfield St., Springfield, Mass.

HUGHES, L. King (*A* 1940; *J* 1936) Vice-Pres. in charge of Sales, Howard Air Conditioning, Ltd., 881 Yonge St., and (for mail) 43 Rivercourt Blvd., Toronto, Ont., Canada.

HUGHES, Samuel (*J* 1940) Sales, Minneapolis Gas Light Co., Minneapolis, and (for mail) Route No. 1, Wayzata, Minn.

HUGHES, William U. (*M* 1936) Pres. (for mail) The Lewis-Brown Co., Ltd., 1411 Crescent St., and 1610 Sherbrooke St. W., Montreal, Que., Canada.

HUGHEY, Thomas M. (*A* 1935) Dist. Mgr. (for mail) Westerlin & Campbell Co., 906 N. Fourth St., and 2527 North 63rd St., Milwaukee, Wis.

HUGHSON, Harry H. (*M* 1927) Sales Engr., The Coon-DeVisser Co., 2051 W. Lafayette Blvd., Detroit, Mich.

HULL, Harry B. (*M* 1931) Research Engr., Frigidaire Div., General Motors Sales Corp., and (for mail) P. O. Box 671, Dayton, O.

HUMMEL, George W. (*M* 1937) Engr. (for mail) The Trane Co., 1424 N. Third St., and Drawer 679, Phoenix, Ariz.

HUMPHREY, Dwight E. (*M* 1921) Htg. & Vtg. Engr., Goodyear Tire & Rubber Co., Akron, and (for mail) 2499 Six St., Cuyahoga Falls, O.

HUMPHREY, Leonard G., Jr. (*J* 1938) Sales Engr. (for mail) Buffalo Forge Co., 640 Woodward Bldg., Washington, D. C., and 4525 Rosedale Ave., Bethesda, Md.

HUMPHREYS, Clark M. (*M* 1931) (Council, 1936-38) Asst. Prof., Mech. Engrg. (for mail) Carnegie Institute of Technology, Schenley Park, and 1934 Remington Dr., Pittsburgh, Pa.

HUNGER, Robert F. (*M* 1927) Assoc. Dist. Mgr. (for mail) Buffalo Forge Co., 220 South 16th St., Philadelphia, and 100 Avon Rd., Narberth, Pa.

HUNGERFORD, Leo (*M* 1930) Sales Mgr., Utility Fan Corp., 48th St. & Alameda St., and (for mail) 828 N. Stanley Ave., Los Angeles, Calif.

HUNT, MacDonald (*A* 1936) Mfrs. Agent (for mail) McDonnell & Miller Co., 12 W. Madison St., and Windsor Court Apts., Baltimore, Md.

HUNTER, Louis N. (*M* 1936) Mgr. of Research (for mail) National Radiator Co., 221 Central Ave., Johnstown, Pa.

HUNZIKER, Chester E. (*A* 1934) Br. Mgr. (for mail) American Blower Corp., 511 State St., Schenectady, and 422 Reynolds St., Scotia, N. Y.

HURWICH, Sidney B. (*A* 1939) Engr. (for mail) American Lubricants, Inc., 1575 Clinton St., and 177 Wyoming Ave., Buffalo, N. Y.

HUST, Carl E. (*M* 1932) Htg. & Air Cond. Engr. (for mail) Cincinnati Gas & Electric Co., Fourth & Main Sts., and Hillcrest Apts., 15 Mason St., Cincinnati, O.

HUSTOEL, Arnold M. (*A* 1930) 2414 N. Kedzie Blvd., Chicago, Ill.

HUTCHEON, Clifford R. (*J* 1938) Chief Engr. (for mail) Plandaire, Inc., P. O. Box 7350, Oakland Station, and 3223 Kennett Square, Pittsburgh, Pa.

HUTCHINS, William H. (*M* 1934) Chief Engr., Delco Appliance Div., General Motors Corp., 391 Lyell Ave., and (for mail) 600 Seneca Pkwy., Rochester, N. Y.

HUTCHINSON, B. Lee, Jr. (*J* 1939) Sales Engr. (for mail) York Ice Machinery Corp., 659 E. Sixth St., and 1945 Madison Rd., Cincinnati, O.

HUTCHINSON, Frederick A. (*J* 1939) Mgr., Repair Dept., C. J. Doughty & Co., 30 Brenan Rd., and (for mail) 31 Lucerne Rd., Shanghai, China.

HUTCHISON, John E. (*M* 1921) Partner (for mail) Moody & Hutchison, 1701 Architects Bldg., and 6723 Emlen St., Philadelphia, Pa.

HUTZEL, Hugo F. (*M* 1918) Mgr. Tech. Training Dept., Div. of Nash Kelvinator Corp., and (for mail) 2635 Woodstock Dr., Detroit, Mich.

HUYBERT, Leslie E., Jr. (*J* 1938) Draftsman, U. S. Naval Operating Base, and (for mail) 604 Carolina Ave., Norfolk, Va.

HVOSLEF, Fredrik W. (*M* 1931; *A* 1921) Htg. Research Engr. (for mail) Kohler Co., and 523 Audubon Rd., Kohler, Wis.

HYDE, Elmer H. (*A* 1937) Tech. Repr., Koppers Co., Tar & Chemical Div., Koppers Bldg., and (for mail) 442 Sulgrave Rd., Chatham Village, Pittsburgh, Pa.

HYDE, Elver V. (*J* 1939) Sales, Bruce Publishing Co., 2642 University Ave., St. Paul, Minn.

HYDE, Eric F. (*M* 1937) Mech. Engr., Giffels & Vallet, Inc., 1000 Marquette Bldg., Detroit, and (for mail) 708 Oakland Ave., Birmingham, Mich.

HYDE, L. Lyman (*M* 1940; *J* 1934) Mgr., Grudem Co., 1939 University Ave., St. Paul, and (for mail) 4089 Hubbard Ave., Robbinsdale, Minn.

HYMAN, Wallace M. (*M* 1920) Pres. (for mail) Reis & O'Donovan, Inc., 12 West 21st St., and 23 West 73rd St., New York, N. Y.

HYNES, Lee P. (*M* 1919) Pres.-Chief Engr. (for mail) Hynes Electric Heating Co., 240 Cherry St., Philadelphia, Pa., and 127 West End Ave., Haddonfield, N. J.

I

IBISON, James L. (*J* 1938) 1st Lt., Asst. Constructing Quartermaster, U. S. Army, Camp Lee, Petersburg, Va.

ICKERINGILL, John C. (*M* 1923) Sales Engr., Spencer Heater Co., 31st & Walnut St., Philadelphia, and (for mail) 477 Flamingo St., Roxborough, Philadelphia, Pa.

ILLIG, Ernest E. (*J* 1938) Sales Engr., Walter R. Illig, 1 Cushing St., and (for mail) 185 Blossom St., Fitchburg, Mass.

ILLIG, Walter R. (*M* 1935; *A* 1927) Owner, Walter R. Illig, 1 Cushing St., and (for mail) 185 Blossom St., Fitchburg, Mass.

INGALLS, Frederick D. B. (*M* 1906) Consulting Htg. & Air Cond. Engr., 1 Hopkins St., Reading, Mass.

INGELS, Margaret* (*M* 1923; *J* 1918) Engrg. Editor (for mail) Carrier Corp., So. Geddes St., and 412 University Pl., Syracuse, N. Y.

INMAN, C. M. (*M* 1940) Supt. of Engrs. (for mail) Cafritz Co., 1404 K St. N. W., and 1445 Ogden St. N. W., Washington, D. C.

ISAACS, Harry A., Jr. (*A* 1939) Pres. (for mail) Oil Delivery, Inc., 105 Monmouth St., Red Bank, and Little Silver, N. J.

ISETT, William M. (*A* 1936) Pres. (for mail) C. B. Isett & Son, Inc., 3035 N. Rockwell St., and 5834 N. Christiana Ave., Chicago, Ill.

d'ISSERTELLE, Henry G.* (*Life Member*; *M* 1913; *A* 1912) Consulting Engr., 31 Park Terrace W., Apt. A-8, New York, N. Y.

IVERSON, Henry R. (*M* 1936) Co-Mgr. (for mail) The Trane Co., 1772 Columbia Rd. N. W., and 1601 Argonne Pl. N. W., Washington, D. C.

IVERSTROM, Carl (*A* 1940; *J* 1939) Project Engr., W. P. A., 260 East 161st St., New York, and (for mail) 35-24 72nd St., Jackson Heights, L. I., N. Y.

J

JACKES, Herman D. (*M* 1915) Sales Engr., Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, N. Y., and (for mail) 185 Liberty St., Bloomfield, N. J.

JACKSON, Charles H. (*M* 1923) Vice-Pres. (for mail) Blower Application Co., 918 N. Fourth St., and 2706 N. Farwell, Milwaukee, Wis.

JACKSON, Gilbert R. (*M* 1938) Mgr., Boiler & Radiator Dept., Crane, Ltd., 45-51 Leman St., London, E. 1, England.

JACKSON, Marshall S. (*M* 1919) Mfrs. Repr. (for mail) 250 Delaware Ave., and 108 Larchmont Rd., Buffalo, N. Y.

JACKSON, Walter F. (*J* 1939) Lab. Engr. (for mail) American Stove Co., 1200 Long Ave., Lorain, and 209 Cascade St., Elyria, O.

JACOBI, Bruce A. (*J* 1939; *S* 1936) Asst. Chief Engr., Johnson & Morris, Inc., 233 West 18th St., New York, and (for mail) 1731 East 17th St., Brooklyn, N. Y.

JACOBSEN, K. C. S. (*A* 1939) Office Mgr. (for mail) Imperial Electric Co., 40th & Chestnut Sts., Philadelphia, and 17 Aldwyn Lane, Villanova, Pa.

JACOBUS, David S. (*Life Member*; *M* 1916) Advisory Engr. (for mail) The Babcock & Wilcox Co., 85 Liberty St., New York, N. Y., and 93 Harrison Ave., Montclair, N. J.

JAKOBY, Albert C. (*A* 1938) Estimator, Sears Roebuck & Co., 4040 Roosevelt Blvd., and (for mail) 1913 E. Clearfield St., Philadelphia, Pa.

JALONACK, Irwin G. (*M* 1940; *A* 1933; *S* 1930) Chief Engr. (for mail) Alfred L. Hart, Inc., 164-07 Hillside Ave., Jamaica, and 44 Carol St., Hempstead Lake Park, Hempstead, L. I., N. Y.

JAMES, Hamilton R. (*M* 1931) Service Equip. Engr., United Engineers & Constructors, Inc., 1401 Arch St., Philadelphia, and (for mail) 55 W. Drexel Ave., Lansdowne, Pa.

JAMES, John W.* (*M* 1937; *J* 1933) Technical Secy., American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York, N. Y.

JANET, Harry L. (*M* 1920) Engr., Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, and (for mail) 688 Decatur St., Brooklyn, N. Y.

JANOS, William A. (*M* 1940) Mech. Engr. (for mail) Western Electric Co., 195 Broadway, New York, and 21-11 23rd Terrace, Astoria, L. I., N. Y.

JARCHO, Martin D. (*A* 1939; *J* 1936) Vice-Pres. (for mail) Jarcho Bros. Inc., 304 East 45th St., New York, and 941 Washington Ave., Brooklyn, N. Y.

JARDINE, Douglas C. (*M* 1929; *A* 1920) Pres. (for mail) Jardine-Wardman, Inc., 516 S. Tejon St., and 1216 N. Cascade Ave., Colorado Springs, Colo.

JARDINE, William H., Jr. (*A* 1938) Pres. (for mail) Iona Ventilator Co., Inc., 2821-29 W. Dauphin St., and 3552 Shelmire St., Philadelphia, Pa.

JEHLE, Ferdinand (*M* 1938; *A* 1937) Dir. of Research (for mail) Hoffman Specialty Co., Inc., 575 Pacific St., Stamford, and 7 Colonial Court, New Canaan, Conn.

JELINEK, Frank R. (*A* 1941; *J* 1937) Br. Mgr. (for mail) Johnson Service Co., 2505 Commerce St., and 1512 N. Fitzhugh, Dallas, Tex.

JENKINS, Sydney D. (*M* 1940) Mgr. Plbg. & Htg. Dept., Bell & Morris, and (for mail) 2212-33rd Ave. S. W., Calgary, Alta., Canada.

JENKINSON, V. Jack (*A* 1940) Sales Engr. (for mail) Minneapolis-Honeywell Regulator Co., Ltd., 117 Peter St., and 1135 Gerrard St. E., Toronto, Ont., Canada.

JENNEY, Hugh B. (*A* 1933) Sales Mgr. (for mail) Standard Sanitary & Dominion Radiator Co., Ltd., Cor. Royce & Lansdowne Aves., and 96 Dawlish Ave., Toronto, Ont., Canada.

JENNINGS, Irving C. (*M* 1924) Pres. (for mail) The Nash Engineering Co., and 138 Flax Hill Rd., South Norwalk, Conn.

JENNINGS, Stanley A. (*M* 1935) Asst. Chief Engr., Trane Co. of Canada, Ltd., 4 Mowat Ave., and (for mail) 17 Avion Ave., Toronto, Ont., Canada.

JENNINGS, Warren G. (*A* 1930) Vice-Pres. (for mail) Minneapolis-Honeywell Regulator Co., 797 Beacon St., Boston, and 20 Chapel St., Brookline, Mass.

JENNINS, Henry H. (*Life Member*; *M* 1901) 29 Greyshiels Ave., Leeds, 6, England.

JESSUP, Benjamin H. (*M* 1937) Pres. (for mail) Richards & Jessup Co., Inc., 615 Main St., Stamford, and 48 Field St., Glenbrook, Conn.

JOHNS, Charles F. (see Special Service Roll, p. 73).

JOHNS, Harold B.* (*M* 1928; *J* 1927) (for mail) Peoples Gas Light & Coke Co., 122 S. Michigan Ave., Chicago, and 543 N. Elmwood Ave., Oak Park, Ill.

JOHNSON, Allen J.* (*M* 1935) Dir. (for mail) Anthracite Industries Laboratory, Primos, Delaware Co., and 344 Congress Ave., Lansdowne, Pa.

JOHNSON, Carl E. (*A* 1939; *J* 1930) Pres. (for mail) Sunbeam Heating & Air Conditioning Co., 752 Spring St. N. W., and 1154 Ridgewood Dr. N. E., Atlanta, Ga.

JOHNSON, Carl W. (*M* 1912) Pres. (for mail) C. W. Johnson, Inc., 211 N. Desplaines St., and 1809 Morse Ave., Chicago, Ill.

JOHNSON, Clarence W. (*M* 1933; *J* 1931) Dist. Mgr. (for mail) Canadian Sirocco Co., Ltd., 630 Dorchester St. W., Montreal, and 123 Dobie Ave., Town of Mt. Royal, Montreal, Que., Canada.

JOHNSON, Edward B. (*M* 1919) Sales Engr., Staten Island Supply Co., Inc., 1390 Richmond Terrace, West New Brighton, and (for mail) 154 Wardwell Ave., Port Richmond, S. I., N. Y.

JOHNSON, Fred W. (*M* 1916) Vice-Pres. (for mail) Johnson Larsen Co., 6530 Beaubien St., Detroit, and Adams Rd., R. F. D. 2, Birmingham, Mich.

JOHNSON, Helge S. (*A* 1933; *J* 1927) Assoc. Dist. Mgr. (for mail) Buffalo Forge Co., 39 Cortlandt St., New York, and 15 Tunstall Rd., Scarsdale, N. Y.

JOHNSON, Leslie O. (*M* 1938; *J* 1930) Sales Engr., H. Y. Keeler Co., 910 Hines Bldg., and (for mail) 2520 First Ave., Huntington, W. Va.

JOHNSON, Oliver W. (*M* 1938) Chemical Engr., Standard Oil Co. of California, 225 Bush St., San Francisco, and (for mail) 1831 Waverly St., Palo Alto, Calif.

JOHNSON, Ralph B. (*M* 1922) Owner (for mail) Ralph B. Johnson & Co., 201 Petroleum Bldg., and 2111 W. Main St., Houston, Tex.

JOHNSON, Tracy R. (*M* 1924) Br. Mgr. (for mail) The Trane Co., 818 Hubbell Bldg., and 3438 University Ave., Des Moines, Ia.

JOHNSTON, J. Ambler (*M* 1912) Partner (for mail) Carneal, Johnston & Wright, Archts. & Engrs., 1000 Atlantic Life Bldg., Richmond, Va.

JOHNSTON, Marriott T. (*M* 1939) Engr., Bankers Life Co., Seventh & High St., and (for mail) 1641 38th St., Des Moines, Ia.

JOHNSTON, Robert E. (*M* 1929; *A* 1926) Pres. (for mail) R. E. Johnston Co., Ltd., 1070 Homer St., and 3342 West 33rd Ave., Vancouver, B. C., Canada.

JOHNSTON, Robert M. (*J* 1937) Instructor (for mail) Virginia Polytechnic Institute, Dept. of Mech. Engrg., Box 548, and 707 Main St., Blacksburg, Va.

JOLLEY, Edward M. (*A* 1939) Sales Engr. (for mail) The Marley Co., 3001 Fairfax Rd., Kansas City, Kans., and 11 West 73rd Terrace, Kansas City, Mo.

JONES, Alfred (*M* 1928) Consulting Engr. (for mail) Armstrong Cork Co., P. O. Box 540, and 402 N. President Ave., Lancaster, Pa.

JONES, Alfred L. (*M* 1926) Owner (for mail) Alfred L. Jones, 431 Greenwich Ave., Greenwich, and Breezemont Ave., Riverside, Conn.

JONES, Allan T. (*M* 1937; *J* 1935) Chief Engr., S. A. Armstrong, Ltd., 720 Bathurst St., and (for mail) 254 Castlefield Ave., Toronto, Ont., Canada.

JONES, Bernard G. (*M* 1928) Mgr. (for mail) Acme Fan & Blower Co., Ltd., 868 Arlington St., and 542 Raglan Rd., Winnipeg, Man., Canada.

JONES, Edwin (*M* 1933; *J* 1924) Engr. & Estimator (for mail) Watt Plumbing, Heating & Supply Co., 608 S. Cincinnati, and 1436 East 17th Pl., Tulsa, Okla.

JONES, Edwin A. (*M* 1919) Consulting Engr. (for mail) 4381 N. Alpine Ave., Milwaukee, Wis.

JONES, Edwin F. (*M* 1923) Utilities Engr. City of St. Paul, 216 Courthouse, and (for mail) 220 Montrose Pl., St. Paul, Minn.

JONES, Harold L. (*M* 1920) Supt. (for mail) W. W. Farrier Co., 44 Montgomery St., Jersey City, and 11 Cambridge Rd., Glen Ridge, N. J.

JONES, Harold S. (*S* 1940) Student (for mail) P. O. Box 338, Carnegie Institute of Technology, Pittsburgh, Pa., and 122 Kensington Rd., Greensboro, N. C.

JONES, James T. (*A* 1939) Air Cond. Engr. (for mail) Empire District Electric Co., and 328 N. Jackson, Joplin, Mo.

JONES, John P. (*M* 1937) (for mail) John Paul Jones, Cary & Miller, 448 Terminal Tower, Cleveland, and 3161 Scarborough Rd., Cleveland Heights, O.

JONES, Lawrence K. (*M* 1939) Mgr., Special Test Section (for mail) Pittsburgh Testing Laboratory, 1330 Locust St., and 320 S. Aiken Ave., Pittsburgh, Pa.

JONES, Sprague (*M* 1936) Pres. (for mail) Sprague Jones, Inc., 1914 Vermont Ave., and Apt. 4, 3769 Lockwood Ave., Toledo, O.

JONES, William T. (*M* 1915) (*Presidential Member*) (Pres., 1933; 1st Vice-Pres., 1932; 2nd Vice-Pres., 1931; Council, 1925-33) Treas., Barnes & Jones, 128 Brookside Ave., Jamaica Plain, Boston, and (for mail) 16 Harvard St., Newtonville, Mass.

JORDAN, Richard C.* (*M* 1940; *J* 1935; *S* 1933) Instructor (for mail) Engrg. Experiment Station, Room 209, Universtiy of Minnesota, and 1101 East River Rd., Minneapolis, Minn.

JORDY, Jules J. (*J* 1940) Engr. & Contractor (for mail) Jordy Bros., Inc., 518 Julia St., and 1323 Webster St., New Orleans, La.

JOSEPHSON, Simon (*J* 1936) Supervising Engr. (for mail) Astor Plumbing & Heating Corp., 1134 Bedford Ave., and 199 E. Second St., Brooklyn, N. Y.

JUERGENS, Walter A. (*A* 1940) Sales Engr. (for mail) John P. Childe Co., P. O. Box No. 2, Lockland Station, and 1447 Aster Pl., Cincinnati, O.

JUNG, John S. (*M* 1930; *A* 1923) Htg. & Piping Contractor & Engr. (for mail) 2409 W. Greenfield Ave., and 1516 S. Layton Blvd., Milwaukee, Wis.

JUNKER, William H. (*M* 1936) Plant Engr., Emery Industries, Inc., 4300 Carew Tower, and (for mail) 6068 Dryden Ave., Cincinnati, O.

K

KACZENSKI, Chester (*A* 1939; *J* 1933) Estimator, The Lummus Co., 420 Lexington Ave., and (for mail) 5 West 63rd St., New York, N. Y.

KADEL, George B. (*A* 1940; *J* 1938) Engr., E. R. Squibb & Sons, 25 Columbia Heights, Brooklyn, and (for mail) 62 E. Beverly Pkwy., Valley Stream, L. I., N. Y.

KAERCHER, Carl M. H. (*M* 1937) Managing Dir. (for mail) Central Bureau for Heating & Air Conditioning, 3030 Euclid Ave., Cleveland, and 2560 Ashurst Rd., University Heights, O.

KAHN, Charles R., Jr. (*J* 1939) Engr., Armo Cooling & Ventilating Co., 30 West 15th St., and (for mail) 110 Riverside Dr., New York, N. Y.

KAISER, Charles W. (*A* 1939; *J* 1938) Instructor Board of Education, City of New York, Jamaica Vocational High School, Jamaica, and (for mail) 37-35 57th St., Woodside, L. I., N. Y.

KAISER, Fred (*M* 1935) Br. Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 415 Brainard St., Detroit, and 874 Westchester Rd., Grosse Pointe Park, Mich.

KAJUK, Andrew E. (*M* 1936) Engr., War Dept., O. Q. M. G. Engrg. Design-Mobilization Dept., Washington, D. C., and (for mail) 2010 Lansdowne Way, Silver Spring, Md.

KALINSKY, Alex G. (*J* 1936; *S* 1934) Htg. Engr. (for mail) Fox Furnace Div., American Radiator & Standard Sanitary Corp., and Y. M. C. A., Elyria, O.

KAMMAN, Arnold R. (*A* 1925; *J* 1921) (for mail) Arnold R. Kamman Co., 493 Franklin St., Buffalo, and R. F. D., Hamburg, N. Y.

KAPPEL, George W. A. (*M* 1921) Pres.-Treas. (for mail) Camden Heating Co., 8 Market St., Camden, and 421 Maple Ave., Westmont, N. J.

KARAKASH, Theodore J. (*A* 1940; *J* 1936) Engr. in Chief (for mail) Carrier Air Cond. Dept., G. & A. Baker, Ltd., P. O. Box 468, Istanbul, Turkey.

KARGES, Albert (*A* 1935) Managing Dir., The James Stewart Manufacturing Co., Ltd., and (for mail) 37 Perry St., Woodstock, Ont., Canada.

KARLSON, Alfred F. (*M* 1918) Chief Engr. (for mail) Parks-Cramer Co., 970 Main St., Fitchburg, and 186 Prospect St., North Leominster, Mass.

KARLSTEEN, Gustav H. (*M* 1935) Plant Engr., Dunlop Tire & Rubber Corp., Buffalo and (for mail) Box 55, Route 1, Tonawanda, N. Y.

KARSUNKY, William K. (*M* 1939) Consulting Engr. (for mail) 1223 Connecticut Ave. N. W., Washington, D. C., and Three Oaks, Kensington, Md.

KARTORIE, V. T. (*J* 1935; *S* 1933) Asst. to Gen. Sales Mgr. (for mail) York Ice Machinery Corp., York, and 1513 Third Ave., Elmwood, York, Pa.

KAUFMAN, Hiram J. (*M* 1937) Htg. & Vtg. Engr., Commonwealth & Southern Corp., Consumers Power Bldg., Jackson, and (for mail) 13215 Roselawn Ave., Detroit, Mich.

KAUP, Edgar O. (*M* 1938) Sales Engr., Edgar O. Kaup, 323 10th St., San Francisco, and (for mail) 1249 Garfield Ave., Albany, Calif.

KAWASE, Sumio (*M* 1936) Chief Engr., Manshu Imono Kabushiki Kaisha, 105 5-dan Toagai yamatoku, Hoten, and (for mail) 22 Awoi-cho, Yamatoku, Hoten, Manchoukuo.

KEANE, Gerard F. (*M* 1939) In charge of Engrg., Cooney Refrigeration Co., 228 Walton St., Syracuse, and (for mail) 316 Haddonfield Dr., Dewitt, N. Y.

KEARNEY, Joseph S. (*M* 1939) Pres. & Mgr., Northwestern Heating & Plumbing Co., 1465 Sherman Ave., and (for mail) 2001 Bennett Ave., Evanston, Ill.

KEATING, Arthur J. (*M* 1937) Engr. (for mail) Powers Regulator Co., 2720 Greenview Ave., and 4420 W. Congress St., Chicago, Ill.

KEELAND, B. W. (*A* 1938) Owner (for mail) B. W. Keeland Heating Co., 2711 Westheimer, and 2723 Kipling St., Houston, Tex.

KEELING, Fred. V. (*A* 1940) Planner & Estimator, Htg. & Plbg., Bldg. No. 11, Public Works Dept., Philadelphia Navy Yard, and (for mail) 960 Arrott St., Philadelphia, Pa.

KEENEY, Frank P. (*A* 1915) Pres. (for mail) Keeney Publishing Co., 6 N. Michigan Ave., and 7059 South Shore Dr., Chicago, Ill.

KEHM, Horace Stevens (*M* 1928))Pres. (for mail) The Kehm Corporation, 51 E. Grand Ave., and 3240 Lake Shore Dr., Chicago, Ill.

KEITH, James P. (*M* 1938) Consulting Engr., Vice-Pres., Canadian Domestic Engineering Co., Ltd., 1440 St. Catherine St. W., and (for mail) 5196 Durocher Ave., Montreal, Que., Canada.

KEITHLEY, Frank R. (*J* 1939) Sales Engr., Malvin & May, Inc., 2427 S. Michigan Ave., Chicago, Ill., and (for mail) 1152 Pearl St., Denver, Colo.

KELBLE, Frank R. (*M* 1928) Vice-Pres. & Mgr. (for mail) Huffman-Wolfe Co. of Philadelphia, 4600 North 18th St., Philadelphia, and 305 Pleasant Ave., Glenside, Pa.

KELLA, Waldon B. (*M* 1939) Mgr., Air Cond. Dept. (for mail) Fairbanks, Morse & Co., 217 S. Eighth St., St. Louis, and No. 4 Salisbury Dr., Air Port Park, St. Louis Co., Mo.

KELLER, George A. (*A* 1938) (for mail) P. O. Box 481, Wantagh, L. I., N. Y.

KELLEY, Francis J. (*A* 1940) Mfrs. Repr. (for mail) 200 Alva Bldg., 810 Union St., and 6301 Constance St., New Orleans, La.

KELLEY, James J. (*A* 1924) Fuel Oil & Burner Asst. (for mail) Colonial Beacon Oil Co., 378 Stuart St., Boston, and 142 Governors Ave., Medford, Mass.

KELLOGG, Alfred S. (*Life Member*; *M* 1916) (Council, 1920-21; 1923-24) Consulting Engr., Belmont, Mass.

KELLOGG, Winston T. (*A* 1938) Owner (for mail) The W. T. Kellogg Co., 218 Pyramid Bldg., and 2020 Country Club Lane, Little Rock, Ark.

KELLY, Charles J. (*M* 1931) Agent, James P. Marsh Corp., 155 East 44th St., New York, N. Y., and (for mail) 440 Fairmount Ave., Jersey City, N. J.

KELLY, H. J. (*A* 1940) Sales Engr. (for mail) 816 Howard Ave., and 4105 Elba St., New Orleans, La.

KELLY, Olin Arthur (*S* 1940) Student, Mech. Engrg. (for mail) Michigan State College, North Hall, East Lansing, Mich.

KELLY, Wilbur C. (*M* 1935) Field Engr. (for mail) Iron Fireman Manufacturing Co. of Canada, Ltd., 602 King St. W., and 58 Elmsthorpe Ave., Toronto, Ont., Canada.

KELSEY, Harold D. (*M* 1940) Engr. (for mail) General Electric Co., 5 Lawrence St., Bloomfield, and 30 Francis Pl., Caldwell, N. J.

KENNEDY, Maron (*A* 1936; *J* 1930) Sales Engr. (for mail) York Ice Machinery Corp., 5051 Santa Fe Ave., Los Angeles, and 2704 Carlaris Rd., San Marino, Calif.

KENNEDY, Owen A. (*J* 1938; *S* 1933) Plbg. & Htg. Engr. (for mail) 2431 Dixie Highway, South Ft. Mitchell, Ky.

KENNEY, Thomas W. (*M* 1937) Sales Engr., E. J. Deckman Co., Oliver Bldg., and (for mail) 214 Gilliland Pl., Bellevue, Pittsburgh, Pa.

KENT, Laurence F. (*A* 1927; *J* 1924) Pres. (for mail) Moncrief Furnace Co., P. O. Box 1673, and 1515 Morningside Dr. N. E., Atlanta, Ga.

KENT, Richard L. (*M* 1936) Dist. Mgr. (for mail) Trane Co. of Canada, Ltd., 365 Hargrave St., Winnipeg, Man., Canada.

KEPLER, Donald A. (*J* 1936; *S* 1934) Operating Engr., N. Y. Stock Exchange Bldg. Co., 20 Broad St., and (for mail) 28 West 73rd St., New York, N. Y.

KERN, Joseph F., Jr. (*A* 1937) Assoc. Editor, Heating & Ventilating, 148 Lafayette St., New York, N. Y., and (for mail) 42-16 80th St., Elmhurst, L. I., N. Y.

KERN, Raymond T. (*M* 1927) Chief Engr., Jennison Co., 17 Putnam St., Fitchburg, and (for mail) 51 Claflin St., Leominster, Mass.

KERR, Gerald C. (*A* 1940) Acoustical Engr. (for mail) Taylor-Seidenbach, Inc., 1401 Tchoupitoulas St., and 625 Pine St., No. 2, New Orleans, La.

KERR, William E. (*M* 1937) Sales Repr. (for mail) 1201 Hyatt Ave., Columbia, S. C.

KERSHAW, Melville G. (*M* 1932; *A* 1926; *J* 1921) Vtg. & Air Cond. Engr. (for mail) E. I. DuPont de Nemours & Co., Wilmington, Del., and 7313 North 21st St., Philadelphia, Pa.

KESSLER, Clarence F. (*M* 1938) Asst. Prof. Mech. Engrg. (for mail) University of Michigan, 241 W. Engrg. Bldg., and 1756 Broadway, Ann Arbor, Mich.

KETTER, Jack W. (*J* 1937) Draftsman, Krenz & Co., 5114 W. Center St., and (for mail) 3042 N. Second St., Milwaukee, Wis.

KEYSER, Herman M. (*A* 1937) Sales Engr., M. W. Sales & Co., 801 W. Baltimore, Detroit, and (for mail) 10703 Hart, Huntington Woods, Royal Oak, Mich.

KICZALES, Maurice D. (*M* 1935) Mech. Engr., U. S. Army Motion Picture Service, Tower Bldg., and (for mail) 6200-31 St. N. W., Washington, D. C.

KIDD, Charles R. (*A* 1938) Owner & Engr., C. R. Kidd Co., 712 N. Broadway, and (for mail) 611 Northwest 28th St., Apt. 1, Oklahoma City, Okla.

KIEFER, Carl J. (*M* 1922) Vice-Pres. & Treas. (for mail) Schenley Products Co., 607 Schmidt Bldg., and 984 Lenox Place, Avondale, Cincinnati, O.

KIEFER, Elmer J. (*A* 1932; *J* 1928) Mgr. (for mail) H. C. Archibald Co., 406 Main St., and 108 N. Sixth St., Stroudsburg, Pa.

KILLEEN, Edmund F. (*A* 1941) Sales Engr. (for mail) Shedlov Oil Burners, Inc., 717 Third Ave. S., and Nordic Hotel, Third Ave. S. and Ninth St., Minneapolis, Minn.

KILLIAN, V. J. (*A* 1937) Pres. (for mail) V. J. Killian Co., 907 Linden Ave., and 1348 Edgewood Lane, Winnetka, Ill.

KILLIAN, William J. (*A* 1940) Sales Repr., Herman Nelson Corp., and (for mail) 3810 Edwards Rd., Cincinnati, O.

KILLOUGH, Robert E. (*A* 1938) Engr., Standard Oil Co. of Pennsylvania, 1618 N. Broad St., and (for mail) 2108 E. Chelten Ave., Germantown, Philadelphia, Pa.

KILNER, John S. (*M* 1929) Sales Engr., Clarage Fan Co., 7310 Woodward Ave., and (for mail) 1091 Seminole Ave., Detroit, Mich.

KILPATRICK, William S. (*M* 1923) (for mail) W. S. Kilpatrick & Co., 1100 East 33rd St., Los Angeles, Calif.

KIMBALL, Charles W. (*M* 1915) Treas. (for mail) Richard D. Kimball Co., 6 Beacon St., Boston, and 65 Prescott St., W. Medford, Mass.

KIMBALL, Dwight D.* (*Life Member*; *M* 1908) (*Presidential Member*) (Pres., 1915; 2nd Vice-Pres., 1914; Board of Governors, 1912-16) Consulting Engr. (for mail) Room 1728, Grand Central Terminal Bldg., and 145 West 58th St., New York, N. Y.

KIMBLE, Carl W. (*J* 1938) Owner-Partner (for mail) Advance Heating & Sheet Metal Works, 315 24th St., and 1620 22nd St., Rock Island, Ill.

KIMMELL, Phillip M. (*A* 1941; *J* 1930) Asst. Mech. Engr., Eastman Kodak Co., Kodak Park, and (for mail) 50 Freemont Rd., Rochester, N. Y.

KINCAIDE, Merrill C. (*M* 1939; *A* 1937; *J* 1936) (for mail) Aero Products Div., and 729 Grand Ave., Dayton, O.

KING, Arthur C. (*M* 1936) Consulting Engr., 35 S. Dearborn St., Chicago, Ill.

KING, John S. (*A* 1940) Mech. Inspector (for mail) c/o Office of Constructing Quartermaster, Ft. Dix, and 134 Garden St., Mount Holly, N. J.

KING, Robert W. (*A* 1940) Sales Engr. (for mail) Asbestos Covering & Roofing Co., 4104 Georgia Ave., Washington, D. C., and 5705 York Lane, Greenwich Forest, Md.

KING, Roy L. (*J* 1936; *S* 1933) Engr., Delco Appliance Div., General Motors Corp., and (for mail) 1011 University Ave., Rochester, N. Y.

KINGSLAND, George D. (*M* 1935) Vice-Pres. (for mail) Perfex Corp., 370 Lexington Ave., and 59 East 56th St., New York, N. Y.

KINGSWELL, William E. (*M* 1935) Pres. (for mail) William E. Kingswell, Inc., 3707 Georgia Ave. N. W., and 2739 Macomb St. N. W., Washington, D. C.

KINNEY, Aldon M. (*M* 1936) Pres. (for mail) A. M. Kinney, Inc., Cons. Engrs., 1301 Enquirer Bldg., Cincinnati, and 7013 Rembold Ave., Mariemont, O.

KIPE, J. Morgan (*M* 1919) Dir. of Education, Anthracite Merchandising School, Primos, and (for mail) 801 Homestead Ave., Beechwood, Del. Co., Pa.

KIPP, Theodore (*M* 1937) Pres. & Managing Dir. (for mail) Kipp-Kelly, Ltd., 68 Higgins Ave., and 1030 Wellington Crescent, Winnipeg, Man., Canada.

KIRKBRIDE, J. Owen (*M* 1938) Partner (for mail) Parent & Kirkbride, N. W. Corner Fourth & Locusts Sts., Philadelphia, Pa., and 1121 Eldridge Ave., West Collingswood, N. J.

KIRKENDALL, Horton J. (*A* 1938) Sales Repr., Htg. Equip., 291 Catalpa Pl., Pittsburgh (16), Pa.

KIRKPATRICK, Arthur H. (*M* 1935; *J* 1931) Dist. Sales Engr. & Mgr. (for mail) Ilg Electric Ventilating Co., 415 Brainard St., and Hotel Webster Hall, Detroit, Mich.

KIRTLAND, Eugene M. (*A* 1940) Mgr., Engineering Specialty Co., Gary, Ind.

KITAURA, Shigeyuki (*M* 1918) 191 Gotanda 6 Chome, Shanagawa-ku, Tokyo, Japan.

KITCHEN, Francis A. (*A* 1927; *J* 1923) Pres. (for mail) American Warming & Ventilating Co., 1514 Prospect Ave., Cleveland, and 2077 Campus Rd., South Euclid, O.

KITCHEN, John H. (*Life Member*; *M* 1906) Pres. & Mgr. (for mail) John H. Kitchen & Co., 1016 Baltimore Ave., and 5015 Westwood Terrace, Kansas City, Mo.

KITCHEN, William H. J. (*A* 1938) Engr. (for mail) Engineering Installations, Ltd., 1154 Beaver Hall Square, Montreal, Que., Canada.

KLAGES, Frank E. P. (*M* 1940) Dist. Mgr. (for mail) The Powers Regulator Co., 1034 Jefferson Standard Bldg., and 1512 Edgedale Road, Greensboro, N. C.

KLEIN, Albert R. (*M* 1920) Managing Dir. (for mail) Lufttechnische Gesellschaft, Stuttgart-W, Konigstrasse 84, and Heidehofstrasse 40, Stuttgart, Germany.

KLEIN, Edward W. (*M* 1917) Dist. Repr. (for mail) Warren Webster & Co., 152 Nassau St. N. W., and 456 Peachtree Battle Ave., Atlanta, Ga.

KLEIN, Fred R. (*A* 1940) Asst. Mgr. Htg. Dept., Globe Machinery & Supply Co., 205 Court Ave., and (for mail) 1434 45th St., Des Moines, Ia.

KLEINKAUF, Henry (*M* 1938; *J* 1937) Mgr. (for mail) Natkin & Co., 1729 Howard St., and 2739 Fontenelle Blvd., Omaha, Nebr.

KLIE, Walter (*M* 1915) Pres. (for mail) The Smith & Oby Co., 6107 Carnegie Ave., Cleveland, and 18411 S. Woodland Ave., Shaker Heights, O.

KLIEFOTH, Max H. (*A* 1939) Treas. (for mail) Research Products Corp., 1011-15 E. Washington Ave., and Fuller's Woods, Madison, Wis.

KLUCKHUHN, Frederick H. (*J* 1940) Engr. & Draftsman, Eberly & Brand, 6105 Blair Rd. N. W., Washington, D. C., and (for mail) 1110 Montgomery Ave., Laurel, Md.

KLUGE, Burnett M. (*J* 1938) Sales Engr., Bayley Blower Co., 1817 South 66th St., and (for mail) 1018-A North 35th St., Milwaukee, Wis.

KNAPP, Andrew E. (*M* 1937) Engr. (for mail) Nash Kelvinator, 14250 Plymouth Rd., and 8050 Sorrento, Detroit, Mich.

KNAPP, Don S. (*A* 1936) Dist. Mgr. (for mail) Chamberlin Metal Weather Strip Co., Inc., 2400 Hennepin Ave., and 4607 Wooddale Ave., Minneapolis, Minn.

KNEPPER, H. H. (*A* 1938) Erection Engr. (for mail) Minneapolis-Honeywell Regulator Co., 378 Saunders-Kennedy Bldg., Omaha, Nebr., and 122 Gold St., Kendallville, Ind.

KNIBB, Alfred E. (*M* 1930) Htg. Engr. (for mail) L. L. McConachie Co., 1003 Maryland Ave., and 9333 E. Jefferson Ave., Detroit, Mich.

KNOWLES, Elwin L. (*A* 1937) Owner (for mail) Knowles Air Conditioning, 1324 Marshall St. N. E., and 400 Thomas Ave. S., Minneapolis, Minn.

KNOWLES, Frank R. (*A* 1938) Dir. Commercial Engrg. Dept. (for mail) Pennsylvania Electric Co., 535 Vine St., and R. D. 4, Box 340, Johnstown, Pa.

KNOX, John C. (*A* 1938) Secy.-Treas. (for mail) Waterloo Register Co., and 176 Gates St., Waterloo, Ia.

KNUDSEN, William R. (*M* 1937) Consulting Engr., 897 S. Eighth St., San Jose, Calif.

KOCH, Albert H. (*M* 1938) Br. Mgr., Minneapolis-Honeywell Regulator Co., 101 Marietta Bldg., and (for mail) 3687 Peachtree Rd., Atlanta, Ga.

KOCH, Richard G. (*A* 1935) Househeating Engr. (for mail) Milwaukee Gas Light Co., 626 E. Wisconsin Ave., and 5707 W. Brooklyn Pl., Milwaukee, Wis.

KOEHLER, C. Stewart (*A* 1936) Sales Engr., Minneapolis-Honeywell Regulator Co., 801 Second Ave., and (for mail) 4374 Richardson Ave., New York, N. Y.

KOENIG, Andrew C. (*J* 1940) Sales Engr., J. A. Koenig Sheet Metal Works, 701 E. Missouri St., Evansville, Ind.

KOHLER, John C. (*A* 1939) Partner (for mail) John C. Kohler Co., 554 North 16th St., and 629 Crestview Road, Philadelphia, Pa.

KOHLER, Walter J., Jr. (*A* 1933) Secy. (for mail) Kohler Co., and "Windway," Kohler, Wis.

KOLB, Fred W. (*M* 1938) Owner (for mail) Fred W. Kolb, 598 Monadnock Bldg., and 82 Macondray St., San Francisco, Calif.

KOLB, Robert P. (*M* 1939) Prof. of Heat Power Engrg. (for mail) Worcester Polytechnic Institute, and 215 May St., Worcester, Mass.

KOLLAS, Will J. (*M* 1939) Chief Engr., Montag Stove & Furnace Works, 2011 N. Columbia Blvd., and (for mail) 6104 N. Missouri Ave., Portland, Ore.

KONZO, S.* (*M* 1937; *A* 1936; *J* 1932) Special Research Asst. Prof., University of Illinois, Mech. Engrg. Lab., and (for mail) 1108 W. Stoughton St., Urbana, Ill.

KOOISTRA, John F. (*M* 1933) Br. Mgr. (for mail) Carrier Corp., 625 Market St., San Francisco, and 1245 Laguna Ave., Burlingame, Calif.

KORN, Charles B. (*M* 1922) Member of Firm, Reber-Korn Co., 817 Cumberland St., and (for mail) 1022 S. Eighth St., Allentown, Pa.

KOSTER, Howard H. (*J* 1939) Asst. Prof. of Mech. Engrg. (for mail) The George Washington University, Washington, D. C., and Sylvan Dr., Sleepy Hollow, Falls Church, Va.

KRAMER, Conrad (*J* 1938) Air Cond. Engr., 1611 Westminster St., Providence, and (for mail) 230 Smithfield Ave., Pawtucket, R. I.

KRAMIG, Robert E., Jr. (*A* 1933) Vice-Pres.-Treas. (for mail) R. E. Kramig & Co., Inc., 222-4 East 14th St., Cincinnati, and 115 Linden Drive, Wyoming, O.

KRAMINSKY, Victor (*M* 1936) Managing Dir. (for mail) Air Conditioning & Engineering, Ltd., 123d Victoria St., Westminster, and 608 Howard House, Dolphin Square, London, S. W. 1, England.

KRATZ, Alonzo P.* (*M* 1925) (Council, 1938-40) Research Prof. (for mail) Dept. of Mech. Engrg., University of Illinois, and 1003 Douglas Ave., Urbana, Ill.

KREINER, Jack (*A* 1940) (for mail) 153 East 26th St., New York, and 5100-15th Ave., Brooklyn, N. Y.

KRENZ, Alfred S. (*M* 1937; *A* 1935) Pres. (for mail) Krenz & Co., Inc., 5114 W. Center St., Milwaukee, and 1766 North 74th St., Wauwatosa, Wis.

KREZ, Leonard (*A* 1935) Secy. (for mail) Paul J. Krez Co., 444 N. LaSalle St., and 4716 N. Paulina St., Chicago, Ill.

KRIBS, Charles L., Jr. (*M* 1935) Pres. (for mail) Kribs & Landauer, 404 Dallas Gas Bldg., and 3608 Drexel Ave., Dallas, Tex.

KRIEBEL, Arthur E. (*M* 1920) Sales Engr. (for mail) Haynes Selling Co., Inc., Ridge Ave. & Spring Garden St., Philadelphia, and Berwyn, Chester Co., Pa.

KROEKER, J. Donald (*M* 1936) Consulting Mech. Engr. (for mail) 301 Failing Bldg., and 6831 N. E. Siskiyou St., Portland, Ore.

KRUEGER, James I. (*M* 1921) Mfrs. Repr. (for mail) 357 Ninth St., and 1920 Sacramento St., Apt. 12, San Francisco, Calif.

KRUSE, W. C., Jr. (*M* 1938) Repr. (for mail) American Air Filter Co., 24 Commerce St., Newark, and 32 University Court, South Orange, N. J.

KUBASTA, Robert W. (*A* 1939; *J* 1936) Sales Engr. (for mail) Carrier Corp., 302 S. Geddes St., and 405 Comstock Ave., Syracuse, N. Y.

KUCHER, Andrew A. (*M* 1938) Mgr. Air Cond. Engrg. (for mail) Frigidaire Corp., 300 Taylor St., Plant No. 1, Dayton, O.

KUCINSKI, William V. (*A* 1939) Pres., General Conditioners, Inc., 24 Commerce St., and (for mail) 10 Peck Ave., Newark, N. J.

KUECHENBERG, William A. (*M* 1937) Pres. (for mail) R. B. Hayward Co., 1714 Sheffield Ave., Chicago, and 427 Elmore Ave., Park Ridge, Ill.

KUEMPEL, Leon L. (*M* 1936; *J* 1929) Hughes Heating & Air Conditioning Co., 125 N. Jefferson St., and (for mail) 200 Briarcliff Rd., Dayton, O.

KUGEL, H. Kenneth (*M* 1938) Chief Engr. Div. of Smoke Regulation & Boiler Inspection, 306 District Bldg., and (for mail) 3825 Morrison St. N. W., Washington, D. C.

KUHLMANN, Rudolf (*M* 1928) (for mail) Ameresco, Inc., 50 Church St., and 122 East 42nd St., New York, N. Y.

KUMMER, Calvin J. (*J* 1938) Engr. (for mail) Carrier Corp., 7-122 Merchandise Mart Bldg., and 4820 N. Winchester, Chicago, Ill.

KUNEN, Herbert K. (*J* 1938) Asst. Chief Engr., Anemostat Corp. of America, 10 East 39th St., New York, N. Y., and (for mail) 117-14 Union Turnpike, Kew Gardens, L. I., N. Y.

KUNTZ, Edward C. (*J* 1937) Sales Engr., Hammond Sheet Metal Co., 119 Cass Ave., and (for mail) 6516a Morganford Rd., St. Louis, Mo.

KUNZOG, Theodore W. (*M* 1939) Consultant, Htg., Air Cond. & Refrig., 570 Newark Ave., and (for mail) 126 Corbin Ave., Jersey City, N. J.

KURTH, Franz J. (*M* 1937) Vice-Pres. & Tech. Mgr. (for mail) Anemostat Corp. of America, 10 East 39th St., New York, and 510 Cortlandt Ave., Mamaroneck, N. Y.

KURTZ, Robert W. (*J* 1936) Asst. Secy. & Treas., Sales & Engrg. (for mail) Air Conditioning Co., 1017 Sampson, and 415 Hyde Park, Houston, Tex.

KWAN, I. K. (*M* 1933) Gen. Mgr., The China Engineering Co., 30 Brenan Rd., Shanghai, China.

L

LABONNE, Henri (*A* 1940) Sales Mgr., Codère, Ltd., 18 Wellington Street N., and (for mail) 126 Bowen St. S., Sherbrooke, Que., Canada.

LADD, David (*M* 1938) Br. Mgr. (for mail) The Powers Regulator Co., 2240 N. Broad St., and 305 E. Wadsworth St., Philadelphia, Pa.

LAGODZINSKI, Harry J. (*A* 1927; *J* 1920) Sales Engr. (for mail) Ilg Electric Ventilating Co., 222 N. LaSalle St., Chicago, and Crystal Lake, Ill.

LAIR, Paul H. (*M* 1940) Consulting Engr. (for mail) The P. H. Lair Associates, 120 Milk St., Boston, and 52 Athelstane Rd., Newton Center, Mass.

LaMONTAGNE, Arthur F. (*A* 1936) Sales Mgr., Htg. Div. (for mail) Gurney Foundry Co., Ltd., P. O. Box 277, Montreal, and 24 Prince Arthur St., St. Lambert, Que., Canada.

LANDAU, Mitchel (*M* 1937) Chief Engr., ABC Oil Burner & Engineering Co., 108 South 22nd St., and (for mail) 1851 Widener Pl., Philadelphia, Pa.

LANDAUER, Leo L. (*M* 1938; *J* 1932) Member of Firm (for mail) Kribs & Landauer, Mech. Engrs., 617 Dallas Gas Bldg., and 4433 Stanhope, Dallas, Tex.

LANDERS, John J. (*M* 1930; *J* 1924) Mfrs. Repr. (for mail) 701 Crosby Bldg., Buffalo, and 120 Burroughs Drive, Snyder, N. Y.

LANDES, Bates E. (*M* 1938) Consulting Engr. (for mail) 915 Hubbell Bldg., and 1603 47th St., Des Moines, Ia.

LANDES, Benjamin D. (*A* 1937) 3221 Beacon Hill Ave., Dormont, Pittsburgh, Pa.

LANE, D. Duffy (*M* 1934) Secy., Frank O'Hara, Inc., 40-10 82nd St., Jackson Heights, and (for mail) 87-65 52nd Ave., Elmhurst, L. I., N. Y.

LANG, Jacob (*A* 1938) Warm Air Htg. Technician, (for mail) Lang & Lang, 91-48 119th St., and 91-48 Lefferts Blvd., Richmond Hill, L. I., N. Y.

LANG, J. Clifford (*J* 1937) Commercial Field Engr., York Ice Machinery Corp., 117 South 11th St., St. Louis, Mo., and (for mail) 606 Washington Pl., E. St. Louis, Ill.

LANGE, Fred F. (*A* 1934) Pres. (for mail) The Mechanical Service Co., 809 Pence Bldg., Minneapolis, and 356 N. Cleveland Ave., St. Paul, Minn.

LANGE, Raymond T. (*M* 1936) Engr., Hartzell Propeller Fan Co., Box 909, and (for mail) 224 Jackson St., Piqua, O.

LANOU, J. Ernest (*M* 1931) Mgr. (for mail) F. S. Lanou & Son, 90 St. Paul, and 48 Brookes Ave., Burlington, Vt.

LaRAUS, Julius (*J* 1940) Engr., Jowein, Inc., 150-17 Liberty Ave., Jamaica, and (for mail) 22-21 76th St., Jackson Heights, L. I., N. Y.

LaROI, George H. (*J* 1936) Adv. Mgr. (for mail) McDonnell & Miller, 1318 Wrigley Bldg., and 4434 N. Monitor Ave., Chicago, Ill.

LARSON, Carl W. (*M* 1936) Sales Engr., Barnes & Jones, Inc., Industrial Trust Bldg., Rm. 1219, Providence, R. I., and (for mail) 641 Hyde Park Ave., Roslindale, Mass.

LARSON, Clifford P. (*A* 1939; *J* 1936) Research Engr., The Insulite Co., 1100 Builders Exchange and (for mail) 2011 Pillsbury Ave., Minneapolis, Minn.

LARSON, Gustus L.* (*M* 1923) (*Presidential Member*) (Pres., 1936; 1st Vice-Pres., 1935; 2nd Vice-Pres., 1934; Council, 1929-37) Prof., Steam and Gas Engrg., and Chairman of Dept. of Mech. Engrg. (for mail) University of Wisconsin, Mech. Engrg. Bldg., and 1213 Sweetbriar Rd., Shorewood Hills, Madison, Wis.

LaRUE, J. A. Delphis (*A* 1939) Mgr. (for mail) Roofers, Inc., 726 Atwater Ave., and 4309A Delaroche St., Montreal, Que., Canada.

LaRUE, Perry (*M* 1938) Dir. of Bldgs. & Grounds, Independent School Dist., 629 Third St., Des Moines, Ia.

LaSALVIA, James J. (*M* 1930) Mech. Engr., Frigidaire Div., General Motors Sales Corp., 300 Taylor St., and (for mail) 317 Otterbein Ave., Dayton, O.

LASETER, Frank L. (*M* 1938) Mgr., Heating Dept. (for mail) Atlanta Gas Light Co., 243 Peachtree St., and 1206 Peachtree St., Apt. A8, Atlanta, Ga.

LASKARIS, Nicholas G. (*J* 1941; *S* 1938) In charge Engrg. Dept., Sears Roebuck & Co., 1246 N. Second St., and (for mail) 759 Aubert Ave., St. Louis, Mo.

LATTERNER, Henry, Jr. (*J* 1940) Sales Engr., McCrea Equipment Co., 516 Second St. N. W., and (for mail) 3600 Macomb St. N. W., Washington, D. C.

LAUCKNER, Charles G., 3rd (*J* 1938) Jr. Engr., General Electric Co., 920 Western Ave., Lynn, and (for mail) 37 Porter St., E. Lynn, Mass.

LAUER, Harold B. (*M* 1930) Vice-Pres. (for mail) English & Lauer, Inc., 1978 S. Los Angeles St., Los Angeles, and 452 S. Spalding Dr., Beverly Hills, Calif.

LAUER, Rodney F. (*A* 1940; *J* 1936) Sales Engr. (for mail) York Ice Machinery Corp., 1238 North 44th St., Philadelphia, and 236 Glentay Rd., Lansdowne, Pa.

LAUFKETTER, Fred C. (*M* 1936) Supt. & Chief Engr. (for mail) Jefferson Hotel, 12th & Locust Sts., and 7056 W. Park Ave., St. Louis, Mo.

LAUTERBACH, Henry, Jr. (*M* 1935) Mech. Engr., in charge of Contract Dept. (for mail) Carrier Corp., Merchandise Mart Bldg., and 6959 Merrill Ave., Chicago, Ill.

LAUTZ, Fritz A. (*M* 1936) Sales Engr., Route No. 13, Box No. 1264, Kirkwood P. O., Des Peres, Mo.

LAVORGNA, Michael L. (*A* 1940) Sales Mgr. (for mail) L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., and 4472 N. Murray Ave., Milwaukee, Wis.

LAWLOR, John J. (*M* 1935) Mgr. Htg. Div., The James Robertson Co., Ltd., 215 Spadina Ave., and (for mail) 35 Tennis Crescent, Toronto, Ont., Canada.

LAWRENCE, Chester T. (*A* 1940) Br. Mgr. (for mail) U. S. Radiator Corp., 901 Washington Ave. S., and 5213 Washburn Ave. S., Minneapolis, Minn.

LAWRENCE, Floyd D. (*A* 1938) Sales Engr. (for mail) Clarage Fan Co., 500 Fifth Ave., New York, and 34-31 81st St., Jackson Heights, L. I., N. Y.

LAWRENCE, L. Frank, Jr. (*J* 1938) Field Engr., Minneapolis-Honeywell Regulator Co., 304-101 Marietta St., and (for mail) 1000 Greenwood Ave. N. E., Atlanta, Ga.

LEBRUN, Paul (*M* 1938) Sales Mgr. (for mail) Chaudieres & Radiateurs "Ideal", 120 Rue Neuve, and 151 Blvd. Brand Whitlock, Brussels, Belgium.

LEDGETT, F. Donald (*J* 1940; *S* 1936) Mech. Engr., Ford Motor Co. of Canada, Ltd., 2591 Danforth Ave., and (for mail) 108 Clinton St., Toronto, Ont., Canada.

LEE, Burton H. (*A* 1940) Customers Service Dept., Consolidated Edison Co. of N. Y., 1812 Carter Ave., and (for mail) 540 Rosedale Ave., New York, N. Y.

LEE, James A. (*A* 1937) Mgr. Contract Dept., Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, and (for mail) 9912 Hubbard, Rosedale Gardens, Plymouth, Mich.

LEE, Robert T. (*J* 1937; *S* 1936) University of North Carolina, and (for mail) 531 E. Franklin St., Raleigh, N. C.

LEEK, Charles W. (*M* 1938) Managing Dir., Leek & Co., Ltd., 1111 Homer St., and (for mail) 4682 W. Sixth Ave., Vancouver, B. C., Canada.

LEEK, Walter (*Life Member*; *M* 1903) Pres. (for mail) Leek & Co., Ltd., 1111 Homer St., and 4769 W. Second Ave., Vancouver, B. C., Canada.

LEFEBVRE, Eugene J. (*M* 1937) Engr., Warden King, Ltd., 2104 Bennett Ave., Montreal, and (for mail) 38 Third St., St. Lambert, Que., Canada.

LEFFEL, Paul C. (*A* 1941) Owner, The Leffel Co., 3323 Main, and (for mail) 316 East 75th St., Kansas City, Mo.

LEGLER, Frederick W. (*M* 1935; *A* 1933) Pres. (for mail) The Waterbury Co., 17 West 28th St., and 2919 Johnson St. N. E., Minneapolis, Minn.

LEHMAN, M. G. (*A* 1937) Owner (for mail) M. G. Lehman, 720 O St., and 2011 Worthington, Lincoln, Nebr.

LEIBY, Robert S. (*A* 1940) Owner (for mail) Robert S. Leiby, 212 N. Grant Ave., Columbus, and 6870 Havens Corners Rd., Blacklick, O.

LEICHNITZ, Robert W. (*J* 1936) Supt., Leichnitz Johnson Co., 112 N. First St., and (for mail) 2506 W. Chestnut, Yakima, Wash.

LEILICH, Roger L. (*M* 1922) Pres. (for mail) Wallace Stebbins Co., 100 S. Charles St., and Garden Apts., Stoney Run Rd. & 40th St., Baltimore, Md.

LEINROTH, J. Paul (*M* 1929) Gen. Ind. Fuel Repr. (for mail) Public Service Electric & Gas Co., 80 Park Pl., Newark, and 37 The Fairway, Montclair, N. J.

LEITCH, Arthur S. (*M* 1908) Pres. and Managing Dir. (for mail) The Arthur S. Leitch Co., Ltd., 1123 Bay St., and 421 Russell Hill Rd., Toronto, Ont., Canada.

LEITGABEL, Kenneth A. (*S* 1939) Apprentice, Experimental Lab., Modine Manufacturing Co., 912 Main St., Racine, and (for mail) 2320 North 58th St., Milwaukee, Wis.

LELAND, Warren B. (*M* 1929) Engr., The H. B. Smith Co., Inc., Westfield, and (for mail) P. O. Box 1522, 159 Sumner Ave., Springfield, Mass.

LELAND, William E. (*M* 1915) Partner (for mail) Leland & Haley, 58 Sutter St., San Francisco, and 704 The Alameda, Berkeley, Calif.

LENIHAN, William O. (*A* 1936) Vice-Pres. (for mail) Laverack & Haines, Inc., White Bldg., and 703 W. Ferry St., Buffalo, N. Y.

LENONE, José M. (*M* 1919) Design Engr. (for mail) Wilson & Co., Inc., 4100 S. Ashland Ave., and 2048 East 69th St., Chicago, Ill.

LEONARD, Lorcan C. G. (*J* 1937) Tech. Mgr. (for mail) Messrs. McCann Jeffreys, Ltd., 19-20 Ellis' Quay, and 265 Clontarf Rd., Dollymount, Dublin, Ireland.

LEONHARD, Lee W. (*M* 1936) Supvr., Eastman Kodak Co., and (for mail) 1075 Winona Blvd., Rochester, N. Y.

LEOPOLD, Charles S. (*M* 1934) Consulting Engr. (for mail) 213 S. Broad St., Philadelphia, and 7600 West Ave., Elkins Park, Pa.

LESER, Frederick A. (*A* 1937) Dist. Mgr. (for mail) Ilg Electric Ventilating Co., 608 Mills Bldg., and 4711 Chesapeake St. N. W., Washington, D. C.

LEUPOLD, George L. (*A* 1937) Sales Engr., Minneapolis-Honeywell Regulator Co., 561 Reading Rd., and (for mail) 1339 Avon Dr., Cincinnati, O.

LEUTHESSER, Fred W., Jr. (*M* 1937) Secy. (for mail) National Metal Products Corp., 21 N. Loomis St., Chicago, and 1715 North 77th Court, River Forest Manor, Ill.

LEVINE, Charles (*J* 1939) Air Cond. Service Engr., Alfred L. Hart, Inc., 315 Vanderbilt Ave., and (for mail) 1171 Ocean Pkwy., Brooklyn, N.Y.

LEVINE, Lawrence J. (*J* 1940) Asst. Engr., Paragon Oil Burner Corp., 75 Bridgewater St., and (for mail) 1378 East 12th St., Brooklyn, N. Y.

LOUGHRAN, Patrick H., Jr. (*J* 1937) Ind. Engr., Washington Gas Light Co., 411 Tenth St. N. W., and (for mail) 4513-49th St. N. W., Washington, D. C.

LOVE, Clarence H. (*M* 1919) Mfrs. Agent, Nash Engineering Co., 317 Chamber of Commerce, and (for mail) 289 Norwalk Ave., Buffalo, N. Y.

LOVING, William H. (*A* 1941; *J* 1936) Asst. Mgr., Govt. Dept. (for mail) Washington Gas Light Co., 411 Tenth St. N. W., Washington, D. C., and McLean, Va.

LOWE, Robert A. (*J* 1938) Engr., Htg. & Vtg. Div., Douglas Aircraft Co., 3000 Ocean Park Blvd., Santa Monica, and (for mail) 4015½ Country Club Dr., Los Angeles, Calif.

LOWE, Walter (*J* 1940; *S* 1938) Sales, Peoples Natural Gas Co., 545 Wm. Penn Way, Pittsburgh, and (for mail) 443 Althea St., Pittsburgh 10, Pa.

LOWNSBERY, Benjamin F. (*M* 1920) Htg. Engr., Benjamin F. Shaw Co., Second and Lombard Sts., and (for mail) 21 S. Sycamore St., Wilmington, Del.

LUCK, Alexander W.* (*Life Member*; *M* 1919) Pres.-Gen. Mgr. (for mail) Reading Heater & Supply Co., Church & Woodward Sts., Reading, and Reiffton, Pa.

LUCKE, Charles E. (*M* 1924) Stevens Prof. Mech. Engrg. (for mail) Columbia University, Pupin Bldg., and 186 Riverside Dr., New York, N. Y.

LUDLOW, Harold M. (*M* 1940) Sales & Engrg. (for mail) P. O. Box 1368, and Sewanee Dr., Jackson, Miss.

LUND, Clarence E.* (*M* 1936; *J* 1935; *S* 1933) Mech. Engr., University of Minnesota, Engrg. Experiment Station, Oak St. Labs., and (for mail) 4817 12th Ave. S., Minneapolis, Minn.

LUNN, Robert J. (*J* 1940) Jr. Apprentice Engr., Donaldson Co., Inc., 666 Pelham Ave., St. Paul, and (for mail) 215 Walnut St. S. E., Apt. 4A, Minneapolis, Minn.

LUTY, Donald J. (*M* 1933) Asst. Gen. Mgr., Air Cond. Div. (for mail) Gar Wood Industries, Inc., 7924 Riopelle St., and 13661 Cloverlawn Ave., Detroit, Mich.

LYFORD, Robert G. (*J* 1939) Branch Mgr. (for mail) The Powers Regulator Co., 1634 Allen Bldg., and 4716 St. Johns Dr., Dallas, Tex.

LYKE, Henry W. (*M* 1938) Gen. Supt., Ames Iron Works, and (for mail) 26 W. Oneida St., Oswego, N. Y.

LYLE, J. Irvine* (*M* 1911) (*Presidential Member*) (Pres., 1917; Council, 1917-18) Pres. (for mail) Carrier Corp., and Orchard Rd., Syracuse, N. Y.

LYMAN, Samuel E. (*A* 1924) Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, N. Y., and (for mail) 865 Hueston St., Union, N. J.

LYNCH, James R. (*A* 1940) Owner (for mail) Lynch Furnace Co., 1804 N. E. Union, and 2952 N. E. Edgehill Pl., Portland, Ore.

LYNCH, William L. (*M* 1928) Pres. (for mail) Rome-Turney Radiator Co., and 1205 N. George St., Rome, N. Y.

LYNN, Frederick E. (*M* 1938) Refrig. Engr., Electric Products Corp., 5624 Penn Ave., Pittsburgh, and (for mail) 312 Moyhend St., Springdale, Pa.

LYNN, Richard G. (*J* 1938) 1716 Marshall Ave., St. Paul, Minn.

LYON, P. S. (*M* 1929) Pres. (for mail) Cochrane Corp., 17th St. & Allegheny Ave., and 3416 Warden Dr., Philadelphia, Pa.

LYONS, Cornelius J. (*A* 1932) Sales Engr. (for mail) Nash Engrg. Co., Wilson Ave., South Norwalk, and 5 Olmstead Pl., East Norwalk, Conn.

M

MABLEY, Louis C. (*M* 1937) Sales (for mail) F. J. Evans Engineering Co., 1110 Watts Bldg., and 36 Pine Crest Rd., Birmingham, Ala.

MABLEY, T. Hollister (*M* 1939) Chief Engr. (for mail) Mechanical Heat & Cold, Inc., 7704 Woodward Ave., Detroit, and 2323 Yorkshire Rd., Birmingham, Mich.

MABON, James E. (*S* 1939) Student, Carnegie Institute of Technology, Pittsburgh, and (for mail) 340 N. Walnut St., Blairsville, Pa.

MACCUBBIN, Howard A. (*M* 1934) Buyer, Htg. Equip., Montgomery Ward & Co., 619 W. Chicago Ave., Chicago, and (for mail) 1511 Colfax St., Evanston, Ill.

MACDONALD, Donald B. (*M* 1930) Sales Engr., Donald B. Macdonald Co., 101 E. Walnut St., Kingston, Pa.

MACDONALD, Douglas J. (*M* 1935) Vice-Pres., Htg. Div. (for mail) Standard Sanitary & Dominion Radiator Co., Ltd., Royce & Lansdowne, and 96 Hudson Dr., Toronto, Ont., Canada.

MacEACHIN, Graham C. (*M* 1938) Dist. Engr., Frigidaire Div., General Motors Sales Corp., 2615 W. Seventh St., and (for mail) 5112 Byers Ave., Fort Worth, Tex.

MacGREGOR, Cecil M. (*A* 1939) Htg. Engr., Portland Gas & Coke Co., Public Service Bldg., and (for mail) 1842 Southeast 41st Ave., Portland, Ore.

MACHEN, James T. (*A* 1938; *J* 1934) Mgr., Sales Agencies (for mail) The Ric-wiL Co., 1562 Union Commerce Bldg., and Sterling Hotel, Cleveland, O.

MACHEREL, Ferdinand (*M* 1939) Mgr. & Owner, INCO, "Industrie et Confort", 50 Rue Daguerre, Algiers, French N. Africa.

MACHIN, Donald W. (*J* 1935) Fuel Engr., Pittsburgh & Midway Coal Mining Co., 816 Dwight Bldg., Kansas City, Mo., and (for mail) 2112 Vermont St., Lawrence, Kans.

MACK, Emil H. (*A* 1938) Asst. Sales Mgr., The Vilter Manufacturing Co., 2217 S. First St., and (for mail) 2225 N. Booth St., Milwaukee, Wis.

MACK, Ludwig (*M* 1935) Dist. Mgr., Cooling & Air Cond. Div., B. F. Sturtevant Co., Cresmont & Haddon Aves., Camden, N. J., and (for mail) 412 W. Hortter St., Germantown, Philadelphia, Pa.

MacLACHLAN, Victor D. (*A* 1939; *J* 1938) Sales Engr., Honeywell-Brown, Ltd., Wadsworth Rd., Perivale, Greenford, Middlesex, England.

MacLEAN, Hector A. (*M* 1939) Prop. & Mgr. (for mail) The MacLean Plumbing Service, P. O. Box 400, and Tremoy Rd., Noranda, Que., Canada.

MacMILLAN, Alexander R. (*M* 1936) Mgr., Detroit Zone, Delco Appliance Div. (for mail) General Motors Sales Corp., 2-160 General Motors Bldg., and 2455 Longfellow Ave., Detroit, Mich.

MACRAE, Robert B. (*A* 1939; *J* 1935) Engr. (for mail) E. J. Nell Co., P. O. Box 1640, Manila, and No. 5 Palm Court, Pasay, Rizal, P. I.

MACROW, Lawrence (*A* 1941; *J* 1936) Branch Chief Engr. (for mail) Carrier Corp., 12 South 12th St., Philadelphia, and Buttonwood Way, Glenside Heights, Pa.

MacWATT, Donald A. (*M* 1938) Sales Engr., Powers Regulator Co., 231 East 46th St., New York, and (for mail) 4611-258th St., Great Neck, L. I., N. Y.

MADDUX, O. Lloyd (*M* 1935; *A* 1933) Owner, O. Lloyd Maddux, 53 Park Pl., New York, N. Y., and (for mail) 17 Tallmadge Ave., Chatham, N.J.

MADELY, Frederick J. (*A* 1936) Chief Estimator, Eastern Steel Products, Ltd., 1335 Delorimier Ave., and (for mail) 6370 Louis-Hemon St., Montreal, Que., Canada.

MADISON, Richard D. (*M* 1926) Research Engr. (for mail) Buffalo Forge Co., 490 Broadway, Buffalo, and 218 Brantwood Rd., Snyder, N. Y.

MAEHLING, Leon S. (*M* 1932) Supt. of Service, Equitable Gas Co., 6304 Penn Ave., and (for mail) 778 Country Club Dr. (16), Pittsburgh, Pa.

MaGEE, Kevin B. (*A* 1938) Cargocaire Engineering Corp., 15 Park Row, New York, N. Y.

MaGIRL, Willis J. (*M* 1934; *A* 1931; *J* 1927) Chief Engr. (for mail) P. H. MaGirl Foundry & Furnace Works, 401-13 E. Oakland Ave., and 1119 E. Monroe St., Bloomington, Ill.

MAGNUSSON, Nicholas (*A* 1938) Estimator-Designer-Sales, Montgomery Ward & Co., 150-15 Jamaica Ave., and (for mail) 138-05 Linden Blvd., Jamaica, L. I., N. Y.

MAHON, B. B. (*M* 1935) Principal, School of Air Conditioning (for mail) c/o Rufus T. Strohm, Dean, International Correspondence Schools, and 433 Fig St., Scranton, Pa.

MAHON, Clarence A. (*A* 1938) Pres.-Mgr. (for mail) Air Control Equipment Co., 1712 Main St., and 6123 Kenwood Ave., Kansas City, Mo.

MAHON, Frank B. (*M* 1937) Ind. Sales Promotion (for mail) Duquesne Light Co., 435 Sixth Ave., and 290 Le Moyne Ave., (16), Pittsburgh, Pa.

MAHONEY, David J. (*M* 1930; *A* 1926) Branch Mgr. (for mail) Johnson Service Co., 503 Franklin St., and 140 Linwood Ave., Buffalo, N. Y.

MAIER, George M. (*M* 1921) Mfg. Dept. (for mail) American Radiator & Standard Sanitary Corp., Bessemer Bldg., Pittsburgh, and 89 Mayfair Dr., Mt. Lebanon, Pa.

MAIER, Herman F. (*M* 1926) Chief Engr. & Secy., The New York Blower Co., 3155 Shields Ave., and (for mail) 7124 S. Morgan St., Chicago, Ill.

MAKIN, Henry T., Jr. (*M* 1939) Engr. & Archts. Repr., American Radiator & Standard Sanitary Corp., 2212 Walnut St., and (for mail) 301 Wadsworth Ave., Mt. Airy, Philadelphia, Pa.

MALIN, Benjamin S. (*M* 1940; *J* 1939) Assoc. Mech. Engr. & Mech. Inspector, Bureau of Agricultural Chemistry & Engrg., U. S. Dept. of Agriculture, and (for mail) 4925 Crescent St., Friendship Station, Washington, D. C.

MALLIS, William (*M* 1914) Archt. (for mail) 330 Lyon Bldg., and 723 Federal Ave., Seattle, Wash.

MALLY, Chester F. (*M* 1940; *A* 1938) Gen. Mgr. (for mail) Mally & Co., 307 Boulevard Bldg., and 2034 Central Ave., Ferndale, Detroit, Mich.

MALONE, Dayle G. (*M* 1929; *A* 1925) Br. Mgr. (for mail) Petroleum Heat & Power Co., 3301 S. California Ave., and 7337 Merrill Ave., Chicago, Ill.

MALONE, James S. (*A* 1936) Dist. Repr. (for mail) Hoffman Specialty Co., 411 N. Tenth St., St. Louis, Mo.

MALVIN, Ray C. (*M* 1920) Pres. (for mail) Malvin & May, Inc., 2015 S. Michigan Ave., and 8220 Dante Ave., Chicago, Ill.

MANDELL, Thomas P. (*A* 1937) Sales, Carrier Corp., 704 Statler Office Bldg., Boston, and (for mail) Walnut Rd., South Hamilton, Mass.

MANK, Merrill (*A* 1939) Owner, Merrill Mank Co., 14 Bonnefoy Pl., and (for mail) 15 North Ave., New Rochelle, N. Y.

MANN, Walter Noah (*M* 1939) Gen. Mgr., Brockhouse Heater Co., Ltd., Victoria Works, West Bromwich, Staffs., and (for mail) "Moneymore" Canwell, Sutton Coldfield, Warwickshire, England.

MANNEN, D. Edward, Jr. (*J* 1939) Vice-Pres., The Mannen & Roth Co., 9108 Woodland Ave., Cleveland, and (for mail) 4157 Silsby Rd., University Heights, O.

MANNING, Charles E. (*J* 1937) Refrig. Sales Engr. (for mail) International Harvester Co., 1413 West 14th St., and 3219 Bellefontaine Ave., Kansas City, Mo.

MANNY, J. Harvey (*A* 1936) Vice-Pres. & Secy. (for mail) Robinson Furnace Co., 213 W. Hubbard St., and 240 N. Parkside Ave., Chicago, Ill.

MARCHIO, Emilio, Jr. (*J* 1940; *S* 1938) Student, Betz Air Conditioning Corp., 1820 Wyandotte, and (for mail) 212 S. Monroe, Kansas City, Mo.

MARCONETT, Vernon G. (*A* 1936) Supt., The Farquhar Furnace Co., and (for mail) 216 Fulton St., Wilmington, O.

MARIN, Axel* (*M* 1935) Assoc. Prof. Mech. Engrg. (for mail) University of Michigan, 241 W. Engineering Bldg., and P. O. Box 175, Ann Arbor, Mich.

MARKERT, John W. (*A* 1940) Assoc. Naval Archt., Htg., Vtg. & Air Cond., U. S. Maritime Commission, Dept. of Commerce Bldg., Washington, D. C., and (for mail) 109 County Rd., Kensington, Md.

MARKLAND, Charles E. (*M* 1939) Mech. Operating Engr. (for mail) University of Illinois, 110 Power Plant, Urbana, and 1117 W. William St., Champaign, Ill.

MARKS, Alexander A. (*A* 1930) Chief Engr., Richmond Radiator Co., 818 Fayette Title & Trust Bldg., Uniontown, Pa.

MARKUSH, Emery U. (*M* 1931) Mech. Engr., Midland Mechanical Installations, Inc., 225 East 21st St., New York, N. Y.

MARRINER, John M. S. (*M* 1934) Taylor Engrg. & Construction Co., Ltd., 80 Richmond St. W., and (for mail) 111½ Balsam Ave., Toronto, Ont., Canada.

MARSCHALL, Peter J. (*M* 1930; *J* 1927) Engr., Kroeschell Engineering Co., 215 W. Ontario St., Chicago, and (for mail) 2009 Greenwood Ave., Wilmette, Ill.

MARSHALL, Albert W. (*M* 1937) Inspector, Hartford Steam Boiler Inspection & Insurance Co., 1806 Arrott Bldg., Pittsburgh, and (for mail) 714 N. St. Clair St., E. E.-Pittsburgh, Pa.

MARSHALL, James (*J* 1939) Engr., The Bahnson Co., 1001 S. Marshall St., and (for mail) A2-121 Twin Castle Apt., Winston-Salem, N. C.

MARSHALL, Orville D. (*A* 1931) Mfrs. Agent (for mail) 514 Anderson Bldg., and 1440 Fisk Rd. S. E., Grand Rapids, Mich.

MARSHALL, Stanley C. (*M* 1939) Chief Engr., Mayflower-Lewis Corp., Duluth & E. Seventh St., St. Paul, and (for mail) 520 Fifth St. S. E., Minneapolis, Minn.

MARSHALL, Thomas A. (*J* 1937) Sales Engr., York Ice Machinery Corp., 1275 Folsom St., San Francisco, Calif.

MARSHALL, William D. (*M* 1935) Br. Mgr. (for mail) Noland Co., Inc., 1823 N. Arlington Ridge Rd., and 1307 N. Wakefield St., Arlington, Va.

MARSTON, Anson D.* (*A* 1937) Ind. Engr. (for mail) Kansas City Power & Light Co., 1330 Baltimore, and 4943 Central, Kansas City, Mo.

MARTENS, Edward D. (*M* 1937) Mech. Engr. (for mail) Thompson Starrett Co., Inc., 444 Madison Ave., New York, and 89 Eldridge Ave., Hempstead, L. I., N. Y.

MARTIN, Albert B. (*M* 1917) Br. Mgr. (for mail) Kewanee Boiler Co., 1858 S. Western Ave., Chicago, and 997 Vine St., Winnetka, Ill.

MARTIN, George W.* (*M* 1911) Supervising Engr. (for mail) U. S. Realty & Improvement Co., 111 Broadway, New York, N. Y., and 340 Prospect St., Ridgewood, N. J.

MARTIN, John O. (*A* 1939) Partner (for mail) J. O. & C. U. Martin, 637 Minna St., San Francisco, and 328 Jerome Ave., Piedmont, Calif.

MARTIN, Raymond (*A* 1937) Sales Mgr., Heal Dept. (for mail) Vapor Car Heating Co. of Canada, Ltd., 65 Dalhousie St., Montreal, and 18 Morris St., Ste. Therese, Que., Canada.

MARTINEZ, Juan J. (*A* 1939; *J* 1929) Research & Rate Engr., The Mexican Light & Power Co., Ltd., Gante 20, and (for mail) Paseo de la Reforma 183, Mexico, D. F., Mexico.

MARTOCELLO, Joseph A. (*M* 1934) Pres., Jos. A. Martocello & Co., 229 North 13th St., Philadelphia, Pa.

MARTY, Edgar O. (*M* 1916) Pennsylvania Turnpike Commission, Harrisburg, and (for mail) 218 North 20th St., Pottsville, Pa.

MARTYN, Henry J. (*A* 1937) Pres. (for mail) Martyn Bros., Inc., 911 Camp St., and 5306 Ridgedale St., Dallas, Tex.

MARZOLF, Frank X. (*A* 1937) Sales Engr., Minneapolis-Honeywell Regulator Co., 415 Brainard St., and (for mail) 15790 St. Marys, Detroit, Mich.

MARZORATI, Giuseppe (*M* 1938) Managing Dir. (for mail) S. i. N. C. Giacomo Jucker, Via Mauro Macchi 28, and Via Baldissera 9, Milan, Italy.

MAST, Clyde M. (*A* 1940) Engr. & Mgr. of Htg. Dept., H. E. Saviers & Son, Inc., W. Second at West St., (for mail) Box 1234, and 536 Nixon St., Reno, Nev.

MATCHETT, James C. (*M* 1923) Vice-Pres. & Gen. Mgr. (for mail) Illinois Engineering Co., S. Racine Ave. & 21st St., and 9936 S. Winchester Ave., Chicago, Ill.

MATHEKA, Charles R. (*S* 1939) Student, New York Technical Inst., 108 Fifth Ave., New York, N. Y., and (for mail) 1506 Summit Ave., Union City, N. J.

MATHER, Harry H. (*A* 1929) Air Cond. Engr. (for mail) Philadelphia Electric Co., 1000 Chestnut St., Philadelphia, and 373 Lakeview Ave., Drexel Hill, Pa.

MATHEWSON, Marvin E. (*M* 1937) Secy. (for mail) A. M. Kinney, Inc., 1301 Enquirer Bldg., and 2156 Alpine Pl., Cincinnati, O.

MATHIS, Eugene* (*M* 1922) Vice-Pres. & Treas., The New York Blower Co., 32nd St. & Shields Ave., Armour P. O. Sta., Chicago, Ill.

MATHIS, Henry (*M* 1921) New York Blower Co., 32nd St. & Shields Ave., Armour P. O. Sta., and (for mail) 10317 Oakley Ave., Chicago, Ill.

MATHIS, John (*A* 1938) Engr., 407 S. Tenth St., Omaha, Nebr.

MATHIS, Julien W. (*A* 1921) New York Blower Co., 32nd & Shields Ave., Chicago, Ill.

MATHISON, Russell S. (*A* 1938) Asst. Mgr. (for mail) Weathermakers (Canada) Ltd., 593 Adelaide St. W., and 44 Strathgowan Ave., Toronto, Ont., Canada.

MATOUSEK, A. G. (*M* 1937) Mgr. (for mail) Gamble Store, and Schuyler, Nebr.

MATTHEWS, John E. (*M* 1934) Dist. Mgr., B. F. Sturtevant Co., 1102 Commerce Trust Bldg., and (for mail) 5642 Lydia St., Kansas City, Mo.

MATTINGLY, Maurice F. (*A* 1939) Sales Engr. (for mail) Johnson Service Co., 1355 Washington Blvd., and 8028 Ingleside Ave., Chicago, Ill.

MATZ, George N. (*M* 1938) Mech. Engr., A. Ernest D'Ambly, 2101 Architects Bldg., Philadelphia, and (for mail) 649 Ferne Ave., Drexel Hill, Pa.

MATZEN, Harry B. (*M* 1919) Consulting Mechanical Engr., 185 Madison Ave., New York, and (for mail) 16 Addison Pl., Rockville Centre, L. I., N. Y.

MAVES, George D. (*A* 1939) Sales Engr. (for mail) Minneapolis-Honeywell Regulator Co., 1305 Capitol, and 2340 Wroxton Rd., Houston, Tex.

MAWBY, Pensyl (*M* 1934) Dist. Sales Mgr., Lehigh Navigation Coal Co., 123 S. Broad St., Philadelphia, and (for mail) 100 Morton Ave., Ridley Park, Pa.

MAXWELL, George W. (*M* 1935; *S* 1932) Engr., Cape Cod Heating & Engineering Co., Lower County Rd., Harwich Port, Mass.

MAXWELL, Robert S. (*M* 1937) Gen. Mgr., Bennett & Wright, Ltd., 72 Queen St. E., Toronto, Ont., Canada.

MAY, Arthur O. (*A* 1938; *J* 1928) Sales Engr. (for mail) Stannard Power Equipment Co., 53 W. Jackson Blvd., and 5736 N. Bernard St., Chicago, Ill.

MAY, Clarence W. (*M* 1933) Consulting Engr. (for mail) 1503 Smith Tower, and 6056 Fourth N. E., Seattle, Wash.

MAY, Edward M. (*M* 1931) Office Mgr. & Combustion Engr., Steel Products Engrg. Co., 1601 S. Michigan Ave., Chicago, and (for mail) 848 N. Ridgeland Ave., Oak Park, Ill.

MAY, George E.* (*M* 1933) Utilization Engr. (for mail) New Orleans Public Service, Inc., 317 Baronne St., and 2031 Short St., New Orleans, La.

MAY, James W. (*M* 1938; *J* 1935) Assoc. Prof., Htg. & Vtg. (for mail) University of Kentucky, College of Engineering, and 261 Lyndhurst Pl., Lexington, Ky.

MAY, Maxwell F. (*M* 1929) Vice-Pres. (for mail) Young Radiator Co., Racine, Wis., and Palos Park, Ill.

MAYETTE, Charles E. (*M* 1926) Sales Engr., E. B. Badger & Sons, Co., 75 Pitts St., and (for mail) Engineers Club, Boston, Mass.

MAYNARD, J. Earle (*M* 1931) Chief Htg. Engr., Sales Engrg. Dept., American Radiator & Standard Sanitary Corp., and (for mail) 324 Fifth St., Elyria, O.

MAYNE, Walter L. (*M* 1938) Vice-Pres. & Sales Mgr. (for mail) Marsh Valve Co., Brigham Rd. at 4th St., and 741 Park Ave., Dunkirk, N. Y.

McBRIDE, J. Nevins (*A* 1941) Officer (for mail) Frank A. McBride Co., Ward St. and Dale Ave., and 231 Wall Ave., Paterson, N. J.

McCAFFERTY, Joseph E. (*A* 1937) Dist. Engr., Petroleum Heat & Power Co., 419 Boylston St., Boston, and (for mail) 747 Front St., Weymouth, Mass.

McCAFFRAY, Charles E. (*M* 1938) Chief Engr. (for mail) McShain-Carrier Air Conditioning Corp., 2801 N. Broad St., Philadelphia, and 10 E. Montgomery Ave., Bala-Cynwyd, Pa.

McCAIN, H. King (*M* 1939; *A* 1938; *J* 1937) Consulting Engr. (for mail) Newcomb & Boyd, 615 Trust Co. of Ga. Bldg., and 28 Old Ivy Rd., Atlanta, Ga.

McCANN, Frank D. (*A* 1939) Sales Supvr. Air Cond. (for mail) Westinghouse Electric & Manufacturing Co., 150 Broadway, New York, and 378 Scarsdale Rd., Crestwood, Yonkers, N. Y.

McCARTHY, John J. (*A* 1937) Htg. & Vtg. Engr. (for mail) Providence Public School Dept., 20 Summer St., and 318 Academy Ave., Providence, R. I.

McCARTHY, Thomas F. (*M* 1938) Sales Engr., Kroeschell Engineering Co., 215 W. Ontario St., and (for mail) 6948 Calumet Ave., Chicago, Ill.

McCAULEY, James H. (*M* 1921) Pres. (for mail) James H. McCauley & Sons, 5620 West 65th St., Chicago, and 707 William St., River Forest, Ill.

McCLANAHAN, Luther C. (*M* 1930) Dist. Mgr. (for mail) Aerofin Corp., 603 Great National Life Bldg., and 811 S. Tyler, Dallas, Tex.

McCLELLAN, James E. (*M* 1922) Dist. Mgr. (for mail) American Blower Corp., 228 N. LaSalle St., Chicago, and 738 Marion Ave., Highland Park, Ill.

McCLINTOCK, William (*M* 1935) Consulting Engr., Board of Education, 34½ East 12th St., and (for mail) 647 East 232nd St., New York, N. Y.

McCLOSKEY, John H. (*A* 1940) Owner, J. H. McCloskey, Plbg. & Htg. Contractor, 109 North St., and (for mail) 304 Elkton Blvd., Elkton, Md.

McCLUNG, Tom H. (*J* 1939) Sales Engr., Brod & McClung, Lewis Bldg., and (for mail) 6626 N. E. Alameda, Portland, Ore.

McCONACHIE, Lorne L. (*A* 1928) Htg. & Plbg. Owner, L. L. McConachie Co., and (for mail) 1415 Harvard Rd., Grosse Pointe Park, Mich.

McCONNER, Charles R. (*A* 1925; *J* 1922) Gen. Sales Mgr. (for mail) Clarage Fan Co., and 1904 Waite Ave., Kalamazoo, Mich.
McCORMACK, Denis (*M* 1933) Mgr. Air Cond. Instruments & Controls Dept., Julien P. Friez & Sons, Inc., 4 N. Central Ave., and (for mail) Ruxton Post Office, Baltimore, Md.
McCOY, C. E. (*M* 1936) Partner (for mail) Turner-McCoy, 315 W. Second St., and 3922 S. Lookout Ave., Little Rock, Ark.
McCOY, Thomas F. (*M* 1924) Mgr. (for mail) The Powers Regulator Co., 125 St. Botolph St., Boston, and Glen Rd., Wellesley Farms, Mass.
McCREA, Joseph B. (*M* 1937) Htg. & Vtg., 2919 Drexel Ave., Detroit, Mich.
McCREA, Lester W. (*M* 1920) Prop. (for mail) McCrea Sales Co., 19 N. Carrollton Ave., and 564 W. University Pkwy., Baltimore, Md.
McCULLOUGH, Henry G. (*M* 1936) Mgr. Air Cond. Dept., S. S. Fretz, Jr., Inc., 1902 Chestnut St., and (for mail) 7042 Lincoln Dr., Mt. Airy, Philadelphia, Pa.
McCULLOUGH, John L. (*M* 1930) Asst. Branch Mgr. (for mail) National Radiator Co., 403 Arrott Bldg., Pittsburgh, and 105 Roycroft Ave., Pittsburgh (16), Pa.
McCUNE, Byron V. (*M* 1928) 807 W. Yakima Ave., Yakima, Wash.
McCUSKER, James P. (*S* 1940) Student, Catholic University, and (for mail) 1445 Evarts St. N. E., Washington, D. C.
McDERMOTT, John P. (*J* 1939) Br. Engr. (for mail) The Trane Co., 310 Postal Bldg., S. W. Third & Washington St., and 3534 S. E. Claybourne, Portland, Ore.
McDONALD, Ivan (*A* 1938) Dist. Repr. (for mail) Minneapolis-Honeywell Regulator Co., Ltd., 44 Princess St., and 132 Kingston Row, Winnipeg, Man., Canada.
McDONALD, Thomas (*A* 1931) Vice-Pres. (for mail) Minneapolis-Honeywell Regulator Co., and 4619 Wooddale Ave., Minneapolis, Minn.
McDONNELL, Everett N. (*M* 1923) (Council, 1940) Pres. (for mail) McDonnell & Miller, 400 N. Michigan Ave., and Drake Hotel, Chicago, Ill.
McDONNELL, John E. (*A* 1936) Sales Engr. (for mail) McDonnell & Miller, 400 N. Michigan Ave., Chicago, and 2299 Lakeside Pl., Highland Park, Ill.
McDOWELL, Harry L. (*J* 1939) Sales Repr., U. S. Radiator Corp., 3403 Blossom St., Columbia, S.C.
McELGIN, John W.* (*A* 1937; *J* 1931) Engr., J. J. Nesbitt, Inc., Holmesburg, Philadelphia, and (for mail) Limekiln & Butler Pikes, Ambler, Pa.
McENTEE, Francis M. (*M* 1940) Asst. Supervising Air Cond. Engr. (for mail) Office of the Architect, U. S. Capitol Bldg., and 718 Somerset Pl. N. W., Washington, D. C.
McGEORGE, Richard H. (*M* 1927) Mgr. Htg. & Air Cond. Dept., McCord Radiator & Manufacturing Co., 2587 E. Grand Blvd., and (for mail) 14565 Glastonbury Rd., Detroit, Mich.
McGINNIS, Frank L. (*M* 1940) Mech. Engr., Williamsburg Restorations, Inc., and (for mail) 332 N. Henry St., Williamsburg, Va.
McGONAGLE, Arthur (*M* 1932) Consulting Engr. (for mail) 1013 Fulton Bldg., Pittsburgh, and 6815 Prospect Ave., Ben Avon, Pa.
McGOWN, Frederick H., Jr. (*J* 1941; *S* 1939) Instructor (for mail) Wyoming Seminary, Kingston, Pa., and P. O. Box 105, Cooperstown, N. Y.
McGRAIL, Thomas E. (*M* 1926) Local Repr., Canadian Sirocco Co., Ltd., 63 Sparks St., Ottawa, Ont., Canada.
McILVAINE, John H.* (*M* 1929) Pres., Landwehr Heating Corp., Sixth and Cayuga Sts., Philadelphia, Pa.
McINDOE, James F. (*M* 1939; *A* 1931) Sales, American Radiator & Standard Sanitary Corp., 1001 Pacific Bldg., and (for mail) 1863 N. W. Aspen St., Portland, Ore.
McINTIRE, James F. (*M* 1915; *A* 1914) (*Presidential Member*) (Pres., 1939; 1st Vice-Pres., 1938; 2nd Vice-Pres., 1937; Council, 1926-28; 1932-40) Vice-Pres. (for mail) U. S. Radiator Corp., 1056-44 Cadillac Square, P. O. Box 686, and 3261 Sherbourne Rd., Detroit, Mich.
McINTOSH, Fabian C. (*M* 1921; *J* 1917) (Council, 1929-31; 1933-35) Branch Mgr. (for mail) Johnson Service Co., 1238 Brighton Rd., and 3650 Perrysville Ave., Pittsburgh, Pa.
McKEE, James W. (*A* 1938) Branch Mgr. (for mail) U. S. Radiator Corp., 439 N. Plankinton Ave., and 6713 W. Bluemound Rd., Milwaukee, Wis.
McKEEMAN, Clyde A.* (*M* 1936) Asst. Prof., Mech. Engrg. (for mail) Case School of Applied Science, Cleveland, and 1359 Lynn Park Dr., Cleveland Heights, O.
McKENZIE, Murdock C., Jr. (*M* 1938) Htg. Engr., Southern California Gas Co., 810 S. Flower St., and (for mail) 3806 Boyce Ave., Los Angeles, Calif.
McKERLIE, Jardine (*M* 1938) Managing Dir. (for mail) Industrial Training Systems, Ltd., 67 Carlton St., and 15 Glen Arden Rd., Toronto, Ont., Canada.
McKINLEY, Carroll B. (*J* 1936; *S* 1934) S. W. Dist. Mgr., General Refrigeration Corp., Beloit, Wis., and (for mail) Box 1482, Albuquerque, N. M.
McKINNEY, Carl A. (*A* 1939; *J* 1937) Air Cond. Engr. (for mail) United Gas Corp., United Gas Bldg., and 1918 Park St., Houston, Tex.
McKINNEY, William J. (*M* 1938; *A* 1934) Dist. Mgr. (for mail) American Blower Corp., Rm. 714, 101 Marietta St. Bldg., and 3363 Mathieson Dr. N. E., Atlanta, Ga.
McKITRICK, Walter D. (*M* 1936) Htg. Vtg. Engr. (for mail) Mills, Rhines, Bellman & Nordhoff, Inc., 518 Jefferson Ave., and 2257 Upton Ave., Toledo, O.
McKITTRICK, Percy A. (*A* 1934) Treas. & Gen. Mgr. (for mail) Parks-Cramer Co., 970 Main St., and 219 Blossom St., Fitchburg, Mass.
McLAREN, T. H. (*A* 1938) Gen. Sales Mgr. (for mail) James Morrison Brass Manufacturing Co., Ltd., 276 King St. W., and 2084 Gerrard St. E., Toronto, Ont., Canada.
McLARNEY, Harry W. (*M* 1933) Air Cond. Engr. (for mail) Union Electric Co. of Missouri, 315 North 12th Blvd., and 5038 Bancroft Ave., St. Louis, Mo.
McLAUGHLIN, Joseph D. (*A* 1930; *J* 1928) Htg. Contractor (for mail) Braley & McLaughlin, 166 Aborn St., and 45 Roslyn Ave., Providence, R. I.
McLEAN, Dermid (*M* 1917) Member of Firm (for mail) Snyder & McLean, Engrs., 2308 Penobscot Bldg., and 12651 Birwood Ave., Detroit, Mich.
McLEISH, William S. (*A* 1932; *J* 1928) Sales Engr., The Ric-wiL Co., New York, and (for mail) 6446-184th St., Flushing, L. I., N. Y.
McLENEGAN, David W.* (*M* 1933) Air Cond. Dept. (for mail) General Electric Co., 5 Lawrence St., Bloomfield, and 73 Arlington Ave., Caldwell, N. J.
McLOUTH, Bruce F. (*M* 1936; *J* 1934) Pres. & Gen. Mgr. (for mail) McLouth Air Cond. Corp., 2400 E. Michigan Ave., Lansing, and 135 Gunson St., East Lansing, Mich.
McMAHON, Thomas W. (*M* 1928) Dist. Mgr. (for mail) American Blower Corp., 1711 Railway Exchange Bldg., St. Louis, Mo.
McMULLEN, Earle W. (*M* 1938) Dir. of Research (for mail) The Eagle Picher Lead Co., C & Porter Sts., and 900 Richmond Rd., Joplin, Mo.
McNAMARA, William (*A* 1930) Sales Mgr. (for mail) The Trane Co., 850 Cromwell Ave., and 1355 Como Ave. W., St. Paul, Minn.
McNAMEE, Earl W. (*M* 1940) Air Cond. Engr. (for mail) B. & J. Jacobs Co., 1729 John St., and 2627 Ocosta Ave., Cincinnati, O.

McNEVIN, Joseph E. (*M* 1937) Mgr. (for mail) Colorado Htg. Co., 650 Cherokee St., and 481 S. Corona, Denver, Colo.

McPHERSON, William A. (*M* 1929) Chief, Htg.-Vtg. Div., Dept. of School Bldgs., 26 Norman St., Boston, and (for mail) 86 Dwinnell St., West Roxbury, Mass.

McQUAID, Dan J. (*M* 1934) Owner (for mail) Dan J. McQuaid Engrg. Service, 614 Cooper Bldg., and 1565 Milwaukee St., Denver, Colo.

McRAE, Malcolm W. (*M* 1939) Research Engr. (for mail) Crane Co., 836 S. Michigan Ave., Chicago, and 816 Fairview Ave., Park Ridge, Ill.

MEAD, Edward A. (*M* 1926) Sales Mgr. (for mail) Nash Engrg. Co., South Norwalk, and 5 Thames St., Norwalk, Conn.

MEAD, Harry K. (*A* 1939) Mfrs. Agent (for mail) 1100 Guardian Bldg., Portland, and Jennings Lodge, Ore.

MEAGHER, Arthur T. (*M* 1938) Dir. & Sales Mgr., Plbg. & Htg. Dept., Wm. Stairs, Son & Morrow, Ltd., 174-190 Lower Water St., and (for mail) 83 Seymour St., Halifax, N. S., Canada.

MEDOW, Jules (*A* 1941; *J* 1937) Liaison Engr., Vega Airplane Co., Burbank, and (for mail) 659 S. Cloverdale Ave., Los Angeles, Calif.

MEHL, Oscar H. (*A* 1941; *J* 1935) Engr. (for mail) Carrier Corp., 710 N. Harwood St., and 5631 Longview St., Dallas, Tex.

MEHMKEN, Herman O. (*A* 1941; *J* 1940) Engr., York Ice Machinery Corp., and (for mail) 400 W. Park, Edwardsville, Ill.

MEHNE, Carl A. (*M* 1929) Htg.-Vtg. Expert, 101 Park Ave., New York, and (for mail) 35 E. Livingston St., Valhalla, N. Y.

MEINHOLTZ, Herbert W. (*M* 1936) (for mail) 608 Mayo Bldg., Tulsa, and 1144½ N. W. 26th St., Oklahoma City, Okla.

MEINKE, Howard G. (*M* 1933) Div. Engr. (for mail) Consolidated Edison Co. of New York, Inc., 4 Irving Pl., Rm. 1500, New York, and 41 Harte St., Baldwin, L. I., N. Y.

MELLON, James T. J. (*M* 1911) (Council, 1915) (for mail) Mellon Co., 4419 Ludlow St., and 431 North 63rd St., Philadelphia, Pa.

MELONEY, Edward J. (*M* 1937) Vice-Pres. (for mail) Bowers Bros. Co., 2015 Sansom St., Philadelphia, and 100 E. Stewart Ave., Lansdowne, Pa.

MENDEN, Peter J. (*M* 1935) Htg. Engr., W. H. Gilcher Co., and (for mail) P. O. Box 762, Fairbanks, Alaska, Until April 1 (for mail) 22 Fifth St., Fond du Lac, Wis.

MENSING, Frederick D. (*M* 1920) (Treas., 1931-32) Consulting Engr., Mensing & Co., 2845 Frankford Ave., Philadelphia, Pa.

MERCER, Charles F. (*M* 1937) Prof. Physics (for mail) University of South Carolina, Physics Dept., and 219 S. Waccamaw, Columbia, S. C.

MERENS, Seymour H. (*A* 1939) Vice-Pres. (for mail) Max Miller & Co., 2720 W. Chicago Ave., and 4955 N. Whipple St., Chicago, Ill.

MERGARDT, Albert P. (*A* 1940) Pres. (for mail) American Heating Engineering Co., Inc., 1005 New York Ave. N. W., Washington, D. C., and 3905 North 5th St., Arlington, Va.

MERRILL, Carle J. (*M* 1919) Treas. (for mail) C. J. Merrill, Inc., 54 St. John St., and 15 Longfellow St., Portland, Me.

MERRILL, Frank A. (*M* 1934) Consulting Engr. (for mail) Office of Hollis French, Cons. Engrs., 210 South St., Boston, and 19 Auburndale Rd., Marblehead, Mass.

MERTZ, Walter A. (*M* 1919) Secy. (for mail) Kehm Bros. Co., 51 E. Grand Ave., and 3753 N. Keeler Ave., Chicago, Ill.

MERWIN, Gile E. (*M* 1924; *J* 1923) Dist. Mgr., The Trane Co., 305 South 51st St., Omaha, Nebr.

MERZ, Robert A. (*S* 1940) Student, Michigan State College, 810 W. Grand River Ave., East Lansing, Mich.

METCALFE, Curtis (*A* 1937) Engr., House Htg. Dept., Michigan Consolidated Gas Co., and (for mail) 14433 Faust, Detroit, Mich.

METZGER, Albert F. (*M* 1940) Supervisor of Steam Utilization (for mail) Allegheny County Steam Heating Co., 435 Sixth Ave., and 3421 Horne St., Pittsburgh, Pa.

METZGER, H. J. (*A* 1937) Pres., Wheeler-Blaney Co., 137 E. Water St., Kalamazoo, Mich.

MEYER, Charles L. (*M* 1930) L. J. Wing Manufacturing Co., 154 West 14th St., New York, and (for mail) American Welfare League, 86-60 Palo Alto Ave., Hollis, L. I., N. Y.

MEYER, Frank L. (*M* 1932; *J* 1928) Vice-Pres., The Meyer Furnace Co., and (for mail) 9 Cole Court, Peoria, Ill.

MEYER, Henry C., Jr.* (*Life Member*; *M* 1898) (Council, 1915-16) Pres. (for mail) Meyer, Strong & Jones, Inc., 101 Park Ave., New York, N. Y., and 25 Highland Ave., Montclair, N. J.

MEYER, Karl A. (*M* 1938) Design Engr., New York Blower Co., 171 Factory St., and (for mail) 109 Woodward Ave., La Porte, Ind.

MICHAELS, Maurice A. (*A* 1939) Owner-Mgr. (for mail) Century Co., 223 S. W. Sixth Ave., and 5239 N. E. Garfield Ave., Portland, Ore.

MICHIE, D. Fraser (*M* 1938; *A* 1930) Htg. Sales Dept. (for mail) Crane, Ltd., 93 Lombard St., and 176 Green Ave., Winnipeg, Man., Canada.

MIDDLETON, David K. (*J* 1936) Engrs. Asst., Michigan Bell Telephone Co., 1365 Cass Ave.. Rm. 1720, and (for mail) 11821 Maiden Ave., Detroit, Mich.

MIDEKE, Joseph M. (*A* 1938) Vice-Pres., Mideke Supply Co., 100 E. Main St., and (for mail) 2505 N. W. 19th St., Oklahoma City, Okla.

MILENER, Eugene D. (*M* 1936) Secy., Ind. Gas Sec., American Gas. Assn., 420 Lexington Ave., New York, N. Y.

MILES, Clarence N. (*A* 1938) Foreman Assembly Dept., Kohlenberger Engineering Corp., 805 S. Spadra Rd., and (for mail) Route 1, Box 174-A, Fullerton, Calif.

MILLARD, Junius W. (*M* 1929) 204 Glendale Ave., Alexandria, Va.

MILLER, Archibald T. (*M* 1938) Mgr. Insulation Sales, The Barrett Co., 40 Rector St., New York, N. Y., and (for mail) 125 Godwin Ave., Ridgewood, N. J.

MILLER, Bruce Rinker (*M* 1930) Asst. Mech. Engr., War Dept., Office of C. Q. M., Ft. Sam Houston, and (for mail) 442 Rigsby Ave., San Antonio, Tex.

MILLER, Charles A. (*A* 1917) Sales, The H. B. Smith Co., Inc., 331 Madison Ave., and (for mail) 2870 Marion Ave., New York, N. Y.

MILLER, Charles W. (*M* 1919; *J* 1908) Pres. (for mail) The Rado Co., 759 N. Milwaukee St., Rm. 405, Milwaukee, and R-1, Box 42, Menomonee Falls, Wis.

MILLER, Edgar R. (*A* 1935) Chief Engr. (for mail) Winnipeg Cold Storage Co., Ltd., Salter & Jarvis Ave., and P. O. Box 1384, Winnipeg, Man., Canada.

MILLER, Floyd A. (*M* 1911) Inspection Engr. (for mail) U. S. Treasury Dept., 377 U. S. Court House, and 944 Montrose Ave., Chicago, Ill.

MILLER, George F. (*M* 1936) Owner (for mail) Geo. F. Miller, Sales Engr., 1625 K St. N. W., Washington, D. C., and 10204 Connecticut Ave., Kensington, Md.

MILLER, Glen (*A* 1937) Htg. Vtg. Engr. (for mail) Southern Counties Gas Co., 810 S. Flower St., Rm. 726, Los Angeles, Calif.

MILLER, Jack E. (*J* 1938) c/o Ellington Miller Co., 25 E. Jackson Blvd., and 7325 Phillips Ave., Chicago, Ill.

MILLER, Jacob (*M* 1936) Pres. (for mail) Hy-Grade Construction Co., Inc., 121 St. Marks Pl., New York, and 20 East 58th St., Brooklyn, N. Y.

MILLER, Leo B. (*M* 1926) Sales Exec. (for mail) Perfex Corp., 500 W. Oklahoma Ave., and 3481 N. Hackett, Milwaukee, Wis.

MILLER, Lorin G.* (*M* 1933) Head, Mech. Engrg. Dept. (for mail) Michigan State College, R. E. Olds Hall of Engrg., and 232 University Dr., East Lansing, Mich.

MILLER, Robert A.* (*M* 1931) Tech. Sales Engr. (for mail) Pittsburgh Plate Glass Co., 2200 Grant Bldg., Pittsburgh, and 1211 Carlisle St., Tarentum, Pa.

MILLER, Robert T. (*A* 1927) Chief Engr. Sales Dept. (for mail) Masonite Corp., 111 W. Washington St., Chicago, and 1412 Schilling St., Chicago Heights, Ill.

MILLER, William T. (*M* 1938) Prof. Htg. & Vtg. (for mail) Purdue University, and 525 Hayes St., West Lafayette, Ind.

MILLHAM, Franklyn B. (*M* 1938) Installation Mgr., S. S. Fretz, Jr., Inc., 1902 Chestnut St., and (for mail) 532 Ellet St., Philadelphia, Pa.

MILLIKEN, J. H.* (*M* 1923) Repr. (for mail) American Air Filter Co., 228 N. LaSalle St., Chicago, and 1021 Ridge Court, Evanston, Ill.

MILLIS, Linn W. (*Life Member*; *M* 1918) Secy.-Treas., Security Manufacturing Co., 1630 Oakland Ave., and (for mail) 3534 Wabash Ave., Kansas City, Mo.

MILLS, D. M. (*A* 1940) Mfrs. Repr. (for mail) 3223 Milam, and 1522 Bonnie Brae, Houston, Tex.

MILLS, Hartzell C. (*A* 1935) Sales, Minneapolis Gas Light Co., 739 Marquette Ave., and (for mail) 4137 Tenth Ave. S., Minneapolis, Minn.

MILNE, Arthur H. (*M* 1938) Dir., Dept. of Bldgs. (for mail) Protestant Board of School Commissioners, 3460 McTavish St., and 4786 Grosvenor Ave., Montreal, Que., Canada.

MILWARD, Robert K. (*A* 1920) Br. Mgr. (for mail) U. S. Radiator Corp., 127 Campbell Ave., and 2441 Calvert Ave., Detroit, Mich.

MINER, H. Harvey (*A* 1940) Partner (for mail) Miner Supply Co., 120 W. Front St., Red Bank, and 94 Silverton Ave., Little Silver, N. J.

MINKLER, William A. (*M* 1940) Asst. Sales Mgr., Young Radiator Co., and (for mail) 1016 Munroe Ave., Racine, Wis.

MIRABILE, J. James (*A* 1938) Installation & Service Mgr. (for mail) Thomas Shipley, Inc., 143 Roosevelt Ave., and 14 Hill St., York, Pa.

MITCHELL, Alva E. (*M* 1939) Mgr., Oil Burner Dept. (for mail) A. P. Woodson Co., 1313 H St. N. W., and 6040-13th Pl. N. W., No. 107, Washington, D. C.

MITCHELL, A. J. (*M* 1938; *J* 1930) Vice-Pres. (for mail) Air Conditioning Co., 1017 Sampson St., and 1936 Dryden Rd., Houston, Tex.

MITCHELL, John A. (*A* 1940; *J* 1938) Owner (for mail) Air Conditioning & Refrigeration Systems, 202 Waterloo Bldg., and Sherwood Park, Waterloo, Ia.

MITCHELL, John G. (*J* 1937; *S* 1936) Sales Engr. (for mail) Fairbanks Morse & Co., 220 East 5th St., St. Paul, and 312 Harvard S. E., Minneapolis, Minn.

MITTENDORFF, Edward M. (*M* 1932) Asst. Engr., Sarco Co., Inc., 222 N. Bank Dr., Chicago, and (for mail) 956 Greenwood Ave., Winnetka, Ill.

MODIANO, Reno N. (*M* 1925) Managing Dir., Carrier Continentale, 4 Rue d'Aguesseau, Paris (8e), and (for mail) 55 Boulevard Beausejour, Paris (16e), France.

MOESEL, F. Albert (*A* 1939) Asst. Mgr. (for mail) W. A. Case & Son Manufacturing Co., 31 Main St., Buffalo, and 382 Argonne Dr., Kenmore, N. Y.

MOFFAT, Ormond G. (*M* 1940; *A* 1937) Application Engr., Canadian Westinghouse Co., Sanford Ave. N., and (for mail) 141 George St., Hamilton, Ont., Canada.

MOHN, H. Leroy (*M* 1937) Chief Engr., Milton Manufacturing Co., and (for mail) 705 Hepburn St., Milton, Pa.

MOHRFELD, Herbert H. (*J* 1935) Vice-Pres. & Engr. (for mail) C. P. Mohrfeld, Inc., 24 Lees Ave., Collingswood, and 131 Chestnut Ave., Haddonfield, N. J.

MOHSIN, Shawki (*J* 1940) Air Cond. Engr., Egyptian Govt. and (for mail) Carrier Corp., C. I. D., Syracuse, N. Y.

MOLFINO, Philip (*M* 1938) Mech. Engr. (for mail) Leland & Haley, 58 Sutter St., and 125 Clayton St., San Francisco, Calif.

MOLLANDER, Eric D. (*A* 1940) Dir., Register & Grille Manufacturing Co., Inc., 70 Berry St., and (for mail) 7002 Sixth Ave., Brooklyn, N. Y.

MOLLENBERG, Harold J. (*M* 1936) Vice-Pres., Mollenberg-Betz Machinery Co., 22 Henry St., Buffalo, and (for mail) 172 Westgate Rd., Kenmore, N. Y.

MOLONEY, Roger R. (*M* 1937) Design Engr., Dept. of Interior, Commonwealth Govt. of Australia, Canberra, F. C. T., and (for mail) 26 Bonner Ave., Manly, Sydney, Australia.

MONICK, Fred R. (*A* 1936) Mgr. (for mail) American Radiator & Standard Sanitary Corp., 605 E. Eighth St., and 1114 S. Sixth Ave., Sioux Falls, S. D.

MONTGOMERY, Edward G. (*A* 1938) Special Repr., Steel Co. of Canada, Ltd., 525 Dominion St., Montreal, and (for mail) 20 Finchley Rd., Hampstead, Que., Canada.

MONTGOMERY, John R. (*A* 1937) Mgr., Window & Door Div., Standards & Research Dept. (for mail) Truscon Steel Co., and 296 Granada Ave., Youngstown, O.

MOODY, Lawrence E. (*M* 1919) Partner (for mail) Moody & Hutchison, 1701 Architects Bldg., Philadelphia, Pa., and 237 Jefferson Ave., Haddonfield, N. J.

MOON, L. Walter (*M* 1915) (Council, 1933-36) Vice-Pres. (for mail) St. Louis Industrial Truck Co., 7700 E. Railroad Ave., and 1137A Hornsby St., St. Louis, Mo.

MOORE, Bryant W. (*A* 1939) Mfrs. Repr., 36 S. W. 3rd Ave., and (for mail) 7827 S. E. 35th Ave., Portland, Ore.

MOORE, Frank C. (*A* 1938) Canadian Mgr. (for mail) Aerofin Corp., 67 Yonge St., and 338 Millwood Rd., Toronto, Ont., Canada.

MOORE, H. Carlton* (*M* 1935) Asst. Prof., Mech. Engrg. (for mail) Massachusetts Institute of Technology, Mechanical Engrg. Dept., Cambridge, and 145 Beaumont Ave., Newtonville, Mass.

MOORE, H. Lee (*M* 1919) (Council, 1927-28) Repr. (for mail) Buffalo Forge Co., 431 Fulton Bldg., and Flaccus Rd., Ben Avon, Pittsburgh, Pa.

MOORE, Henry W. (*M* 1935) Mgr., Air Cond. Engrg. Dept., The Bimel Co., 305 Walnut St., and (for mail) 1406 Myrtle Ave., Cincinnati, O.

MOORE, Herbert S. (*A* 1923) Dist. Repr., Iron Fireman Manufacturing Co. of Canada, Ltd., 602 King St., and (for mail) 107 Clendenan Ave., Toronto, Ont., Canada.

MOORE, MacDonell (*A* 1940) Pres. & Gen. Mgr. (for mail) The Moore Fuel Corp., 23 Rose St., and 14 Fairview Ave., Danbury, Conn.

MOORE, R. Edwin (*A* 1928) Vice-Pres., Bell & Gossett Co., Chicago, and (for mail) 425 Merrill Ave., Park Ridge, Ill.

MOORE, Wesley R. (*M* 1937) Branch Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 4501 Prospect Ave., and 14211 Ashwood Rd., Cleveland, O.

MOREHOUSE, H. Preston (*M* 1933) Gen. Htg. & Air Cond. Repr., Public Service Electric & Gas Co., 80 Park Pl., Newark, N. J.

MOREHOUSE, J. Stanley (*M* 1938) Dean of Engrg. & Prof. Mech. Engrg., Villanova College, Villanova, and (for mail) 102 Llandaff Rd., Upper Darby, Pa.

MORGAN, Arthur S. (*M* 1938) Mgr., Fess Oil Burners of Canada, Ltd., 85 King St. W., and (for mail) 156 Glenmanow Dr., Toronto, Ont., Canada.

MORGAN, Glenn C. (*M* 1911) Partner (for mail) Morgan-Gerrish Co., 84 S. Tenth St., 307 Essex Bldg., and 4308 Fremont Ave. S., Minneapolis, Minn.

MORGAN, Robert C. (*M* 1915) Pres. (for mail) Stewart A. Jellett Co., 1200 Locust St., and 314 W. Seymour St., Philadelphia, Pa.

MORGAN, Robert W. (*M* 1938) Design & Development Engr., Peerless of America, Inc., 515 West 35th St., Chicago, and (for mail) 1347 Walnut St., Western Springs, Ill.

MORIARTY, John M. (*M* 1937) Owner (for mail) Consolidated Heating & Ventilating Co., 1709 West 8th St., Los Angeles, and 1616 Baldwin Ave., Arcadia, Calif.

MORIN, A. R. (*A* 1938) Mgr. Refrigeration Dept., Macklanburg Brass & Copper Products, Inc., 111 N. W. 23rd, and (for mail) 2115 Sherman, Oklahoma City, Okla.

MORRIS, C. Raymond (*M* 1921) Pres., Power & Htg. Equipment Sales, Inc., 14 Burnett Pl., Nutley, N. J.

MORRIS, John A. (*A* 1939; *J* 1936) Htg. Dept., James Robertson Co., Ltd., 946 William St., and (for mail) 4134 Marlowe Ave., Montreal, Que., Canada.

MORRISON, Chester B. (*M* 1931) Managing Dir., York Shipley Ltd., North Circular Rd., London, N. W. 2, England.

MORRISON, Walter B. (*J* 1939) Engr. Htg. Dept., Meier & Frank Co., and (for mail) 1805 N. E. 27th, Portland, Ore.

MORRISON, Wayne L. (*A* 1938) Owner (for mail) W. L. Morrison Co., 1908 Broadway, and P. O. Box 697, Great Bend, Kans.

MORRO, John J. (*M* 1940) Service Engr., Paragon Oil Co., Inc., 75 Bridgewater St., Brooklyn, and (for mail) 28 East 28th St., New York, N. Y.

MORROW, J. DeWitt (*A* 1938) Secy.-Treas. & Mgr. (for mail) The Warren Co., Inc., 614 Walker Ave., and 5503 La Branch, Houston, Tex.

MORSE, Clark T. (*M* 1913) Pres. (for mail) American Blower Corp., 6000 Russell St., and 8120 E. Jefferson, Detroit, Mich.

MORSE, Floyd W. (*A* 1934) Vice-Pres. (for mail) Chamberlin Metal Weather Strip Co., 15 Oak St., and 132 Villa St., Mt. Vernon, N. Y.

MORSE, Louis S., Jr. (*M* 1938; *J* 1936) Air Cond. Sales Engr. (for mail) Westerlin & Campbell Co., 5924 Second Blvd., and 19480 Canterbury Rd., Detroit, Mich.

MORSE, Robert D. (*M* 1936) Mfrs. Repr. (for mail) R. D. Morse Agency, 1534 First Ave. S., and 4316 East 43rd St., Seattle, Wash.

MORTON, Charles H. (*A* 1931) Sales Repr. (for mail) Warren Webster & Co., 228 Ottawa Ave. N.W., and 1106 Sherman St. S.E., Grand Rapids, Mich.

MORTON, Harold S. (*M* 1931) Sales Engr., Sutherland Air Cond. Corp., 385 Minnesota St., St. Paul, and (for mail) 4330 Wooddale Ave., Minneapolis, Minn.

MORTON, Paul S. (*J* 1939) Engrg.-Sales & Installation, 609 Bangor Rd., Lawrence, Mich.

MOSES, Walter B., Jr. (*J* 1940; *S* 1936) Engr., Leo S. Weil & Walter B. Moses, 427 S. Peters St., and (for mail) 8330 Spruce St., New Orleans, La.

MOSHER, Clarence H. (*A* 1919) Owner, C. H. Mosher Co., 423 Ashland Ave., Buffalo, N. Y.

MOTZ, O. Wayne (*M* 1932) Consulting Engr. (for mail) 234 Paramount Bldg., and 2605 Briarcliffe Ave., Cincinnati, O.

MOULD, Delmar E. (*M* 1936) Mgr. (for mail) J. W. Mould & Son, Ltd., 10642-102nd Ave., and 8619-108 A St., Edmonton, Alta., Canada.

MOULDER, Albert W.* (*M* 1917) Vice-Pres. (for mail) Grinnell Co., Inc., 260 W. Exchange St., Providence, and Waterway & Highland Ave., Barrington, R. I.

MUCKLE, James (*M* 1939) Chief Bldg. Supplies Dept. (for mail) Andersen, Meyer & Co., Ltd., 21 Yuen Ming Yuen Rd., and 11 Park Rd., Apt. 8, Shanghai, China.

MUELLER, Harald C. (*M* 1936; *A* 1930) Mgr. Contract Div. (for mail) Powers Regulator Co., 2720 Greenview Ave., Chicago, and 2720 Lawndale Ave., Evanston, Ill.

MUELLER, Harold P. (*M* 1936) Pres.-Treas. (for mail) L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., and 511 E. Monrovia Ave., Milwaukee, Wis.

MUELLER, John E. (*M* 1937) Mgr., Commercial Sales (for mail) West Penn Power Co., 14 Wood St., Pittsburgh, and 570 Cyrstal Dr., Mt. Lebanon, Pa.

MUESSIG, James W. (*M* 1938) Sales Engr. (for mail) Clarage Fan Co., 333 N. Michigan Ave., Chicago, and 313 Edgewood Ave., Lombard, Ill.

MUIRHEID, John G. (*A* 1940; *J* 1937) Sales Engr., Worthington Pump & Machinery Corp., 1621 Queens Rd., Charlotte, N. C.

MULCEY, Paul A.* (*A* 1939; *J* 1938) Asst. Dir., Anthracite Industries Laboratory, Primos, and (for mail) 300 Springfield Rd., Aldan, Del. Co., Pa.

MULLEN, Thomas J., Jr. (*J* 1935) Industrial Mgr., B. F. Sturtevant Co., Western Div., Inc., 602 Wrigley Bldg., Chicago, Ill.

MUMFORD, Albert R. (*M* 1940) Assoc. Dir. of Research (for mail) Consolidated Edison System Cos. of New York, 4 Irving Pl., New York, N. Y., and 107 Palisade Ave., Bogota, N. J.

MUMFORD, William W. (*J* 1940; *S* 1939) Oil Well Improvements Co., and (for mail) 1920 N. Delaware Pl., Tulsa, Okla.

MUNIER, Leon L. (*M* 1919; *J* 1915) Pres. (for mail) Wolff & Munier, Inc., 222 East 41st St., New York, and 63 Columbia Ave., Hartsdale, N. Y.

MUNKELT, Frederick H. (*M* 1938) Vice-Pres., Dorex Div. (for mail) W. B. Connor Engineering Corp., 114 East 32nd St., New York, and 317 East 17th St., Brooklyn, N. Y.

MUNN, E. Fitz (*M* 1935) Archt. (for mail) 1111 McArthur Bldg., Winnipeg, and 65 Berrydale Ave., St. Vital, Man., Canada.

MUNRO, Donald R., Jr. (*J* 1939) Mfrs. Repr. (for mail) D. R. Munro & Son, 112 S. W. Pine St., and 2709 S. W. Buena Vista Dr., Portland, Ore.

MURDOCH, John P. (*M* 1937) Pres. (for mail) John P. Murdoch Co., S. W. Cor. 30th & Oakford Sts., Philadelphia, and 735 Beechwood Dr., Beechwood, Pa.

MURHARD, Erroll A. (*M* 1939) Pres. & Engr. (for mail) Muirhead & Murhard Co., 338 S. W. 9th Ave., and 2136 N. W. Upshur St., Portland, Ore.

MURNIN, Edward A., Jr. (*A* 1938) Supt. of Development & Assembly, Sarco Manufacturing Co., Clewell & Itaska Sts., and (for mail) 802 Broadway, Bethlehem, Pa.

MURPHREE, Robert L. (*A* 1940; *J* 1936) Engr. (for mail) Rogers Plumbing & Heating Co., 2127 Eighth St., and 914-35th Ave., Tuscaloosa, Ala.

MURPHY, Daniel C. (*A* 1940) Mfrs. Repr. (for mail) 214 Old Colony Bldg., and 3900 Grand Ave., Des Moines, Ia.

MURPHY, Edward T.* (*M* 1915) Vice-Pres. in charge of Marketing (for mail) Carrier Corp., and 1055 James St., Syracuse, N. Y.

MURPHY, Howard C.* (*M* 1923) Vice-Pres. (for mail) American Air Filter Co., Inc., 215 Central Ave., and 495 Lightfoot Rd., Louisville, Ky.

MURPHY, Joseph R. (*M* 1934; *A* 1925) Vice-Pres., Taco Heaters, Inc., 342 Madison Ave., New York, N. Y., and (for mail) Terrace Ave., Riverside, Conn.

MURPHY, William W. (*M* 1930) Treas. (for mail) W. W. Murphy Co., 424 Worthington St., and 25 Mansfield St., Springfield, Mass.

MURRAY, Hayward, G. S. (*A* 1941; *J* 1936) Sales Engr., Canadian Comstock Co., Ltd., 80 King St. W., Toronto, Ont., Canada.

MURRAY, John J. (*A* 1933) Sales-Vice-Pres., Pierce Perry Co., Ltd., 236 Congress St., Boston, and (for mail) 60 Commonwealth Park West, Newton Centre, Mass.

MURRAY, Thomas F. (*M* 1923) State Archt., and (for mail) 14 S. Lake Ave., Albany, N. Y.

MURSINNA, Gilbert P. (*A* 1939) Htg. & Air Cond. Contractor (for mail) 411 Poplar St., and 3657 Boudinot Ave., Cincinnati, O.

MUSGRAVE, Merrill N. (*A* 1935) Pres., Harrison Sales Co., Inc., 314 Ninth Ave. N., and (for mail) 1005 E. Roy St., Apt. 13, Seattle, Wash.

MYER, Haydn (*A* 1920) Pres. (for mail) Haydn Myer Co., Inc., 2224 Comer Bldg., and 1411 Avon Circle, Redmont Park, Birmingham, Ala.

MYERS, George W. F. (*M* 1930; *A* 1928; *J* 1923) Myers Engineering Equipment Co., 3736 W. Pine Blvd., St. Louis, and (for mail) 476 Pasadena Ave., Webster Groves, Mo.

MYLER, William M., Jr. (*M* 1937) Chief Engr., Janitrol Div. (for mail) Surface Combustion Corp., 400 Dublin Ave., and 1340 Glenn Ave., Columbus, O.

MYTINGER, Kenneth L. (*M* 1936) Engr., Capitol City Aut. Heating Co., 89 Lexington Ave., and (for mail) 383 Hudson Ave., Albany, N. Y.

N

NACHMAN, George P. (*M* 1938) Treas. (for mail) Spohn Heating & Ventilating Co., 1775 East 45th St., and 2870 Meadowbrook Blvd., Cleveland, O.

NAMAN, Israel A. (*J* 1940) Engr., R. F. Taylor, Cons. Engr., 910 Bankers Mortgage Bldg., and (for mail) 402 Avondale Ave., Houston, Tex.

NAROWETZ, Louis L., Jr. (*M* 1929; *A* 1912) Secy. & Gen. Mgr. (for mail) Narowetz Heating & Ventilating Co., 1711 Maypole Ave., Chicago, and 112 S. Northwest Highway, Park Ridge, Ill.

NASS, Arthur F. (*M* 1927) Pres. (for mail) McGinness Smith & McGinness Co., 527 First Ave., and 29 Elmhurst Rd. (Wabash Station), Pittsburgh, Pa.

NEAL, James P. (*A* 1939) Captain, Post Ordnance Shop Officer, 15th Ordnance Co., First Army, Fort Bragg, N. C.

NEARINGBURG, Arthur (*A* 1938) Sales Engr. (for mail) Sheldons, Ltd., 1221 Bay St., and 130 Floyd Ave., Toronto, Ont., Canada.

NEE, Raymond M. (*M* 1936) Steam Service Engr. (for mail) Boston Edison Co., 39 Boylston St., Boston, and 10 Orkney Rd., Brookline, Mass.

NEILER, Samuel G. (*Life Member*; *M* 1898) Owner (for mail) Neiler Rich & Co., 431 S. Dearborn St., Chicago, and 737 N. Oak Park Ave., Oak Park, Ill.

NELSON, Chester L. (*A* 1937; *J* 1929) Chief Air Cond. Engr., Sears & Piou, 814 S. Vandeventer St., St. Louis, and (for mail) 1015 Nolan Dr., Glendale, St. Louis Co., Mo.

NELSON, D. W.* (*M* 1928) Assoc. Prof., Mech. Engrg. (for mail) College of Engrg., University of Wisconsin, Mech. Engrg. Bldg., and 3906 Council Crest, Madison, Wis.

NELSON, Edwin L. (*A* 1936) Engrg. Dept. (for mail) The Union Ice Co., 1315 E. Seventh St., and 4313 Victoria Ave., Los Angeles, Calif.

NELSON, George O. (*M* 1923) Engr., Carstens Bros., Ackley, Ia.

NELSON, Harold M. (*M* 1937) Pres. (for mail) H. M. Nelson & Co., Inc., 1223 Connecticut Ave., and Rear 2208 Que St. N. W., Washington, D. C.

NELSON, Herman W. (*M* 1909) Pres. & Gen. Mgr. (for mail) The Herman Nelson Corp., 1824 Third Ave., and 2615-12th St., Moline, Ill.

NELSON, Laurence K. (*M* 1940) Assoc. Engr. (for mail) James M. Todd, Consulting Engr., 405 Citizens Bldg., and 2502 Palmer Ave., New Orleans, La.

NELSON, Richard H. (*A* 1933; *J* 1928) Secy.-Treas., The Herman Nelson Corp., 1824 Third Ave., and (for mail) 1303-30th St., Moline, Ill.

NELSON, Roy O. (*M* 1938) Sales Engr. (for mail) C. H. Bevington Co., 600 S. Michigan Ave., Rm. 905, and 5927 N. Rockwell St., Chicago, Ill.

NESBITT, Albert J.* (*M* 1921) Secy.-Treas. (for mail) John J. Nesbitt, Inc., State Rd. and Rhawn St., Philadelphia, and Babylon & Davis Grove Rds., Hatboro, Pa.

NESBITT, John J. (*Life Member*; *M* 1923) Pres. (for mail) John J. Nesbitt, Inc., State Rd. & Rhawn St., Philadelphia, and Welsh Rd. & Tennis Ave., Ambler, Pa.

NESMITH, Oliver E. (*A* 1928) Engr., Williams Oil-O-Matic Heating Corp., Bell & Hanna, and (for mail) 107 Warner, Bloomington, Ill.

NESS, William H. C. (*M* 1931) Gen. Mgr. (for mail) Master Fan Corp., 1323 Channing St., and 215 N. Kingsley Dr., Los Angeles, Calif.

NESSELL, Clarence W. (*M* 1937) Field Application Engr., Minneapolis-Honeywell Regulator Co., 4501 Prospect Ave., Cleveland, O.

NESSI, André (*M* 1930) Ingr. des Arts et Mfrs., Expert pres le Tribunal Civil de la Seine, 1 Avenue du President Wilson, Paris, XVI, France.

NEST, Richard E. (*M* 1936) Asst. Chief Engr., Anchor Post Fence Co., Fluid Heat Div., and (for mail) 5018 Morello Rd., Baltimore, Md.

NEUBAUER, Edwin W. (*M* 1939) Engr. (for mail) Campbell Norquist & Co., 1127 S. W. Morrison St., and 4804 N. E. Davis St., Portland, Ore.

NEWMAN, Harold E. (*M* 1938) Asst. Mgr. (for mail) B. A. Newman Co., P. O. Box 107, and 419 Buckingham, Fresno, Calif.

NEWPORT, Charles F.* (*Life Member*; *M* 1906) Sales Engr., Weil-McLain Co., 641 W. Lake St., and (for mail) 10001 Longwood Dr., Chicago, Ill.

NEWTON, Alwin B.* (*M* 1938) Mgr. Refrig. Div. (for mail) Minneapolis-Honeywell Regulator Co., 2747 Fourth Ave. S., and 18 W. Rustic Lodge Ave., Minneapolis, Minn.

NICHOLLS, John M. (*M* 1939) Htg. Engr., Robbins Gamwell Corp., 68 West St., and (for mail) 17 Buel St., Pittsfield, Mass.

NICHOLLS, P.* (*M* 1920) Supervising Engr. Fuels Section (for mail) U. S. Bureau of Mines, 4800 Forbes St., and 5251 Forbes St., Pittsburgh, Pa.

NICKLE, Arthur J. (*A* 1936) Sales Engr. (for mail) Darling Brothers, Ltd., 140 Prince St., and 4356 Marcil Ave., Montreal, Que., Canada.

NICOLL, Scott F.* (*M* 1939) Air Cond. Mech. Engr. (for mail) York Ice Machinery Corp., and 1433 First Ave., Elmwood, York, Pa.

NICOLS, John A. (*M* 1941) Dist. Mgr. (for mail) B. F. Sturtevant Co., Western Div., Inc., 1217 McKnight Bldg., and 1733 Alpine Pass, Tyrol Hills, Minneapolis, Minn.

NIELSEN, Howard B. (*A* 1939) Mfrs. Repr. (for mail) 409 Couch Bldg., and 5004 N. E. Wisteria Dr., Portland, Ore.

NIESSE, Joe H. (*M* 1938) Dist. Mgr., Ilg Electric Ventilating Co., 836 Architects & Builders Bldg., and (for mail) 5837 Winthrop Ave., Indianapolis, Ind.

NIGHTINGALE, George F. (*A* 1931) Western Sales Mgr. (for mail) Tuttle & Bailey, Inc., 61 W. Kinzie St., Chicago, and 1125 Schneider Ave., Oak Park, Ill.

NININGER, Christian H. (*A* 1938) Sales Engr. (for mail) Frigidaire Div., General Motors Sales Corp., 29 Franklin Rd., and Route 1, Box 198, Roanoke, Va.

NOBBS, Walter W. (*M* 1919) Consulting Engr., 26 Victoria St., London, S. W. 1, and (for mail) 50 Fairhazel Gardens, London, N. W. 6, England.

NOBIS, H. M. (*M* 1914) Owner (for mail) Harry Nobis Co., Union Commerce Bldg., Cleveland, and 1827 Stanwood Rd., East Cleveland, O.

NOBLE, James Paul (*A* 1937) Air Cond. Engr. & Archt., 1960 Beverly Rd., Columbus, O.

NOBLE, Milner (*M* 1940; *A* 1929; *J* 1924) Gen. Mgr., Aerofin Corp., 410 S. Geddes St., Syracuse, N. Y.

NOLAN, James J., Jr. (*M* 1939) Chief Engr., Carrier Div., United Clay Products Co., 931 Investment Bldg., and (for mail) 4024 Calvert St. N. W., Washington, D. C.

NOLL, William F. (*M* 1924) Prop. (for mail) Wm. F. Noll, 629 North 27th St., and 5260 N. Idlewild Ave., Milwaukee, Wis.

NORAIR, Henry (*M* 1938) Pres. (for mail) Norair Engineering Corp., 1124-22nd St. N. W., and 5908-32nd St. N. W., Washington, D. C.

NORBY, Karl H. (*A* 1938) Mgr. Htg. Dept. (for mail) Tacoma Plumbing Supply Co., 315 South 23rd St., and 1316 South 25th St., Tacoma, Wash.

NORDINE, Louis F. (*M* 1914) Office Mgr. (for mail) The Trane Co., 1772 Columbia Rd., Washington, D. C., and 812 Silver Springs Ave., Silver Spring, Md.

NORFOLK, Leslie W. (*A* 1941; *J* 1939) Consulting Engr., 42 Hampden St., Nottingham, England.

NORMAN, Roy A. (*M* 1937) Prof., Mech. Engr., Iowa State College, Mech. Engrg. Dept., and (for mail) 715 Ridgewood Ave., Ames, Ia.

NORRINGTON, Walter L. (*J* 1938) Sales Engr., The V. D. Anderson Co., 1935 West 96th St., Cleveland, and (for mail) 1280 Cranford Ave., Lakewood, O.

NORRIS, William P. (*J* 1938) Sales Engr., Natkin & Co., 3920 Lindell Blvd., and (for mail) 410 N. Newstead Ave., St. Louis, Mo.

NORTH, William R. (*A* 1940) Sales Engr. (for mail) 27 S. Gay St., and 2115 W. Baltimore St., Baltimore, Md.

NORTON, John A. (*M* 1940) Mgr., Htg. Sales Div., Crane, Ltd., 306 Front St. W., Toronto, and (for mail) 71 Donegall Dr., Leaside, Ont., Canada.

NOTTBERG, Gustav (*A* 1933) Vice-Pres. (for mail) U. S. Engineering Co., 914 Campbell St., and 1835 East 68th St. Terrace, Kansas City, Mo.

NOTTBERG, Henry (*M* 1919) Pres. (for mail) U. S. Engineering Co., 914 Campbell St., and 150 West 54th St., Kansas City, Mo.

NOTTBERG, Henry, Jr. (*J* 1937) Secy. (for mail) U. S. Engineering Co., 914 Campbell St., and 150 West 54th St., Kansas City, Mo.

NOVOTNEY, Thomas A. (*M* 1928) Mgr., Convector Div. & Govt. Dept., The National Radiator Co., Johnstown, Pa.

NOWITZKY, Herman S. (*A* 1931) Supt., Construction Maintenance & Repairs, Wilmer & Vincent Corp., 1776 Broadway, New York, N. Y., and (for mail) 821 Llewellyn Ave., Norfolk, Va.

NOYES, Richard R. (*J* 1938) Sales Engr. (for mail) Canadian Sirocco Co., Ltd., 630 Dorchester St. W., and 2010 Mansfield St., Montreal, Que., Canada.

NUSBAUM, Lee* (*M* 1915) Owner (for mail) Pennsylvania Engineering Co., 1119-21 N. Howard St., and 315 Carpenter Lane, Germantown, Philadelphia, Pa.

NUSBAUM, S. Richard (*J* 1940) Mgr. (for mail) Pennsylvania Engineering Co., 1119 N. Howard St., Philadelphia, and 552 Montgomery Ave., Haverford, Pa.

NUTTING, Arthur* (*M* 1940) Chief Engr., American Air Filter Co., 215 Central Ave., Louisville, Ky.

NUTTING, H. George D. (*M* 1938) Consulting Engr., 604 Donovan Bldg., and (for mail) 1461 Calvert Ave., Detroit, Mich.

NYE, L. Bert, Jr. (*J* 1936) Htg. Engr., Washington Gas Light Co., 411 Tenth St. N. W., Washington, D. C., and (for mail) 309 Piedmont St. Arlington, Va.

O

OAKLEY, LeRoy W. (*M* 1937) Owner (for mail) L. W. Oakley Sales Co., 408 W. Clinch Ave., and 2003 Laurel Ave., Knoxville, Tenn.

OAKS, Orion O. (*M* 1917) 119 Oakridge Ave., Summit, N. J.

O'BANNON, Lester S.* (*M* 1928) Research Engr. (for mail) Agricultural Experiment Station, University of Kentucky, and 123 State St., Lexington, Ky.

OBERG, H. C. (*A* 1933) Mgr., Engrg. Dept., Crane Co., Fifth and Broadway, and (for mail) 1362 W. Minnehaha St., St. Paul, Minn.

OBERLIN, James A. (*S* 1940) Engr., Allied Products Plant 3, and (for mail) 211 Union St., Hillsdale, Mich.

OBERSCHULTE, Richard H. (*J* 1938) Sales Engr. (for mail) D. T. Randall & Co., 404 Blvd. Bldg., Detroit, and Franklin, Mich.

O'CONNELL, Presly M. (*M* 1916) Mech. Engr., Austin Co., Sand Point Aviation Field, and (for mail) 5749-31st Ave. N. E., Seattle, Wash.

O'DOWER, Hugh J. (*A* 1938) Sales Engr., Vilter Manufacturing Co., Milwaukee, Wis., and (for mail) 114 W. Tenth St., Kansas City, Mo.

ODUM, Ralph A. (*A* 1939) Supt. Htg. Dept. (for mail) Grover Odum, Plbg. & Htg., 218 Broadway, Daytona Beach, and Holly Hill, Fla.

OELGOETZ, J. F. (*M* 1938) Owner (for mail) J. F. Oelgoetz Co., 3365 N. High St., and 279 E. North Broadway, Columbus, O.

OERTEL, Fritz H. E. (*M* 1939) Consulting Engr., R. F. D. 2, Greensboro, N. C.

OESTERLE, Arthur L. (*M* 1940) Chief Engr. (for mail) Gulf Engineering Co., Inc., 916 S. Peters St., and 3420 Live Oak Pl., New Orleans, La.

OFFEN, Ben (*M* 1928) (for mail) B. Offen & Co., 608 S. Dearborn St., and 3740 Lake Shore Dr., Chicago, Ill.

OFFNER, Alfred J.* (*M* 1922) (Treas., 1935-38; Council, 1935-40) Consulting Engr. (for mail) 139 East 53rd St., New York, and 160-15-11th Ave., Beechhurst, L. I., N. Y.

O'FLAHERTY, John G. (*M* 1937) Chief Engr. (for mail) Unifin Tube Co., 1109 York St., and 290 Central Ave., London, Ont., Canada.

O'GORMAN, John S., Jr. (*A* 1934) Branch Mgr. (for mail) Johnson Service Co., 230 E. Alexandrine Ave., Detroit, and 147 Abbey Rd., Birmingham, Mich.

OKE, William C. (*M* 1938; *J* 1934) Air Cond. Engr. (for mail) Weathermakers Canada Ltd., 593 Adelaide St. W., and 460 Merton St., Toronto, Ont., Canada.

OLD, William H. (*M* 1937) Asst. Mgr. (for mail) Glanz & Killian Co., 1761 W. Forest Ave., Detroit, and 18245 Devonshire Rd., R. F. D. 3, Birmingham, Mich.

OLDES, Willard E. A. E. (*A* 1939; *J* 1936) 650 West 204th St., New York, N. Y.

OLSEN, Carlton F. (*A* 1925; *J* 1920) Engr. & Sales., Kewanee Boiler Corp., 1858 S. Western Ave., and (for mail) 1040 West 104th Pl., Chicago, Ill.

OLSEN, Gustav E. (*M* 1930) Sales Mgr., Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, and (for mail) 68-09 Beach Channel Dr., Arverne, L. I., N. Y.

OLSON, Bernhard (*A* 1929) Pres. (for mail) Barney Olson, Inc., 122 S. Michigan Ave., and 5724 N. Natoma Ave., Chicago, Ill.

OLSON, Gilbert E. (*M* 1930) 8359 Park Pl. Blvd., Houston, Tex.

OLSON, Milton J. (*A* 1941; *J* 1937) Vice-Pres., Olson Bros., 2612 Leavenworth St., and (for mail) 5627 Williams St., Omaha, Nebr.

OLSON, Robert G. (*M* 1923) Eastern Mgr. (for mail) Hydraulic Coupling Div., American Blower Corp., 50 West 40th St., and 22 East 38th St., New York, N. Y.

PATERSON, Frederick C., Jr. (*M* 1936; *J* 1928) Pres., F. C. Paterson & Co., Inc., and (for mail) 70 Stone Ave., Bradford, Pa.

PATORNO, Sullivan A. S. (*M* 1923) Consulting Engr. (for mail) 101 Park Ave., and 312 East 163rd St., New York, N. Y.

PATTERSON, Granville P. (*M* 1939) Engr. & Sales (for mail) W. B. Haggerty, Inc., P. O. Box 2971, and Hotel Mirasol, Tampa, Fla.

PAUL, Donald I. (*M* 1936; *A* 1936; *J* 1932) Chief Engr. (for mail) Gurney Foundry Co., Ltd., 4 Junction Rd., and 408 Bayview Ave., Toronto, Ont., Canada.

PAULEY, Robert D. (*J* 1940) Development Engr. (for mail) Wood Conversion Co., and 310 Avenue E., Cloquet, Minn.

PAULING, Robert E. (*A* 1936) Sales, Illinois Malleable Iron Co., 1801 Diversey Pkwy., Chicago, Ill., and (for mail) 211 S. Gary, Tulsa, Okla.

PAVEY, Charles A. (*M* 1937) Dist. Mgr., B. F. Sturtevant Co., 812 Michigan Bldg., Detroit, Mich.

PAWKETT, Lawrence S. (*A* 1938) Owner (for mail) L. S. Pawkett & Co., Insurance Bldg., and 131 North Drive, San Antonio, Tex.

PEACOCK, Glenn S. (*M* 1939) Htg. Engr., University of Pittsburgh, and (for mail) 111 Elmont St., Twenty-Eighth Ward, Pittsburgh, Pa.

PEACOCK, Herbert (*M* 1930) Dist. Mgr. (for mail) Carrier Corp., 927 Investment Bldg., and 5073 Lowell St., Washington, D. C.

PEART, Allen M. (*A* 1937) Dist. Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 637 Craig W., Rm. 812, and 4635 Melrose, Montreal, Que., Canada.

PECK, Henry E. (*A* 1938) Div. Mgr. (for mail) Delco Appliance Div., General Motors Sales Corp., 840 N. Michigan Ave., Chicago, and Bloomington, Ill.

PEEBLES, John K., Jr. (*A* 1925; *J* 1924) Architectural Engr., Baskerville & Son, Archts., Central National Bank Bldg., and (for mail) 1708 Park Ave., Richmond, Va.

PEISER, Maurice B. (*J* 1937) Sales Engr. (for mail) Natkin & Co., 1729 Howard St., and 22 Carter Lake Club, Omaha, Nebr.

PELLEGRINI, Louis C. (*M* 1939) Vice-Pres., Marlo Coil Co., 6135 Manchester Ave., and (for mail) 6549 Murdoch St., St. Louis, Mo.

PELLER, Leonard (*J* 1934) Consulting Engr., Peller Construction Co., and (for mail) 6808 Wayne Ave., Chicago, Ill.

PELLMOUNTER, Thomas (*A* 1936) Mfrs. Agt., 903 McGee St., and (for mail) 3308 Euclid Ave., Kansas City, Mo.

PELLMOUNTER, Thomas V. (*J* 1938) Engr., Cooling Coil Dept. (for mail) The Trane Co., and 401 South 14th St., LaCrosse, Wis.

PENNEY, Gaylord W. (*M* 1938) Mgr., Electro-Physics Dept., Research Lab. (for mail) Westinghouse Electric & Manufacturing Co., East Pittsburgh, and 171 Orchard Rd., Wilkinsburg, Pa.

PENNOCK, William B. (see Special Service Roll, p. 73).

PERKINS, Robert C. (*A* 1935) Branch Mgr. (for mail) Ilg Electric Ventilating Co., 515 Texas Bank Bldg., and 2803 Burlington, Dallas, Tex.

PERRAS, George E. (*M* 1936) Mgr., Htg. Dept. (for mail) Thomas Robertson & Co., Ltd., 262 Craig St. W., and 5915 Christophe Colomb Ave., Montreal, Que., Canada.

PERSSON, N. Bert (*M* 1937) Consulting Engr., Food Service Equipment Engineering, 1418 Simpson Ave., St. Paul, Minn.

PESTERFIELD, Charles Henry (*M* 1938; *J* 1936; *S* 1932) Asst. Prof., Mech. Engrg. (for mail) Michigan State College, Dept. of Mech. Engrg., and 142 Gunson St., East Lansing, Mich.

PETERSEN, Bruce W. (*J* 1940) Supt., Twin City Furnace Co., 13 S. Third St., and (for mail) 4211 Russell Ave. N., Minneapolis, Minn.

PETERSEN, Christian P. (*A* 1937) Owner (for mail) Petersen Sheet Metal Works, 4120 Cedar Ave., and 3914 Cedar Ave., Minneapolis, Minn.

PETERSEN, Robert H. (*S* 1941) Student, University of Minnesota, and (for mail) 3914 Cedar Ave., Minneapolis, Minn.

PETERSON, Carl M. F.* (*M* 1936) Asst. Prof. Mech. Engrg., Asst. Supt. of Bldgs. & Power (for mail) Massachusetts Institute of Technology, 77 Massachusetts Ave., Cambridge, and 40 Fletcher Rd., Woburn, Mass.

PETERSON, Clarence L. (*M* 1938) Branch Mgr., Minneapolis-Honeywell Regulator Co., 1136 Howard St., San Francisco, and (for mail) 2 Indian Rock Path, Berkeley, Calif.

PETERSON, DuWayne J. (*M* 1940) Dist. Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 45 Allen, and 14 N. Drive, Buffalo, N. Y.

PETERSON, Hans P. (*J* 1939) Air Cond. Engr. (for mail) Bush Manufacturing Co., 100 Wellington St., and 224 S. Whitney St., Hartford, Conn.

PETERSON, J. Raymond (*A* 1941; *J* 1940) Draftsman, Gausman & Moore, E1026 First Natl. Bank Bldg., and (for mail) 719 E. Nevada Ave., St. Paul, Minn.

PETERSON, Neil H. (*M* 1937) Br. Mgr. (for mail) The Trane Co., 1129 Folsom St., and 2744 Green St., San Francisco, Calif.

PETERSON, Sterling Donald (*A* 1930) N. W. Mgr. (for mail) Johnson Service Co., 514 Colmon Bldg., and 5051 Prince St., Seattle, Wash.

PETTIT, Ernest N., Jr. (*M* 1937) Air Cond. Engr. (for mail) United Gas Corp., United Gas Bldg., and 2930 Quenby, Houston, Texas.

PETTY, Charles E. (*A* 1939) Sales Engr. (for mail) U. S. Radiator Corp., P. O. Box 1301, and 2120 Providence Rd., Charlotte, N. C.

PEXTON, Frank S. (*A* 1936) Sales Engr. (for mail) Kansas City Gas Co., 824 Grand, and 43 West 73rd Terrace, Kansas City, Mo.

PFEIFFER, David C. (*M* 1940) Power Sales Engr. (for mail) Dallas Power & Light Co., and 3516 St. Johns, Dallas, Tex.

PFEIFFER, Frank F. (*M* 1938) Engr., Ind. Div., United Engrs. & Constructors, Inc., 1401 Arch St., and (for mail) 7421 Sommers Rd., Philadelphia, Pa.

PFRIEM, Peter G. (*A* 1937) Sales Engr., The Knapp Supply Co., Ohio and Dudley Sts., and (for mail) 211 N. Hackley St., Muncie, Ind.

PFUHLER, John L. (*A* 1925; *J* 1923) Owner (for mail) John L. Pfuhler, Plbg. & Htg., 600 Manor Rd., W. New Brighton, S. I., N. Y.

PHILIP, William (*M* 1937) Sales Engr., Standard Sanitary Dominion Radiator, Ltd., Royce & Lansdowne Aves., and (for mail) 74 Bastedo Ave., Toronto, Ont., Canada.

PHILIPPI, Joseph J. (*M* 1930) Sales Engr. (for mail) Johnson Service Co., 1355 Washington Blvd., and 7807 S. Winchester St., Chicago, Ill.

PHILLIPS, Frederic W. (*M* 1921) Htg. & Vtg. Engr., Queens Borough Gas & Electric Co., 1610 Far Rockaway Blvd., Far Rockaway, L. I., and (for mail) 825 East 38th St., Brooklyn, N. Y.

PHILLIPS, Ralph E. (*M* 1936) Consulting Mech. & Elec. Engr. (for mail) 816 W. Fifth St., and 5153 Angeles Vista Blvd., Los Angeles, Calif.

PHILLIPS, Robert H. (*A* 1941; *J* 1938) Engr. (for mail) Carrier Corp., 1500 S. Santa Fe Ave., and 2343 London St., Los Angeles, Calif.

PHILLIPS, Walter L. (*A* 1938) Mgr. Air Cond. Dept. (for mail) Griffith Consumers Co., 1413 New York Ave. N. W., Washington, D. C., and Falls Church, Va.

PHIPPS, Frederick G. (*M* 1930) Vice-Pres., Preston Phipps, Inc., 955 St. James St., and (for mail) 5431 Earnscliffe Ave., Montreal, Que., Canada.

PIATNITZA, John, Jr. (*S* 1939) Student, 4043 W. Parker Ave., Chicago, Ill.

PICOT, John W. (*A* 1937) Dir., John Taylor & Son (Aust.) Pty., Ltd., 252 George St., Sydney, and (for mail) 19 Marine Parade, Watsons Bay, N. S. W., Australia.

PILLEN, Harry A. (*A* 1933) Owner (for mail) Harry A. Pillen Co., 626 Broadway, and 2124 Crane Ave., Cincinnati, O.

PINES, Sidney (*M* 1920) Gen. Mgr. (for mail) Pines-Natkin Co., 2413 N. Pearl St., and 4441 Livingston Ave., Dallas, Tex.

PISTLER, Willard C. (*M* 1934) Consulting Engr., 61 Leverone Bldg., 4 W. Seventh St., and (for mail) N. W. Corner Orchard Lane and Crestview Ave., Cincinnati, O.

PITCHER, Lester James (*M* 1929; *A* 1928; *J* 1924) Electrimatic Corp., 2100 Indiana Ave., and (for mail) 1224 East 69th St., Chicago, Ill.

PLACE, Clyde R. (*M* 1924) Consulting Engr. (for mail) 420 Lexington Ave., and 333 East 57th St., New York, N. Y.

PLANT, Edward B. (*M* 1938) Asst. Engr. (for mail) Canadian Pacific Railway Co., Rm. 401, Windsor St. Sta., and 2 Thurlow Rd., Hampstead, Montreal, Que., Canada.

PLATZ, John F. (*A* 1940) Sales Engr. (for mail) J. M. & L. A. Osborn Co., 1541 East 38th St., and 3364 East 135th St., Cleveland, O.

PLAYFAIR, George A. (*A* 1924) Mgr. (for mail) Johnson Temperature Regulating Co. of Canada, Ltd., 113 Simcoe St., Toronto, and 19 Highland Cr., West Hill, Ont., Canada.

PLEUTHNER, Richard Louis (*J* 1938) Mech. Engr., Buffalo Forge Co., and (for mail) 393 Starin Ave., Buffalo, N. Y.

PLEWES, Stanley E. (*M* 1917) Branch Mgr. (for mail) Johnson Service Co., 2853 North 12th St., and 341 E. Hortter St., Philadelphia, Pa.

PLOSKEY, Edward J. (*J* 1940) Engr., Redwood Manufacturers Co., 1600 Hobart Bldg., and (for mail) 2307-32nd Ave., San Francisco, Calif.

PLUM, Leroy H. (*M* 1935; *A* 1934) Engr. (for mail) Warren Webster & Co., 17th & Federal Sts., Camden, and Home Acres Farm, Medford, N. J.

PODOLSKE, Arthur R. (*A* 1938) Prop. (for mail) Arthur R. Podolske Sheet Metal Works, 818A E. Center St., and 820 E. Center St., Milwaukee, Wis.

POEHNER, Robert E. (*M* 1928) Prop. (for mail) R. E. Poehner, Htg. Contractor, 849 Massachusetts Ave., and 2308 Coyner Ave., Indianapolis, Ind.

POGALIES, Louis H. (*M* 1931) Mech. Engr., Wilbur Watson & Associates, 4614 Prospect Ave., and (for mail) 4102 Archwood Ave., Cleveland, O.

POHLE, Kenneth F. (*A* 1930) Vice-Pres. (for mail) W. F. Hirschman Co., Inc., 143 Federal St., Boston, and 172 Hamilton Ave., Quincy, Mass.

POLLAK, Rudolf (*M* 1937) Chief Engr. (for mail) Rockefeller Center, Inc., 50 Rockefeller Plaza, New York, and Larchmont, N. Y.

POLLARD, Alfred L. (*A* 1932) Gen. Supt., Puget Sound Power & Light Co., 860 Stuart Bldg., Seattle, Wash.

POLLOCK, Carl A. (*A* 1937) Vice-Pres. & Gen. Mgr. (for mail) Dominion Electrohome Industries, Ltd., 39 Edward St., and 120 Sterling Ave., Kitchener, Ont., Canada.

POND, William H. (*M* 1938) 820 W. Front St., Plainfield, N. J.

PONDER, Everett A. (*A* 1939) Owner, Everett A. Ponder, Oil Burner Distributor, 3118 Northeast 70th St., Portland, Ore.

POPE, S. Austin (*M* 1917) Pres. (for mail) William A. Pope Co., 26 N. Jefferson St., Chicago, and 831 Ashland Ave., River Forest, Ill.

PORTER, Carl W. (*A* 1940; *J* 1936) Engr. (for mail) Richards & Porter, 42 W. Concord Ave., and 915 Bradshaw Terrace, Orlando, Fla.

PORTER, Knight C. (*M* 1940) Air Cond. Engr. (for mail) Commonwealth Edison Co., 72 W. Adams St., Chicago, and 144 Linden Ave., Glencoe, Ill.

PORTER, Noel E. (*J* 1938) Application Engr., General Air Conditioning Co., 1313 J St., and (for mail) 2600 M St., Sacramento, Calif.

POSEY, James (*M* 1919) Consulting Engr. (for mail) 10 East Pleasant St., and 4005 Liberty Heights Ave., Baltimore, Md.

POTTER, John R. (*A* 1939; *J* 1938) Design Engr., Lockwood-Green, Engrs., 2 Rockefeller Plaza New York, and (for mail) 2 Grace Court, Brooklyn, N. Y.

POUGHER, Bernard R. E. (*J* 1940) Engr., Ernest W. Pougher & Son, Htg. & Vtg. Engrs., Old Trafford, Manchester, and (for mail) 99 Mauldeth Rd. W., Withington, Manchester 20, England.

POUGHER, Ernest W. (*M* 1939) Dir., Engr., Ernest W. Pougher & Son, Engrs., Old Trafford, and (for mail) 99 Mauldeth Rd. W., Manchester, England.

POWELL, George W., Jr. (*M* 1938) Consulting Mech. Engr. (for mail) Otis Bldg., 16th & Sansom Sts., Philadelphia, and 458 S. Fourth St., Darby, Pa.

POWERS, Earle C. (*M* 1939) Partner (for mail) E. C. Powers & Son, 240 Cherry St., Philadelphia, Pa., and 22 E. Stiles Ave., Collingswood, N. J.

POWERS, Edgar C. (*A* 1934; *J* 1931) Partner (for mail) E. C. Powers & Son, 240-242 Cherry St., Philadelphia, Pa., and 35 Madison Ave., Erlton, N. J.

POWERS, F. W. (*Life Member*; *M* 1911) Pres. (for mail) The Powers Regulator Co., 2720 Greenview Ave., and 900 Castlewood Terrace, Chicago, Ill.

POWERS, Lowell G. (*A* 1937; *J* 1930) Br. Mgr. (for mail) Carrier Corp., 795 Union Commerce Bldg., Cleveland, and 2730 Cranlyn Rd., Shaker Heights, O.

PRATT, Foster J. (*M* 1937) Marine Engr., U. S. Navy Yard, Puget Sound, Bremerton, and (for mail) Annapolis Terrace, Port Orchard, Wash.

PRAWL, Frank E. (*M* 1940; *J* 1936) Owner & Engr. (for mail) Prawl Engineering Co., P. O. Box 844, and 401 East 24th, Scottsbluff, Nebr.

PREBENSEN, Harold J. (*M* 1938) Vice-Pres. (for mail) Air Comfort Corp., 1307 S. Michigan Ave., Chicago, and Meadow Lark Rd., Northfield, Ill.

PRENTICE, Oliver J. (*A* 1927) Dir. of Publicity & Public Relations (for mail) C. A. Dunham Co., 450 E. Ohio St., and 850 Lake Shore Dr., Chicago, Ill.

PREWITT, H. B. (*A* 1939) Asst. Branch Mgr. (for mail) American Blower Corp., Suburban Station Bldg., and 615 E. Allen Lane, Philadelphia, Pa.

PRICE, Charles E. (*A* 1933) Treas. (for mail) Keeney Publishing Co., 6 N. Michigan Ave., Chicago, and 800 Vernon Ave., Glencoe, Ill.

PRICE, Charles F. (*J* 1937) Htg. & Air Cond. Engr., The Knapp Supply Co., Ohio Ave. & Dudley St., and (for mail) 1015 W. Washington St., Muncie, Ind.

PRICE, D. O. (*M* 1934) Htg. & Air Cond. Engr., General Steel Wares, Ltd., 199 River St., and (for mail) 131 St. Germain Ave., Toronto, Ont., Canada.

PRICE, Ernest H. (see Special Service Roll, p. 73).

PRIESTER, Gayle B. (*J* 1935; *S* 1934) Sales Engr., Carrier Corp., 3629 Third Ave. S., Minneapolis, Minn.

PRINCE, Raymond F. (*J* 1936) Htg. & Sales Engr., R. B. Dunning & Co., 54 Broad St., and (for mail) 27 McKinley St., Bangor, Me.

PROEBSTLE, Leonard (*J* 1938) Air Cond. Engr., Belden-Porter Co., 65 North 17th St., and (for mail) 4424–34th Ave. S., Minneapolis, Minn.

PROIE, John (*M* 1936) Gen. Mgr. (for mail) Proie Brothers, 856 W. North Ave., and 101 Dilworth St., Pittsburgh, Pa.

PROSSER, Robert G. (*J* 1940) Sales Engr., The Trane Co., 115 N. Eighth Ave. E., Duluth Minn.

PRYIBIL, Paul L. (*A* 1932) Partner, Hucker-Pryibil Co., 1700 Walnut St., and (for mail) 328 E. Phil-Ellena St., Philadelphia, Pa.
PRYKE, John K. M. (see Special Service Roll, p. 73).
PUGH, Daniel C. (*S* 1939) Engrg. Draftsman, Carbide & Carbon Chemicals Corp., S. Charleston, and (for mail) 9 McFarland St., Charleston, W. Va.
PULLUM, Clarence E. (*M* 1940) Vice-Pres., Bell & Gossett Co., 3000 S. Wallace, Chicago, and (for mail) 1011 N. Grove Ave., Oak Park, Ill.
PULLEN, Royal R. (*M* 1935) Chief Mech. Engr., Homestake Mining Co., and (for mail) 109 East Hill St., Lead, S. D.
PURCELL, Frederick C. (*M* 1926) Sales Engr., Minneapolis-Honeywell Regulator Co., 415 Brainard St., and (for mail) 18680 Santa Rosa Dr., Detroit, Mich.
PURINTON, Dexter J. (*A* 1923) 101 Park Ave., and (for mail) 157 East 39th St., New York, N. Y.
PURINTON, Ivan H. (*M* 1940) New Orleans Public Service, Inc., 317 Baronne St., New Orleans, La.

Q

QUACKENBUSH, Seelye M. (*M* 1940) Owner (for mail) Quackenbush Co., 597 Michigan Ave., and 251 Parkside Ave., Buffalo, N. Y.
QUALL, Clarence O. (*A* 1937) Owner, Quall Plumbing & Heating Co., 65 Ninth St., and (for mail) 54 Pearl St., Clintonville, Wis.
QUEER, Elmer Roy* (*M* 1933) Asst. Prof., Engrg. Research (for mail) Pennsylvania State College, Engrg. Experiment Station, and 338 Arbor Way, State College, Pa.
QUICK, Blair A. (*A* 1938) Sales Mgr., Independent Register Co., 3747 East 93rd St., and (for mail) Fenway Hall, Cleveland, O.
QUIRK, Clinton H. (*M* 1916; *J* 1915) Eastern Repr. (for mail) The Trane Co., 250 East 43rd St., New York, and 465 Front St., Hempstead, L. I., N. Y.

R

RABE, Albert E. (*M* 1938) Dir. (for mail) Carrier Engineering, South Africa, Ltd., P. O. Box 7821, and 152 Mowbray Rd., Greenside Ext., Johannesburg, South Africa.
RABER, Benedict F. (*M* 1937) Prof. Mech. Engrg. (for mail) University of California, Rm. 114 Engrg. Bldg., and 1124 Arch St., Berkeley, Calif.
RACHAL, John M. (*A* 1936; *J* 1930) Mgr. Air Cond. Dept. (for mail) Volkart Bros., 8 Clive St., and 1 Bishop Lefroy Rd., Calcutta, India.
RAINE, John J. (*M* 1912) Vice-Pres. (for mail) G. S. Blodgett Co., 190 Bank St., Burlington, and Essex Junction, Vt.
RAINGER, Wallace F. (*A* 1930; *J* 1924) Jaros, Baum & Bolles, 415 Lexington Ave., New York, and (for mail) 441 Hawthorne Ave., Yonkers, N. Y.
RAINSON, Samuel (*J* 1940) Installation & Service Air Cond. Technician, Nassau Air Conditioning Co., Inc., 27 Haven Ave., and (for mail) 50 Highland Ave., Port Washington, L. I., N. Y.
RAISLER, Robert K. (*A* 1933; *J* 1930) Treas. (for mail) Raisler Corp., 129 Amsterdam Ave., and 38 East 85th St., New York, N. Y.
RALPH, David S. (*J* 1938) Engr. (for mail) The L. R. Krumm Co., 121 E. Gay St. Columbus, and 620 Kenilworth Ave., Dayton, O.
RAMSEUR, Vardry Dixon, Jr. (*J* 1940) Htg. Engr., Ramseur Roofing Co., Inc., (for mail) P. O. Box 331, and Woodvale Ave., Greenville, S. C.
RAND, Fred R. (*M* 1938) Htg. Engr. & Sales Mgr., Enamel & Heating Products, Ltd., and (for mail) Squire St., Sackville, N. B., Canada.
RANDALL, Robert D. (*A* 1930) Partner (for mail) D. T. Randall & Co., 404 Blvd. Bldg., and 340 E. Grand Blvd., Detroit, Mich.
RANDALL, W. Clifton* (*M* 1928) Chief Engr. (for mail) Detroit Steel Products Co., 2250 E. Grand Blvd., Detroit., and 770 Shirley Dr., Birmingham, Mich.
RANDOLPH, Charles H. (*M* 1930; *A* 1928; *J* 1926) Air Cond. Engr., Wisconsin Electric Power Co., 231 W. Michigan St., and (for mail) 1614 East Royall Pl., Milwaukee, Wis.
RANDOLPH, Harold F. (*M* 1940) Vice-Pres. (for mail) International Heater Co., 101 Park Ave., and 12 Woodlawn Ave. E., Utica, N. Y.
RANZINGER, Gustav (*S* 1939) 147-05 Willets Pt. Blvd., Whitestone, L. I., N. Y.
RATHER, Maxwell F. (*M* 1919) Mgr. Eastern Dist. (for mail) Johnson Service Co., 28 East 29th St., New York, N. Y., and 90 Orchard Dr., Greenwich, Conn.
RAVEN, Andrew H. (*M* 1938) Htg. Engr., Wigman Co., 316 Perry St., and (for mail) 2119 George St., Sioux City, Ia.
RAY, George E. (*J* 1939) Engr., Wise Furnace Co., 100 Lincoln St., Akron, and (for mail) 1520 Front St., Cuyahoga Falls, O.
RAY, Lewis B. (*M* 1932) Pres. (for mail) Ray Engineering Co., Inc., 830 Broad St., Newark, and 151 Augusta St., Irvington, N. J.
RAYMER, William F., Jr. (*A* 1936; *J* 1934) Sales Engr. (for mail) American Blower Corp., 249 High St., Newark, and 18 Bradley Terrace, West Orange, N. J.
RAYMOND, Fred I.* (*A* 1929) Owner, F. I. Raymond Co., 629 W. Washington Blvd., Chicago, Ill.
RAYNIS, Theodore (*A* 1939; *J* 1934) Assoc. Naval Archt., New York Navy Yard, Brooklyn, and (for mail) 58 Hilltop Dr., Manhasset, L. I., N. Y.
READER, Joseph T. (*A* 1938) Partner (for mail) Kerr Machinery Co., 608 Kerr Bldg., Detroit, and 263 Roosevelt Pl., Grosse Pointe, Mich.
RECK, William E. (*M* 1927) Civil Engr. (for mail) The Reck Heating Co., Ltd., Esromgade 15, Copenhagen, and Sundvej 16, Hellerup, Denmark.
REDRUP, Will D. (*M* 1936) Pres. (for mail) The Majestic Co., and 310 Randolph St., Huntington, Ind.
REDSTONE, Arthur L. (*M* 1931) Research Engr., Proctor & Schwartz, Seventh & Tabor Rd., and (for mail) 1636 E. Duval, Philadelphia, Pa.
REED, Frederick Jerome (*M* 1939) Asst. Prof. Mech. Engrg. (for mail) Duke University, 263 College Station, and 2203 Englewood Ave., Durham, N. C.
REED, Van A., Jr. (*M* 1930) Mech. Engr. (for mail) Federal Engineering Co., 239 Fourth Ave., Pittsburgh, and 114 Water St., Elizabeth, Pa.
REED, Virgil C. (*M* 1938) Owner (for mail) James H. Pinkerton Co., 640 Natoma St., and 1234 Second Ave., San Francisco, Calif.
REED, William H., III (*A* 1938) Mgr. Carrier Dept. (for mail) Dravo Corp., 302 Penn Ave., and 5675 Beacon St., Pittsburgh, Pa.
REGER, Henry P. (*M* 1934) Pres. (for mail) H. P. Reger & Co., 1501 East 72nd Pl., and 6939 Bennett Ave., Chicago, Ill.
REID, Henry P. (*M* 1931; *A* 1927) Operating Engr., Universal Atlas Cement Co., 135 East 42nd St., New York, N. Y.
REID, Herbert F. (*A* 1932) Partner, Reid-Graff Co., 1417 Peck St., Muskegon Heights, Mich.
REIF, Allan F. (*M* 1937) Pres. (for mail) Reif-Rexoil, Inc., 37-43 Carroll St., Buffalo, and 10 Livingston Pkwy., Snyder, N. Y.
REIF, Charles A. (*M* 1937) Vice-Pres., Treas. (for mail) Reif-Rexoil, Inc., 37-41 Carroll St., Buffalo, and 77 Ruskin Rd., Eggertsville, N. Y.
REIFSCHNEIDER, Jake (*A* 1938) Chief Engr. (for mail) Eppley Hotels Co., 1802 Dodge St., and 4409 Poppleton, Omaha, Nebr.
REILLY, Bertram B. (*J* 1938) Engr., Dravo Corp., 300 Penn Ave., Pittsburgh, and (for mail) 400 Ridgewood Ave., West View, Pa.

REILLY, J. Harry (*M* 1931; *J* 1929) Sales Engr., American Radiator Co., 528 Ferry St., Newark, and (for mail) 14 Watson Ave., East Orange, N.J.

REINKE, Alfred G. (*A* 1940; *J* 1933) Secy., Gus Reinke Machinery & Tool Co., 63 Dickerson St., Newark, and (for mail) 321 Park Pl., Irvington, N. J.

REINKE, Louis F. (*A* 1937) Owner (for mail) Reinke Sheet Metal Works, 534 S. Fifth St., and 1535 W. Walker St., Milwaukee, Wis.

REIS, Robert (*J* 1939) Htg. Vtg. & Air Cond. Engr., Reis & O'Donovan, Inc., 12 West 21st St., and (for mail) 155 East 47th St., New York, N.Y.

REISBERG, Lester K. (*A* 1939) Vice-Pres. (for mail) Goodin Co., 707 N. Third St., and 2724 N. E. Hayes St., Minneapolis, Minn.

RENOUF, E. Prince (*M* 1933) Air Cond. Supvr., Westinghouse Electric & Manufacturing Co., 1005 Insurance Bldg., and (for mail) 3427 Rankin, Dallas, Tex.

REPP, Harry L. (*M* 1922) Br. Mgr. (for mail) U. S. Radiator Corp., 3221 Carnegie Ave., and 15612 Braemar Dr., Cleveland, O.

RESCH, Roy J. (*A* 1940) Pres. (for mail) McQuay, Inc., 1600 Broadway N. E., and Minneapolis Athletic Club, Minneapolis, Minn.

RESS, Otto J. (*A* 1940; *J* 1938) Gas Htg. Engr. (for mail) Iowa-Nebraska Light & Power Co., 1401 O St., and 3408 South 29, Lincoln, Nebr.

RETTEW, Harvey F. (*M* 1929) Chief Engr., Board of Education, 21st & Parkway, and (for mail) 4413 Walnut St., Philadelphia, Pa.

REYNOLDS, Thurlow W. (*M* 1922) Consulting Engr., 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

REYNOLDS, Walter V. (*A* 1928) Pres. (for mail) Walter Reynolds, Inc., 861 Third Ave., and 444 East 52nd St., New York, N. Y.

RHINE, George R. (*A* 1938) Capt., 111 Ordnance Co., 36 Div., and (for mail) 1804 Vincent St., Brownwood, Tex.

RHOTON, Walter R. (*M* 1936) Pres. (for mail) The W. R. Rhoton Co., 5915 Bonna Ave., Cleveland, and 1728 Lee Rd., Cleveland Heights, O.

RICE, Robert B. (*M* 1934) Prof. of Experimental Engrg. (for mail) State College of Argiculture & Engineering, University of North Carolina, Mech. Engrg. Dept., and 2002 White Oak Rd., Raleigh, N. C.

RICHARD, Edwin J. (*M* 1933) Owner (for mail) Richard Equipment Co., 2137 Reading Rd., and 3147 Victoria Ave., Cincinnati, O.

RICHARDS, Guy H. (*A* 1939) Mgr. Htg. Dept., Crane Co., 2 South 20th St., and (for mail) 420 North 78th St., Birmingham, Ala.

RICHARDS, Leslie V. (*A* 1940) Owner-Mgr., Richards Oil Burner Sales-Service, 97 Lawrence St., Malden, Mass.

RICHARDSON, Henry G. (*M* 1934) Htg. & Air Cond., Williams & Richardson, 204 Dooly Bldg., and (for mail) 1433 Harvard Ave., Salt Lake City, Utah.

RICHARDSON, Laurence S. (*A* 1939) Mgr. & Engr., Htg. Dept., Stark-Davis Co., 110 S. W. Front St., and (for mail) 6929 S. E. Yamhill St., Portland, Ore.

RICHARDSON, R. Donald (*J* 1938) Htg. Engr., Hope's Heating & Lighting, Ltd., Smethwick, and (for mail) 85 Silhill Hall Rd., Solihull, Birmingham, England.

RICHFIELD, Nicholas H. (*M* 1937) Engr., Oil Burner Div., American Radiator & Standard Sanitary Corp., 40 West 40th St., New York, and (for mail) 173 N. Tyson Ave., Floral Park, L. I., N. Y.

RIDLEY, Walter H. (*M* 1939) Mgr., Dryer Div., Riggs & Lombard, Inc., Suffolk St., Lowell, and (for mail) High St., Chelmsford, Mass.

RIES, Lester S. (*M* 1929) Supt. Dept. of Bldgs. & Grounds (for mail) Oberlin College, 32 E. College St., and 291 Oak St., Oberlin, O.

RIESMEYER, Edward H., Jr. (*A* 1936; *J* 1930) Htg. Engr., Schaffer Heating Co., 231-33 Water St., and (for mail) 4702 Stanton Ave., Pittsburgh, Pa.

RIETZ, Elmer W.* (*M* 1923) Mgr., Specialty Div. (for mail) The Powers Regulator Co., 2720 Greenview Ave., Chicago, and 2250 S. Sheridan Rd., Highland Park, Ill.

RITCHIE, Alexander G. (*M* 1933) Pres. (for mail) John Ritchie, Ltd., 102 Adelaide St. E., and 41 Garfield Ave., Toronto, Ont., Canada.

RITCHIE, Edmund J. (*M* 1923) Vice-Pres., Sales, Sarco Co., Inc., 183 Madison Ave., New York, and (for mail) 2 Grace Court, Brooklyn, N. Y.

RITCHIE, William (*Life Member*; *M* 1909) 17 Van Reipen Ave., Jersey City, N. J.

RITTER, Arthur (*M* 1911) Office Mgr. (for mail) American Blower Corp., 50 West 40th St., New York, and 29 Edgemont Rd., Scarsdale, N. Y.

RIVARD, Melvin M. (*M* 1935) Mgr., Rivard Sales Co., 4550 Main St., and (for mail) 1805 West 49th St. Terrace, Kansas City, Mo.

ROBB, Joseph E. (*A* 1936) Sales Repr., Western Gas Div., Minneapolis-Honeywell Regulator Co., 1717 Illinois St., Lawrence, Kans.

ROBERTS, Henry L. (*M* 1916) Htg. Engr. & Contr. (for mail) 228 North 16th St., Philadelphia, and 1014 Allston Rd., Brookline, Del. Co. (Upper Darby P. O.), Pa.

ROBERTS, Henry P. (*A* 1936) Secy. & Engr. (for mail) Roberts-Hamilton Co., 713 S. Third St., and 1901 James Ave. S., Minneapolis, Minn.

ROBERTSON, James A. M. (*A* 1936) Vice-Pres. (for mail) The James Robertson Co., Ltd., 946 William St., Montreal, and 109 Sunnyside Ave., Westmount, Que., Canada.

ROBINSON, Arthur S. (*M* 1936) Engrg. Dept., E. I. duPont de Nemours Co., Wilmington, Del., and (for mail) 730 Ogden Ave., Swarthmore, Pa.

ROBINSON, Donald M. (*A* 1936) Sales Engr. (for mail) Buffalo Forge Co., 640 Woodward Bldg., Washington, D. C., and 5509 Lambeth Rd., Bethesda, Md.

ROBINSON, Edgar R. (*A* 1938) Draftsman, Ventilation Dept., New York Shipbuilding Corp., Camden, and (for mail) 216 Carlton Ave., Westmont, N. J.

ROBINSON, George L. (*A* 1935) Designer, E. I. duPont de Nemours, Engrg. Dept., duPont Bldg., and (for mail) 210 West 28th St., Apt. 1, Wilmington, Del.

ROBINSON, Jack A. (*A* 1940; *J* 1936) Air Cond. Engr., Australian Gas Light Co., Parker St., Box 481 AA. G. P. O., Sydney, and (for mail) 595 New South Head Rd., Rose Bay, N. S. W., Australia.

ROCHE, Ivor F. (*A* 1936) Mgr., Fess Oil Burners of Canada, Ltd., 1405 Drummond St., and (for mail) 4709 Cote St. Catherine Rd., Montreal, Que., Canada.

ROCK, George A. (*M* 1937) Plbg. & Htg. Contractor, 1343 S. W. Eighth St., Miami, Fla.

ROCKWELL, Theodore F. (*M* 1933; *J* 1932) Asst. Prof. Mech. Engrg. (for mail) Carnegie Institute of Technology, Schenley Park, and Glenover Pl., Aspinwall, Pittsburgh, Pa.

RODEE, E. John (*M* 1936) Chief Engr. (for mail) John B. Pierce Foundation, 290 Congress Ave., New Haven, and 130 Bellevue Ave., West Haven, Conn.

RODENHEISER, George B. (*M* 1933) Asst. Dir. (for mail) David Ranken Jr. School of Mechanical Trades, 4431 Finney Ave., and 3639a Dover Pl., St. Louis, Mo.

RODGERS, Frederick A. (*A* 1934) Pres., Rodgers Engineering Co., 204 Thomas Bldg., and (for mail) 3913 Amherst, Dallas, Tex.

RODGERS, Joseph S. (*A* 1937; *J* 1934) Principal Engrg. Draftsman, U. S. Government, War Dept., Edgewood Arsenal, Edgewood, and (for mail) 16 Fourth Ave. S., Glen Burnie, Md.

RODMAN, R. W. (*M* 1922) Supt. of Custodians (for mail) Board of Education, City of New York, Bureau of Plant Operation & Maintenance, 110 Livingston St., Brooklyn, and 175 West 73rd St., New York, N. Y.

ROEBUCK, William, Jr. (*M* 1917) Mfrs. Agent (for mail) 220 Delaware Ave., and 1240 Delaware Ave., Buffalo, N. Y.

ROGERS, Charles S. (*A* 1940) Htg. Engr., L. P. Steuart & Bro., Inc., 138-12th St. N. E., Washington, D. C., and (for mail) 304 Maple Ave., Falls Church, Va.

ROGERS, Robert C. (*A* 1937) House Htg. Engr., Community Natural Gas Co., and (for mail) 5722 Worth, Dallas, Tex.

ROGERS, Thomas L. C. (*A* 1939) Managing Dir. (for mail) National Air Conditioning Co., Ltd., 200 Piccadilly St. W., and 158 Victoria St., London, Ont., Canada.

ROLLAND, Sverre L. (*A* 1934) Mech. & Struc. Design Engr. (for mail) Oklahoma Gas & Electric Co., 321 N. Harvey, and 2207 N. W. 19th St., Oklahoma City, Okla.

RONSICK, Edward H. (*M* 1937) Htg. & Industrial Engr. (for mail) St. Louis County Gas Co., 231 W. Lockwood Ave., Webster Groves, and 7739 Stanford Ave., University City, Mo.

ROOT, Edwin B. (*M* 1936) Htg. Engr., Superior Safety Furnace Pipe Co., 5816-44 Forsythe Ave., Detroit, and (for mail) 964 Pierce St., Birmingham, Mich.

ROPER, Richard F. (*A* 1940) Pres., Pleasantaire Corp., Tower Bldg., 1401 "K" St. N. W., and (for mail) 3001 Woodland Dr. N. W., Washington, D. C.

ROSE, Harold J.* (*M* 1938) Senior Industrial Fellow (for mail) Mellon Institute, 4400 Fifth Ave., and 219 Lytton Ave., Pittsburgh, Pa.

ROSE, Howard J. (*M* 1934) Gen. Mgr., Suburban Air Conditioning Corp., 7 Depot Plaza, White Plains, and (for mail) 100 Siebrecht Pl., New Rochelle, N. Y.

ROSE, Jerome C. (*M* 1937) Air Cond. Engr., Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, and (for mail) 8831 Fort Hamilton Pkwy., Brooklyn, N. Y.

ROSEBROUGH, J. Stoddard (*A* 1937) Sales Engr., L. J. Mueller Furnace Co., 4246 Forest Park, and (for mail) 5937 McPherson Ave., St. Louis, Mo.

ROSEBROUGH, Robert M. (*M* 1920) Br. Mgr., L. J. Mueller Furnace Co., 4246 Forest Park Blvd., St. Louis, and (for mail) 519 S. Gore Ave., Webster Groves, Mo.

ROSEBY, Thomas A. (*M* 1939) Chief Designing Engr. (for mail) Carrier Australasia, Ltd., 36-40 Bourke St., Sydney, N. S. W., and 11 Gillies St., North Sydney, N. S. W., Australia.

ROSELL, Axel F. (*M* 1935) Engr., A. B. Svenska Flaktfabriken, Kungsgatan 16-18, Stockholm, and (for mail) Per Horbergs vag 4, S. Angby, Sweden.

ROSEN, Edmond J. (*A* 1939) Mgr., Temperature Control Dept. (for mail) Parker-Carpenter, Inc., 991 Bryant St., San Francisco, and 3272 Dakota St., Oakland, Calif.

ROSENBERG, Irwin (*S* 1939) Research Asst., A. S. H. V. E. Research Lab., U. S. Bureau of Mines, and (for mail) 612 N. St. Clair St., Pittsburgh, Pa.

ROSENBERG, Philip (*A* 1928) Secy.-Treas., Universal Fixture Corp., 135 West 23rd St., and (for mail) 250 West 104th St., New York, N. Y.

ROSENBLATT, Arthur M. (*M* 1938) Partner (for mail) Rosenblatt & Hunt, P. O. Box 828, and 1250 Edgewood Dr., Charleston, W. Va.

ROSS, John D. (*A* 1937) Sales Engr. (for mail) Railway & Engineering Specialties, Ltd., 417 St. Peter St., and 4376 Earnscliff Ave., Montreal, Que., Canada.

ROSS, John O.* (*M* 1920) Pres. (for mail) Ross Industries Corp., 350 Madison Ave., and 3 East 69th St., New York, N. Y.

ROSS, Roderick (*M* 1937) Consulting Engr. (for mail) Nicholas Bldg., 37 Swanston St., P. O. Box 1381 M, Melbourne, C. 1, and 5 Burns St., Elwood, Melbourne, S. 3, Australia.

ROSSITER, Irvin J. (*A* 1939) Office Mgr. (for mail) American Blower Corp., 803 Sherland Bldg., South Bend, Ind.

ROTH, Charles F. (*A* 1930) Pres. (for mail) International Exposition Co., Grand Central Palace, and 141 East 30th St., New York., N. Y.

ROTH, Harold R. (*M* 1935) Sales Engr. (for mail) Canadian Sirocco Co., Ltd., 57 Bloor St. W., and 5 Castleview Ave., Toronto, Ont., Canada.

ROTHMANN, S. C. (*M* 1936) Industrial Hygiene Engr. (for mail) West Virginia Workmen's Compensation Commission, State Capitol Bldg., and 2008½ Kanawha Blvd. E., Charleston, W. Va.

ROTTMAYER, Samuel I. (*A* 1933; *J* 1928) Mech. Engr. (for mail) Samuel R. Lewis, Cons. Engr., 407 S. Dearborn St., and 8041 Drexel Ave., Chicago, Ill.

ROWE, William A.* (*M* 1921) (Council, 1929-31) (for mail) The Trane Co., LaCrosse, Wis., and 718 Longfellow Ave., Detroit, Mich.

ROWE, William M. (*J* 1936) Sales (for mail) American Blower Corp., 1302 Swetland Bldg., Cleveland, and 151 Bradley Ave., Chagrin Falls, O.

ROWLEY, Frank B.* (*M* 1918) (*Presidential Member*) (Pres., 1932; 1st Vice-Pres., 1931; 2nd Vice-Pres., 1930; Council, 1927-33) Prof. of Mech. Engrg. and Director of Engineering Experiment Station, University of Minnesota, and (for mail) 4801 E. Lake Harriet Blvd., Minneapolis, Minn.

ROY, Arthur C. (*A* 1937) Mgr., Warm Air Div., Hoffman Specialty Co., Waterbury, and (for mail) 24 Valleywood Rd., Cos Cob, Conn.

ROY, Léo (*A* 1937) Power Sales Engr. (for mail) Quebec Power Co., 229 St. Joseph St., and 41 Laurentides Ave., Quebec, Que., Canada.

ROYER, Earl B. (*M* 1928) Designing Engr., Fosdick & Hilmer, Cons. Engrs., 1703 Union Trust Bldg., and (for mail) 6635 Iris Ave., Cincinnati, O.

RUDD, Dann J. (*M* 1937) Htg. & Vtg. Engrg., New York City Board of Education, 49 Flatbush Ave. Ext., Brooklyn, and (for mail) 307 Deer Park Ave., Babylon, L. I., N. Y.

RUEMMELE, Albert M. (*J* 1938) Sales Engr. (for mail) Silkensen-Shaw Furnace Co., 412-23rd St., and 3628-0½, Galveston, Tex.

RUFF, Adolph G. (*M* 1935) Supt. of Power, U. S. Playing Card Co., Park Ave., Norwood, and (for mail) 3824 Woodford Rd., Cincinnati, O.

RUFF, DeWitt C. (*M* 1922) Treas. (for mail) Healy-Ruff Co., 2255 University Ave., and 2211 St. Clair Ave., St. Paul, Minn.

RUGART, Karl (*A* 1924) Mfrs. Dist. Repr. (for mail) 26 South 20th St., Philadelphia, and 612 Bryn Mawr Ave., Penn Valley, Narberth, Pa.

RUGGLES, Robert F. (*M* 1936; *A* 1927; *J* 1926) Gen. Sales Mgr., Autovent Fan & Blower Co.,1805 N. Kostner Ave., and (for mail) 7041 N. Ozark, Edison Park, Chicago, Ill.

RUMMEL, Adolph J.* (*M* 1937) Air Cond. Engr. (for mail) San Antonio Public Service Co., 201 N. St. Mary's St., and 235 North Drive, San Antonio, Tex.

RUNGE, Carl H. (*A* 1940) Mgr. & Engr., Warm Air Htg. & Air Cond. Dept., American Radiator & Standard Sanitary Corp., 1635 W. St. Paul Ave., Milwaukee, and (for mail) 917 E. Meadow Pl., Whitefish Bay, Wis.

RUPPERT, C. Farrell (*S* 1939) Student, Purdue University, and (for mail) 400 Northwestern Ave., W. Lafayette, Ind.,

RUSSELL, Edward A. (*M* 1936) Chief Engr., Vapor Car Heating Co., Inc., 1600 S. Kilbourn Ave., and (for mail) 8103 Dorchester Ave., Chicago, Ill.

RUSSELL, J. Nelson (*Life Member*; *M* 1899) Dir. (for mail) Rosser & Russell, Ltd., 30 Conduit St., London, W. 1, and Fernacres, Fulmer, Buckinghamshire, England.

RUSSELL, Wayne B. (*A* 1936) Engr., Russell Furnace Works, 211 S. Greene St., and (for mail) 6064 E. Fourth, Spokane, Wash.

RUSSELL, William A. (*M* 1921) (Council, 1934-39) Zone Repr., Hoffman Specialty Co., and (for mail) 444 West 58th St., Kansas City, Mo.

RYAN, Harold J. (*M* 1940) Pres. (for mail) Harold J. Ryan, Inc., 122 East 42nd St., New York, and 461-81st St., Brooklyn, N. Y.

RYAN, Joseph B. (*M* 1938) Engr., 3904 Tracy Ave., Kansas City, Mo.

RYAN, William F. (*M* 1940; *A* 1939; *J* 1933) Chief Engr., The Salina Supply Co., 302-304 N. Santa Fe, and (for mail) 310 W. Republic, Salina, Kan.

RYBOLT, Arthur L. (*A* 1938) Gen. Mgr. (for mail) The Rybolt Heater Co., Miller St., and 75 Samaritan Ave., Ashland, O.

RYERSON, Herbert E. (*M* 1937) Architects Div. (for mail) Peoples Gas Light & Coke Co., 122 S. Michigan Ave., Chicago, and 908 S. Wheaton Ave., Wheaton, Ill.

S

SABIN, Edward R. (*M* 1919) Pres. (for mail) Edward R. Sabin & Co., 4710 Market St., Philadelphia, Pa., and 205 Page Ave., Allenhurst, N. J.

SABLE, Edward J. (*M* 1939) Treas. (for mail) The T. O. Murphy Co., 25-27 E. College St., and 246 W. Lorain St., Oberlin, O.

SACHS, Sam* (*J* 1940) Research Asst. in Mech. Engrg. (for mail) University of Illinois, 213 Mechanical Engrg. Lab., Urbana, and 1011½ S. First St., Champaign, Ill.

SADLER, C. Boone (*M* 1928) Associate Civil Engr. (for mail) Public Works Office, 11th Naval Dist., and 4828 Orchard Ave., San Diego, Calif.

SAGINOR, S. V. (*M* 1939) Production Mgr. (for mail) Davey Compressor Co., 206 N. Water St., and 307 Woodard Ave., Kent, O.

SAHLMANN, Frank L. (*A* 1937) Transportation Dept. (for mail) General Electric Co., East Lake Rd., and 3926 Beech Ave., Erie, Pa.

SAITO, Shozo (*M* 1923) (for mail) Saito Shozo Shoten, Ltd., Marunouchi Bldg., Opposite Tokyo Station, and 3, No. 4, 3chome, Kamata-ku, Tokyo, Japan.

SALE, Francis B. (*A* 1939) Sales Engr., Preferred Utilities Co., 2032 Belmont Rd., Washington, D. C.

SALINGER, Robert J. (*J* 1937) Mech. Engr., Reg. F. Taylor, Cons. Engr., 910 Bankers Mortgage Bldg., and (for mail) 1654 Danville, Houston, Tex.

SALTER, Ernest H. (*M* 1936) Engr. (for mail) Electrical Testing Laboratories, 2 East End Ave., New York, and 182 Cleveland Ave., Great Kills, S. I., N. Y.

SALZER, Alfred R., Jr. (*J* 1940) Mech. Engr., F. H. Chisholm, Cons. Mech. Engr., 810 Queen & Crescent Bldg., and (for mail) 2322 N. Villere St., New Orleans, La.

SAMPSON, Edwin T. (*A* 1938) Mgr., Acoustical Div. (for mail) Atlas Asbestos Co., Ltd., 110 McGill St., Montreal, and 28 Finchley Rd., Hampstead, Que., Canada.

SAMPSON, Will D. (*M* 1940) Resident Engr. (for mail) Kribs & Landauer, Mech. Cons., Ambassador Bldg., and 5818 Mardel Ave., St. Louis, Mo.

SAMUELS, Sidney (*A* 1928; *J* 1925) Pres. (for mail) Sidney Samuels, Inc., 146 West 99th St., and 245 West 107th St., New York, N. Y.

SANBERN, E. N.* (*M* 1923) Asst. Secy., Hoffman Specialty Co., Inc., 500 Fifth Ave., New York, N. Y., and (for mail) 118 Woodside Village, Stamford, Conn.

SANDERS, Charles M., Jr. (*J* 1938) Precipitron Repr. (for mail) Westinghouse Electric & Manufacturing Co., 411 North Seventh St., and 601 Westgate, Apt. 310, St. Louis, Mo.

SANDFORT, John F. (*J* 1938) Instructor, Mech. Engrg. (for mail) Iowa State College, Mech. Engrg. Dept., and 821½ Duff Ave., Ames, Ia.

SANDS, Clive C. (*M* 1929) G. P. O. Box 601 F. F., Sydney, N. S. W., Australia.

SANFORD, Arthur L. (*M* 1915) Mech. Engr., C. H. Johnston, Archts. & Engrs., 715 Empire Bank Bldg., and (for mail) 1129 Portland Ave., St. Paul, Minn.

SANFORD, Sterling S.* (*M* 1930) Sales Engr. (for mail) The Detroit Edison Co., 2000 Second Ave., and 1503 Seyburn Ave., Detroit, Mich.

SAPP, Charles L. (*A* 1936) Sales Mgr., Farquhar Furnace Co., and (for mail) 620 N. Walnut St., Wilmington, O.

SATTERLEE, Harry A. (*J* 1940) Partner (for mail) The Schulte Plumbing & Heating Co., 526 Joplin St., and 420 West "B" St., Joplin, Mo.

SAUNDERS, Lawrence P. (*M* 1933) Chief Engr., Research Div. (for mail) Harrison Radiator Div., General Motors Corp., and The Tuscarora Club, Lockport, N. Y.

SAURWEIN, George K. (*M* 1938) Supt., Engrg. Bldgs. & Grounds (for mail) Harvard University, Lehman Hall, Cambridge, and 247 Slade St., Belmont, Mass.

SAWDON, W. M.* (*M* 1920) Prof. of Experimental Engrg. (for mail) Cornell University, College of Engrg., and 1018 E. State St., Ithaca, N. Y.

SAWHILL, R. V. (*A* 1929) Exec. Vice-Pres. (for mail) Domestic Engineering Co., 110 East 42nd St., New York, and 115 Townsend Ave., Pelham Manor, N. Y.

SCALINGI, Ciro R. (*J* 1940; *S* 1938) Melchior, Armstrong, Dessau Co., 614 Memorial Dr., Cambridge, and (for mail) 94 Josephine Ave., Somerville, Mass.

SCANLON, Edward L. (*A* 1934) Chief Htg. Engr. (for mail) Equitable Gas Co., 435 Sixth Ave., and 3310 Regan Ave., Pittsburgh (10), Pa.

SCHAD, Clifford A. (*A* 1938; *J* 1937) Engr., United States Air Conditioning Corp., 2101 N. E. Kennedy St., and (for mail) 4425 43rd Ave. S., Minneapolis, Minn.

SCHAFER, Harry C. (*M* 1937) Sales Mgr. (for mail) Iroquois Gas Corp., 45 Church St., Buffalo, and 197 Union St., Hamburg, N. Y.

SCHECHTER, Jack E. (*A* 1941; *J* 1937) Chief Engr., Air Cond. Div. (for mail) S. J. O'Brien Sales Corp., 124 West 124th St., New York, and 2520 Kings Highway, Brooklyn, N. Y.

SCHECHTER, John P. (*A* 1941; *J* 1935) Mech. Engr., Austin Co., 429 Curtis Bldg., and (for mail) 1812 Burns Ave., Detroit, Mich.

SCHEIDECKER, Daniel B. (*A* 1919) Pres. (for mail) Hunter-Clark Ventilating System Co., 2800 Cottage Grove Ave., and 4626 N. Kilbourn Ave., Chicago, Ill.

SCHERNBECK, Fred H. (*A* 1930) Sales (for mail) William Bros. Boiler & Mfg. Co., Nicollet Island, and 5045 Portland Ave., Minneapolis, Minn.

SCHERRER, Leon B. (*A* 1941; *J* 1936) Sales Engr., St. Louis Cooperage Co., 101 Arsenal St., and (for mail) 6112 Simpson Terrace, St. Louis, Mo.

SCHLICHTER, Charles F. (*M* 1938) Dist. Mgr., Surface Combustion Corp., 600 W St. N. E., Washington, D. C., and (for mail) 6740 Fairfax Rd., Bethesda, Md.

SCHLICHTING, Walter G. (*M* 1932) Mgr., Air Cond. Dept., Clarage Fan Co., and (for mail) 1417 W. Lovell St., Kalamazoo, Mich.

SCHLICK, Paul F. (*A* 1940) Br. Mgr., Grinnell Co., Inc., and (for mail) 1410 Edgcumbe Rd., St. Paul, Minn.

SCHMIDT, E. Georg (*M* 1938) Consulting Engr., Berlin-Wilmersdorf, Kaiserallee 43, Germany.

SCHMIDT, Harry (*M* 1937) Executive Engr., Fedders Manufacturing Co., Inc., 57 Tonawanda St., and (for mail) 277 Norwalk Ave., Buffalo, N. Y.
SCHMIDT, Karl, Jr. (*J* 1937) Sales Engr., Michigan Consolidated Gas Co., 415 Clifford St., and (for mail) 14438 Mayfield Ave., Detroit, Mich.
SCHMIELER, Joseph B.* (*J* 1938) Jr. Aeronautical Engr., Naval Aircraft Factory, U. S. Navy Yard, Philadelphia, Pa., and (for mail) 311 Delaware St., Woodbury, N. J.
SCHNEIDER, Charles H. (*J* 1938) Sales Engr. (for mail) Ilg Electric Ventilating Co., 1031 Commercial Trust Bldg., Philadelphia, Pa., and 370 North 32nd St., Camden, N. J.
SCHNELL, Robert H. (*A* 1938) Asst. Mech. Engr. (for mail) B. E. Landes, Mech. Engr., 915 Hubbell Bldg., and 1617–33rd St., Des Moines, Ia.
SCHOEFFTER, Hans M. (*A* 1941; *J* 1939) Mech. & Sales Engr. (for mail) Aerofin Corp., 410 S. Geddes St., and 666 W. Onondaga St., Syracuse, N. Y.
SCHOENIJAHN, Robert P. (*M* 1919) Consulting Engr. (for mail) 304-5 Industrial Trust Bldg., and 719 Nottingham Rd., Wilmington, Del.
SCHOEPFLIN, Paul H. (*M* 1920) Pres. (for mail) Niagara Blower Co., 6 East 45th St., New York, and 91 Valley Rd., Larchmont, N. Y.
SCHREIBER, Herbert W. (*A* 1937) Br. Mgr. (for mail) Johnson Service Co., 507 E. Michigan St., and 3136 N. Eighth St., Milwaukee, Wis.
SCHROEDER, William R. (*A* 1939) Sales Engr. (for mail) General Heating Corp., 7120 N. Clark St., and 3631 Janssen St., Chicago, Ill.
SCHROTH, August H. (*M* 1911) N. Y. Mgr., Columbia Radiator Co., 101 Park Ave., New York, N. Y., and (for mail) 167 N. Grove St., P. O. Box 47, E. Orange, N. J.
SCHUBERT, Arno G. (*M* 1939) Asst. Prof. Mech. Engrg., Rensselaer Polytechnic Institute, Troy, and (for mail) 1301 Broadway, Watervliet, N. Y.
SCHUETZ, Clyde C. (*A* 1936) Research Engr. (for mail) United States Gypsum Co., 1253 Diversey Pkwy., Chicago, and 206 S. Edward St., Mt. Prospect, Ill.
SCHULEIN, Ernst H. (*A* 1939; *J* 1937) Consulting Engr., Birch & Krogboe, v. Farimagsgade 31, Copenhagen, V., and (for mail) 3 Dalgas Blvd., Copenhagen, F., Denmark.
SCHULER, William B. (*A* 1937) Sales, Taco Heaters, Inc., 342 Madison Ave., New York, N. Y., and (for mail) 7655 Merrill Ave., Chicago, Ill.
SCHULTZ, Albert W. (*M* 1936) Engr., Grinnell Co., Inc., 240 Seventh Ave. S., and (for mail) 5204 France Ave. S., Minneapolis, Minn.
SCHULZ, Edward L. (*J* 1937) Engr. (for mail) Carrier Corp., 12 South 12th St., Philadelphia, and 411 Pembroke Rd., Bala-Cynwyd, Pa.
SCHULZ, Howard I. (*A* 1915) Mgr. (for mail) Crane Co., 1223 W. Broad St., and 1600 Monument Ave., Richmond, Va.
SCHULZE, Ben H. (*M* 1921) Eastern Sales Mgr. (for mail) Kewanee Boiler Corp., 37 West 39th St., and 67 Park Ave., New York, N. Y.
SCHURMAN, John A. (*M* 1936; *J* 1935) Sales Engr., York Ice Machinery Corp., 2700 Washington Ave. N. W., Cleveland, and (for mail) 16307 Lakewood Heights Blvd., Lakewood, O.
SCHWARTZ, Jacob (*A* 1936; *J* 1929) Contractor (for mail) Samuel Schwartz & Son, Inc., 30 West 27th St., Bayonne, and 12 Van Houten Ave., Jersey City, N. J.
SCHWARTZ, Maurice (*A* 1938) Air Cond. Supervisor (for mail) Queens Borough Gas & Electric Co., 1610 Far Rockaway Blvd., and 739 Caffrey Ave., Far Rockaway, L. I., N. Y.
SCHWARTZ, Norman E. (*M* 1939; *A* 1938) Dist. Engr., Chrysler Corp., Airtemp Div., 551 S. Harrison Ave., Kirkwood, Mo.
SCOFIELD, Paul C. (*A* 1937; *J* 1933) Air Cond. Engr., Lockheed Aircraft Corp., Burbank, and (for mail) 3879 Edenhurst Ave., Los Angeles, Calif.
SCOTT, Allison F. II. (*M* 1937) Vice-Pres. & General Sales Mgr. (for mail) Hoffman Specialty Co., 77 Bedford St., Stamford, Conn.
SCOTT, George M. (*M* 1915) Pres. (for mail) Child & Scott-Donohue, Inc., 153 East 38th St., New York, and Fort Hill Village, Scarsdale, N. Y.
SCOTT, William P., Jr. (*A* 1941; *J* 1939) Vice-Pres., Scott Co., 243 Minna St., and (for mail) 255 Santa Paula Ave., San Francisco, Calif.
SCOTTI, Frederico D. (*M* 1939) Doctor Engr., Htg. & Air Cond. (for mail) via Brunelleschi, 4, and via Maffei, 77, Florence, Italy.
SEAL, Alfred T. (*M* 1938) Asst. Purchasing Agent, Research Corp., Bound Brook, and (for mail) 260 Sandford Ave., North Plainfield, N. J.
SEARLE, William J., Jr. (*M* 1938) Air Cond. Engr., The Ballinger Co., 105 South 12th St., Philadelphia, and (for mail) 110 Woodside Ave., Narberth, Pa.
SEEBER, Rex R.* (*M* 1934) Head Dept. Mech. Engrg. (for mail) Michigan College of Mining & Technology, and Houghton, Mich.
SEELBACH, Herman, Jr. (*A* 1937) Sales Engr., Minneapolis-Honeywell Regulator Co., 45 Allen St., and (for mail) 120 Mariner, Buffalo, N. Y.
SEELERT, Edward H. (*A* 1935) Secy.-Treas. (for mail) McQuay, Inc., 1600 Broadway N. E., and 2927 Ulysses St. N. E., Minneapolis, Minn.
SEELEY, Lauren E.* (*M* 1930) Assoc. Prof. (for mail) Mason Laboratory, Yale School of Engineering, and 130 Everit St., New Haven, Conn.
SEELIG, Alfred E. (*M* 1926) Pres. & Gen. Mgr., L. J. Wing Mfg. Co., 154 West 14th St., and (for mail) 640 Riverside Dr., New York, N. Y.
SEELIG, Lester (*M* 1925) Chief Engr., Museum of Science & Industry, 57th St. & Lake Michigan, and (for mail) 5459 Cornell Ave., Chicago, Ill.
SEIGEL, Lawrence J. (*J* 1939) Test Engr., Cleveland Electric Illuminating Co., Cleveland, and (for mail) 350 East 211th St., Euclid, O.
SEITER, J. Earl* (*M* 1928) Asst. Mgr. in charge of Dist. Steam Sales, Consolidated Gas, Electric Light & Power Co., Rm. 500 Lexington Bldg., and (for mail) 7117 Bristol Rd., Baltimore, Md.
SEKIDO, Kunisuke (*Life Member*; *M* 1903) Consulting Engr., 19 Momozono, Nakano, Tokyo, Japan.
SELIG, Ernest T., Jr. (*M* 1936) Chief Research Engr. (for mail) Campbell Taggart Research Corp., 4049 Pennsylvania Ave., and 7400 Belleview Ave., Kansas City, Mo.
SELLMAN, Nils T. (*M* 1922) Asst. Vice-Pres., Consolidated Edison Co. of N. Y., Inc., 4 Irving Pl., New York, N. Y.
SELTZER, Paul A. (*J* 1938) Engr., Bryant Air Conditioning Corp., 915 N. Front St., Philadelphia, and (for mail) 154 E. Marshall Rd., Lansdowne, Pa.
SENIOR, Richard L. (*M* 1925) Pres. (for mail) R. L. Senior, Inc., 103 Park Ave., New York, and 10 Cherry Ave., New Rochelle, N. Y.
SETTELMEYER, James T. (*A* 1940; *J* 1938) Instructor, Refrig. & Air Cond., Essex County Vocational Schools, 294 Norfolk St., Newark, and (for mail) 332 N. Maple Ave., East Orange, N. J.
SEVERNS, William H.* (*M* 1933) Prof. Mech. Engrg., University of Illinois, and 609 Indiana Ave., Urbana, Ill.
SEYFANG, William G. (*M* 1939) Managing Engr., Div. of Plant, Board of Education, 814 City Hall, and (for mail) 116 Dorchester Rd., Buffalo, N. Y.
SEYMOUR, James E. (*A* 1937) Owner, Lee & Seymour, 346 Russell St., and (for mail) 208 Lakewood Blvd., Madison, Wis.
SGAMBATI, Anthony P. (*S* 1939) Sears & Roebuck Co., Engrg. Dept., 5029 Morewood Pl., Pittsburgh, Pa., and (for mail) 3 Spencer St., Youngstown, O.

SHAER, I. Ernest (*A* 1934) Treas.-Sales Engr., Shaer & Turner Engineering Co., 88 Broad St., Boston, and (for mail) 43 Ormond St., Dorchester, Mass.

SHAFER, W. P., Jr. (*J* 1941; *S* 1939) Dist. Mgr. (for mail) The Trane Co., 235 Sterick Bldg., and 766 N. Auburndale Ave., Memphis, Tenn.

SHAFFER, Chester E. (*M* 1937) Research Engr. (for mail) Koppers Co., Kearny, and Troy Hills, N. J.

SHANKLIN, Arthur P. (*M* 1929) Dist. Mgr. (for mail) Carrier Corp., 12 South 12th St., Philadelphia, and 40 Amherst Ave., Swarthmore, Pa.

SHANKLIN, John A. (*M* 1928) Vice-Pres.-Treas. (for mail) West Virginia Heating & Plumbing Co., 233 Hale St., Charleston, W. Va.

SHAPIRO, Charles A. (*J* 1938) Sales Engr., Johnson Service Co., 2853 North 12th St., and (for mail) 2041 N. Wanamaker St., Philadelphia, Pa.

SHARP, Henry C. (*M* 1935) Sales Repr., Herman Nelson Corp., Fullerton Bldg., Rm. 221, St. Louis, and (for mail) 7442 Melrose Ave., University City, Mo.

SHARP, John E. (*A* 1939) Owner (for mail) Sharp's, 215 N. Fifth St., and 1529 W. Grand Ave. S., Springfield, Ill.

SHARP, John R. (*A* 1937) Major (for mail) Corps of Engineers, 104th Engineers, Ft. Dix, and Maple St., Haworth, N. J.

SHAW, Burton E.* (*A* 1936; *J* 1934) Research Chief (for mail) Penn Electric Switch Co., Goshen, and Bristol, Ind.

SHAW, John A. (*M* 1938) General Elec. Engr. (for mail) Canadian Pacific Railway Co., Windsor Station, Montreal, and 448 Lansdowne Ave., Westmount, Que., Canada.

SHAW, N. J. H. (*M* 1927; *J* 1925) Sales Engr., Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain, and (for mail) 37 Benjamin Rd., Arlington, Mass.

SHEA, Michael B. (*M* 1921) Sales Dept. (for mail) American Radiator & Standard Sanitary Corp., 8019 Jos. Campau, and 4080 Blaine St., Detroit, Mich.

SHEARER, William A., Jr. (*J* 1941; *S* 1939) Jr. Engr., E. I. duPont de Nemours Co., and (for mail) 9 McFarland St., Charleston, W. Va.

SHEARS, Matthew W. (*M* 1922) Engr. (for mail) C. A. Dunham Co., Ltd., 1523 Davenport Rd., and 39 Sylvan Ave., Toronto, Ont., Canada.

SHEFFLER, Morris (*M* 1921) Pres. (for mail) Sheffler-Gross Co., Inc., 1000 Drexel Bldg., Philadelphia, and 419 Chapel Rd., Melrose Park, Montgomery Co., Pa.

SHELDON, Nelson E. (*M* 1927) Br. Mgr. (for mail) Carrier Corp., Syracuse, and 41 Lanark Crescent, Rochester, N. Y.

SHELDON, William D., Jr. (*A* 1936; *J* 1934) Chief Engr., Sheldon's, Ltd., and (for mail) Cedar St., Galt, Ont., Canada.

SHELEY, Earle D. (*M* 1937) Pres. (for mail) Glanz & Killian Co., 1761 W. Forest Ave., Detroit, and Box 243, Birmingham, Mich.

SHELL, Jack (*M* 1940) Chief Engr., Air Cond. Dept. (for mail) Jefferson Amusement Co., Box 3191, and R. F. D. 1, Box 1015, Beaumont, Tex.

SHENK, Donald H. (*M* 1934) Assoc. Prof. Mech. Engrg. (for mail) Clemson Agricultural College, Riggs Hall, and 106 Calhoun Circle, Clemson, S. C.

SHEPARD, John deB. (*M* 1937; *J* 1929) Air Cond. Engr. (for mail) Consolidated Gas Electric Light & Power Co., Lexington Bldg., Rm. 406, and 4823 Keswick Rd., Baltimore, Md.

SHEPPARD, Frank A. (*M* 1918) Sales (for mail) Johnson Service Co., 1031 Wyandotte St., and 27 East 70th St., Kansas City, Mo.

SHEPPERD, Parker D. (*A* 1940; *J* 1938) Br. Mgr. (for mail) Johnson Service Co., 315 St. Charles St., and 17 Metairie Court, New Orleans, La.

SHERBROOKE, Walter A. (*M* 1938) Grinnell Co., Inc., 29-50 Northern Blvd., Long Island City, and (for mail) 117-01 Park Lane South, Kew Gardens, L. I., N. Y.

SHERET, Andrew (*M* 1929; *A* 1925) Pres. (for mail) Andrew Sheret, Ltd., 1114 Blanshard St., and 1030 St. Charles St., Victoria, B. C., Canada.

SHERMAN, Ralph A.* (*M* 1933) Supvr., Fuels Div. (for mail) Battelle Memorial Institute, 505 King Ave., and 1893 Coventry Rd., Columbus, O.

SHERMAN, Victor L. (*M* 1935) 643 Hillside Ave., Glen Ellyn, Ill.

SHERMAN, W. P. (*M* 1937) Chief Engr. (for mail) c/o T. Louis Murray, 1807 Main St., and 1802 Pendleton St., Columbia, S. C.

SHERWOOD, Laurence T. (*M* 1937) Glass Technologist (for mail) Pennsylvania Wire Glass Co., Dunbar, and 11 Angle St., Connellsville, Pa.

SHIELDS, Carl D. (*J* 1937; *S* 1936) Engr. (for mail) U. S. Rubber Co., 6600 E. Jefferson, Detroit, and 1167 Kensington, Grosse Pointe, Mich.

SHIPLEY, Sylvanus C. (*M* 1938) Cost Engr. (for mail) Minneapolis-Honeywell Regulator Co., 2753 Fourth Ave. S., and 1550 E. River Terrace, Minneapolis, Minn.

SHIVERS, Paul F. (*M* 1930) Chief Engr. (for mail) Minneapolis-Honeywell Regulator Co., and 75 W. Maple St., Wabash, Ind.

SHORB, Will A. (*Life Member*; *M* 1909) Dist. Mgr., Decatur Pump Co., Decatur, Ill., and (for mail) 47 N. Lime St., Apt. 6, Lancaster, Pa.

SHORE, David* (*J* 1938) New York Shipbuilding Corp., Camden, N. J., and (for mail) 4044 Walnut St., Apt. 4, Philadelphia, Pa.

SHROCK, John H. (*M* 1924) Vice-Pres. (for mail) New York Blower Co., and 1002 Indiana Ave., LaPorte, Ind.

SHULTZ, Earle (*A* 1919) Vice-Pres. (for mail) Illinois Maintenance Co., 72 W. Adams St., and Edgewater Beach Apts., Chicago, Ill.

SHUMAN, Laurence (*M* 1939) Mech. Engr., U. S. Housing Authority, North Interior Bldg., Washington, D. C., and (for mail) 8367 16th St., Silver Spring, Md.

SIEGEL, Daniel E. (*J* 1940; *S* 1938) Estimator, Guaranteed Heating & Engineering Co., 1923 Olive St., St. Louis, and (for mail) 7716 Wise Ave., Richmond Heights, Mo.

SIEGEL, Roy C. (*A* 1939) Owner (for mail) International Chimney Co., 503 Ellicott Sq., and 243 Norwalk Ave., Buffalo, N. Y.

SIGMUND, Ralph W. (*M* 1932) Dist. Mgr. (for mail) B. F. Sturtevant Co., 913 Provident Bank Bldg., and 130 Wm. H. Taft Rd., Cincinnati, O.

SILBERSTEIN, Bernard G. (*M* 1937) Dist. Mgr. (for mail) Ilg Electric Ventilating Co., 622 Broadway, and 814 East Mitchell Ave., Cincinnati, O.

SILVERA, Americo (*J* 1939) Engr., Carrier Corp., Syracuse, N. Y.

SIMKIN, Milton (*J* 1936; *S* 1933) Engr., Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, N. Y., and (for mail) 103 Brighton Ave., Perth Amboy, N. J.

SIMONS, Byron C. (*M* 1938) Br. Mgr. (for mail) Minneapolis-Honeywell Regulator Co., 3033 Locust Blvd., St. Louis, and 20 Orchard Ave., Webster Groves, Mo.

SIMONS, Edward W. (*M* 1938) Chief Mech. Engr., Redwood Manufacturers Co., 1600 Hobart Bldg., and (for mail) 2418 30th Ave., San Francisco, Calif.

SIMONSON, George M. (*M* 1937) Consulting Engr. (for mail) 625 Market St., Rm. 309, San Francisco, and 20 Loreta Ave., Piedmont, Calif.

SIMPSON, William K. (*M* 1919) Engrg. Consultant & Proprietor, R. H. Brown & Co., New Haven, and (for mail) 9 Sands St., Waterbury, Conn.

SIPP, Edmund F. (*J* 1940) Jr. Engr., Mills Novelty Co., 2060 N. Kolmar Ave., and (for mail) 3755 Nora Ave., Chicago, Ill.

SKAGERBERG, Rutcher (*M* 1924; *J* 1921) Sr. Mech. Engr., U. S. Housing Authority, North Interior Bldg., Washington, D. C., and (for mail) 420 Tyler Pl., Alexandria, Va.

SKELLEY, Jerome H. (*A* 1938) Owner (for mail) Skelley Heating Equipment Co., 1234 Speer Blvd., and 4101 E. Ellsworth Ave., Denver, Colo.

SKINNER, Alton, Jr. (*J* 1940) Engr. (for mail) J. R. Bagwell Co., 320 Holland St., P. O. Box 2007, and 1202 Vickers Ave., Durham, N. C.

SKINNER, Henry W. (*M* 1920) Consulting Engr. (for mail) 4816 Dexter St., Ft. Worth, Tex.

SKLAREVSKI, Rimma (*A* 1940; *J* 1936) Jr. Mech. Engr., Sewerage Works, City of Baltimore, and (for mail) 226 East University Pkwy., Baltimore, Md.

SKLENARIK, Louis (*A* 1937; *J* 1928) 305 East 72nd St., New York, N. Y.

SLAWSON, Lloyd E. (*A* 1938) Office Mgr. (for mail) Barber-Colman Co., 3030 Euclid Ave., Cleveland, and 109 Southwick Dr., Bedford, O.

SLEMMONS, John D. (*M* 1937) Mgr. Sales Office (for mail) American Blower Corp., 666 Marion Rd., Columbus, and Rte. 2, Wilson Rd., Worthington, O.

SLOANE, David (*S* 1939) Student, Carnegie Institute of Technology, Pittsburgh, Pa., and (for mail) 922 Main St., LaCrosse, Wis.

SLUSS, Alfred H.* (*M* 1935) Prof. Mech. Engrg., University of Kansas, 224 Fowler Shops, and (for mail) 827 Mississippi St., Lawrence, Kans.

SMAK, Julius R. (*A* 1934) Supt. of Service Section, Crane Company, South Ave., and (for mail) 3135 Park Ave., Bridgeport, Conn.

SMALL, Bartlett R. (*M* 1938; *A* 1937; *J* 1932) Staff Engr. (for mail) Aluminum Co. of America, 801 Gulf Bldg., Pittsburgh, and 34 Hillman St., Brentwood, Pittsburgh, Pa.

SMITH, Elmer G.* (*M* 1929) Assoc. Prof. of Physics, Agricultural & Mechanical College of Texas, Dept. of Physics, College Station, Tex.

SMITH, Gard W. (*M* 1927) Sales Engr., Premier Furnace Co., Dowagiac, Mich., and (for mail) 1131 Guilford St., Huntington, Ind.

SMITH, Gerald E. (see Special Service Roll, p. 73).

SMITH, Milton S. (*M* 1919) Treas. & Gen. Mgr. (for mail) Buensod-Stacey Air Conditioning, Inc., 60 East 42nd St., New York, N. Y., and 13 N. Terrace, Maplewood, N. J.

SMITH, Nelson J. (*M* 1938) Air Cond. Design Engr., Frigidaire Division, 300 N. Taylor St., Dayton, O.

SMITH, Reginald J. (*M* 1936) Mgr. (for mail) Smith & Elston, 71 Third Ave., and 112 S. Maple St., Timmins, Ont., Canada.

SMITH, Roger C. (*A* 1940) Branch Engrg. Dept., York Ice Machinery Corp., 117 South 11th St., St. Louis, and (for mail) 7040 Tulane Ave., University City, Mo.

SMITH, Roy L. (*S* 1939) Student, Carnegie Institute of Technology, 5010 Morewood Pl., Pittsburgh, Pa.

SMITH, Stuart (*A* 1936) American Radiator & Standard Sanitary Corp., Connecticut Ave. & S St. N. W., Washington, D. C., and (for mail) 6922 Fairfax Rd., Bethesda, Md.

SMITH, Walter H. (*M* 1939) Chief Engr. (for mail) The T. Eaton Co., Ltd., Engineers Office, and 7 Kingscourt Dr., Toronto, Ont., Canada.

SMITH, Wilbur F. (*M* 1920) Consulting Engr., W. M. Anderson Co., 600 Schuylkill Ave., Pittsburgh, and (for mail) 709 Braeburn Lane, Penn Valley, Narberth P. O., Pa.

SMITH, William D. (*M* 1937; *A* 1935) Pres. (for mail) Bryant-Smith Inc., 2153 Prospect Ave., Cleveland, and R. D. No. 1, Peninsula, O.

SMITH, William O. (*A* 1937) Pres. (for mail) Smith Automatic Heat Service Co., 19250 John R St., Detroit, and 343 E. Maplehurst, Ferndale, Mich.

SMOOT, T. H. (*M* 1935) Gen. Mgr. & Chief Engr., Anchor Post Fence Co., Htg. Div., 6500 Eastern Ave., and (for mail) 1302 Southview Rd., Baltimore, Md.

SMYERS, Edward C. (*A* 1933) Sales Engr., Barber-Colman Controls, 1013 Penn Ave., Wilkinsburg, and (for mail) 148 Jamaica Ave., West View, Pittsburgh, Pa.

SNAVELY, A. Bowman (*M* 1937) Chief Engr., Hershey Chocolate Corp., Hershey, Pa.

SNAVELY, Earl R. (*M* 1937) Dean, New York Technical Inst., 108 Fifth Ave., New York, N. Y., and Sales Engr. (for mail) Thomas A. Edison, Inc., West Orange, and 10 Parkway Dr., Clark Township, N. J.

SNOOK, Alfred H. (*A* 1940) Mfrs. Agent, Box 117, Wayland, Mich.

SNYDER, Edwin F., Jr. (*J* 1940) Htg. Engr., Bick Heating Supply Co., and (for mail) 308 Courtland St., Dowagiac, Mich.

SNYDER, Jay W. (*M* 1917) Member of Firm (for mail) Snyder & McLean, 2308 Penobscot Bldg., and 8987 Martindale Ave., Detroit, Mich.

SOBEL, Frank (*S* 1939) Student, New York Technical Institute, 108 Fifth Ave., and (for mail) 745 East 175th St., New York, N. Y.

SODEMANN, Paul (*M* 1926; *J* 1920) Sales Engr. (for mail) Sodemann Heat & Power Co., 2306 Delmar Blvd., and 4136 Farlin Ave., St. Louis, Mo.

SODEMANN, William C. B. (*M* 1919) Pres. (for mail) Sodemann Heat & Power Co., 2306 Delmar Blvd., St. Louis, and 7542 Teasdale Ave., University City, Mo.

SOETERS, Matthew (*M* 1937) Consulting Engr., 858 E. Grand Blvd., Detroit, Mich.

SOLSTAD, Lester L. (*J* 1930) Engr. (for mail) Austin Sheet Metal Works, 5109 W. Chicago Ave., and 5348 W. Potomac, Chicago, Ill.

SOMERS, William S. (*M* 1938; *A* 1928; *J* 1926) Chief Engr., Lamneck Products, Inc., 1025 Lamneck St., and (for mail) 113 Kenwood Dr., Middletown, O.

SOMMERFIELD, Sumner S. (*A* 1941; *J* 1936) 4106 Keystone Ave., Chicago, Ill.

SOMMERS, William J. (*M* 1937) Sales Repr., Ilg Electric Ventilating Co., 505 Delaware Ave., Buffalo, and (for mail) 150 Stillwell Ave., Kenmore, N. Y.

SOPER, H. A. (*M* 1916) Vice-Pres. (for mail) American Foundry & Furnace Co., and 1122 E. Monroe St., Bloomington, Ill.

SOULE, Lawrence C.* (*M* 1908) Consulting Engr., Aerofin Corp., Syracuse, N. Y., and (for mail) Gordon & Stewart Rds., Essex Fells, N. J.

SOUTHMAYD, Richard T. (*J* 1936) Sales (for mail) American Blower Corp., 1302 Swetland Bldg., Cleveland, and 31 South St., Chagrin Falls, O.

SPALL, Edward G. (*A* 1939) Sales Engr., Powerlite Devices, Ltd., Penn Electric Switch Division, 171 John St., and (for mail) 3 Thornhill Ave., Toronto, Ont., Canada.

SPARKS, James D. (*A* 1937) Northwest Repr., Ilg Electric Ventilating Co., 7331 W. Green Lake Way, Seattle, Wash.

SPECKMAN, Charles H. (*M* 1918) Consulting Htg. & Vtg. Engr., Rm. 482, Bourse Bldg., Philadelphia, Pa.

SPELBRINK, Robert G. (*J* 1940) Office Engr., York Ice Machinery Corp., 117 South 11th St., St. Louis, and (for mail) 8515 Antler Dr., Richmond Heights, Mo.

SPELLER, F. N.* (*M* 1908) Advisory Engr. (for mail) National Tube Co., P. O. Box 266, and 6411 Darlington Rd., Pittsburgh, Pa.

SPENCE, Morton R. (*J* 1934) (for mail) Rundle & Spence Mfg. Co., 445 N. Fourth St., and 709 E. Lexington Blvd., Milwaukee, Wis.

SPENCE, Robert A. (*J* 1937) Asst. to Supt., Engrg. Dept., Harvard University, Lehman Hall, Cambridge, and (for mail) 33 Barnard Rd., Belmont, Mass.

SPENCER, Roland M. (*A* 1940; *J* 1934) Br. Mgr. (for mail) The Powers Regulator Co., 329 M. & M. Bldg., and 3010 Blodgett, Houston, Tex.

STEVENS, Earl K. (*A* 1940) Treas. (for mail) International Exposition Co., 480 Lexington Ave., New York, N. Y., and Fairfield Ave., Greenwich, Conn.

STEVENS, Harry L. (*M* 1934; *A* 1927; *J* 1924) Pres. (for mail) M. M. Stevens Co., 108 W. Sherman, and 320 West 20th, Hutchinson, Kan.

STEVENS, Kenneth M. (*J* 1936) Sales Engr. (for mail) Powers Regulator Co., 409 East 13th St., and 4734 Summit, Kansas City, Mo.

STEVENS, Wayne H. (*A* 1939) Engr. & Service Mgr. (for mail) Shellenberger, Gregg & Co., 2203 N. Prospect Ave., and 1609 E. Riverside Pl., Milwaukee, Wis.

STEVENS, William R. (*A* 1934) Partner (for mail) L. E. Stevens Co., 626 Broadway, Cincinnati, O., and 30 Chalfonte Court, Ft. Thomas, Ky.

STEVENSON, Melvin J. (*M* 1935) Prof. Mech. Engr., Htg., Vtg. & Air Cond., 5321 Cornell Ave., Chicago, Ill.

STEVENSON, Wilbur W. (*M* 1928) Steam Htg Engr. (for mail) Allegheny Co. Steam Heating Co., 435 Sixth Ave., and 1125 Lancaster Ave., Pittsburgh, Pa.

STEWART, Charles W. (*M* 1919; *A* 1918) Asst. Secy. (for mail) Hoffman Specialty Co., Waterbury Natl. Bank Bldg., and 21 Yates Ave., Waterbury, Conn.

STEWART, Duncan J.* (*M* 1936; *A* 1930) Mgr., Electrical Div. (for mail) Barber-Colman Co., and R. R. No. 4, Rockford, Ill.

STEWART, James P. (*A* 1940; *J* 1937) Engr., Power Engineering Corp., 517 Brooks Bldg., Wilkes-Barre, and (for mail) 338 Maple Ave., Kingston, Pa.

STEWART, John N. (*A* 1939) Plan Examiner, Div. of Smoke Regulation, District Bldg., and (for mail) 2019 N St. N. W., Washington, D. C.

STEWART, Wesley O. (*A* 1938) Branch Mgr. (for mail) Johnson Service Co., 153 West Ave. 34, and 1921½ Rodney Dr., Los Angeles, Calif.

STILES, Gordon S. (*A* 1941; *J* 1936) Heat Prover, Cities Service Oil Co., Cedar Rapids, and (for mail) 224 Welch Ave., Ames, Ia.

STILLER, Fredrick W. (*J* 1933) Estimator (for mail) F. C. Stiller & Co., 129 S. Tenth St., and 138 West 49th St., Minneapolis, Minn.

STITES, Richard (*J* 1937) Sales Engr. (for mail) Coon DeVisser Co., 2051 W. Lafayette Blvd., and 2170 E. Jefferson, Detroit, Mich.

STOCK, Charles Stevens (*M* 1936) Dist. Repr. (for mail) The Herman Nelson Corp., Rm. 404, 1108-16th St. N. W., Washington, D. C., and 6752 Fairfax Rd., Bethesda, Md.

STOCKWELL, William R. (*Life Member*; *M* 1903; *J* 1901) Gen. Mgr. Mfg. Div., Weil McLain Co., Michigan City, Ind.

STOKES, Alvin D. (*M* 1936) Engr. (for mail) Adsco Bldg. Contractors, 901 Seventh St. S. W., Washington, D. C., and 4010 Ellendale Rd., Drexel Hill, Pa.

STOKES, Arledge (*J* 1936) Engr. (for mail) Mehring & Hanson Co., 12 H St. N. E., Washington, D. C., and 4908 North 17th St., Arlington, Va.

STONE, Frank M. (*M* 1940) Secy.-Treas. (for mail) Sam Stone, Jr. & Co., Inc., Archts., 1800 Masonic Temple, and 2214 Adams St., New Orleans, La.

STORCH, Clemens A. (*M* 1930) Sales Engr., Johnson Service Co., 1355 Washington Blvd., Chicago, and (for mail) 331 Cumnor Rd., Kenilworth, Ill.

STORMS, Robert M. (*M* 1936) Consulting Engr., Htg. & Vtg. (for mail) 816 W. Fifth St., Los Angeles, and 354 W. Wilson, Glendale, Calif.

STORY, James W. (*A* 1940) Engr. (for mail) Leo S. Weil & Walter B. Moses, Cons. Engr., 425 S. Peters St., and 920 S. Carrollton Ave., New Orleans, La.

STOTT, Douglas A. (*A* 1940) Mgr. (for mail) Canadian Powers Reg. Co., Ltd., 195 Spadina Ave., and 19 Gresham Rd., Toronto, Ont., Canada.

STOTT, F. W. (*M* 1938) Sales Engr., C. A. Dunham Co., Ltd., 1139 Bay St., Toronto, and (for mail) 86 Allan St., Oakville, Ont., Canada.

STOTZ, Robert B. (*J* 1938) Sales Engr. (for mail) Frigidaire Div., General Motors Sales Corp., 675 Greenwood Ave., and 1702 Harvard Rd., Atlanta, Ga.

STRAND, Charles A. (*A* 1940) Supt., Bruce Wigle Plumbing & Heating Co., 9117 Hamilton Ave., and (for mail) 6533 Barlum Ave., Detroit, Mich.

STRAUCH, Paul C. (*A* 1934) Sales Engr., Henry Furnace & Foundry Co., 24 South 18th St., Pittsburgh, and (for mail) Cambridge Court Apts., 131 Edgewood Ave., Edgewood, Pa.

STREATER, Edward C. (*A* 1939) Mgr. (for mail) L. E. Streater Lumber Co., Spring Park, and Mound, Minn.

STREVELL, Roger P. (*M* 1934) Pres. and Treas. (for mail) William R. Hogg Co., Inc., 900 Fourth Ave., Asbury Pk., and Victor Pl. & State Highway, Neptune, N. J.

STROCK, Clifford (*M* 1937; *A* 1929) Editor (for mail) Heating & Ventilating, 148 Lafayette St., New York, and P. O. Box 750, Amityville, L. I., N. Y.

STROMGREN, Sven G. (*M* 1938) c/o Svenska Flaktfabriken, Kungsgatan 16, Stockholm, Sweden.

STROUSE, Sherman W. (*A* 1934) Sales Mgr., Trane Co., 493 Franklin St., Buffalo, and (for mail) 95 Mayville Ave., Kenmore, N. Y.

STROUSE, Sidney B. (*M* 1921) Consulting Engr. (for mail) S. B. & B. H. Strouse, 500-520 Guarantee Trust Bldg., and 22 S. Illinois Ave., Atlantic City, N. J.

STRUNIN, Jay (*A* 1939; *J* 1933) Engr. & Contractor (for mail) Strunin Plumbing & Heating Co., 408 Second Ave., and 54 West 89th St., New York, N. Y.

STUART, Milton C.* (*M* 1935) Prof. of Mech. Engrg. (for mail) Dept. of Mech. Engrg., Lehigh University, and 1828 Jennings St., Bethlehem, Pa.

STUART, W. W. (*A* 1940) Owner (for mail) W. W. Stuart Co., 417 Ninth St., and 1920 Pleasant St., Des Moines, Ia.

STUBBS, William Carlisle (*M* 1934) Assoc. Naval Archt. (for mail) Norfolk Navy Yard, and 39 Channing Ave., Portsmouth, Va.

STURDY, Oswald C. (*M* 1938) Sales Engr. (for mail) Foster Wheeler, Ltd., Commerce & Transportation Bldg., and 139 Indian Rd., Toronto, Ont., Canada.

STURM, William (*J* 1937; *S* 1936) Draftsman, Gausman & Moore, Cons. Engrs., 1026 First National Bank Bldg., and (for mail) 1822 Ashland, St. Paul, Minn.

SUDDERTH, Leo, Jr. (*J* 1936) Br. Mgr. (for mail) Johnson Service Co., 311 Bona Allen Bldg., and 1115 Los Angeles Ave. N. E., Atlanta, Ga.

SULLIVAN, Tim J. (*M* 1930) Pres., Sullivan Valve & Engineering Co., 910 S. Arizona St., Butte, Mont.

SUNDERLAND, Richard P. (*A* 1938) Pres. (for mail) General Meters-Controls Co., 205 W. Wacker Dr., Chicago, and 936 Judson Ave., Evanston, Ill.

SUPPLE, Graeme B. (*M* 1934) Dist. Mgr. (for mail) American Blower Corp., 625 Architects & Builders Bldg., and 420 East 55th St., Indianapolis, Ind.

SUTCH, Harry C. (*A* 1940) Mgr., Plumbing & Htg. Dept., Goldblatt Bros., 3932 S. Wolcott St., and (for mail) 552 Briar Pl., Chicago, Ill.

SUTCLIFFE, Arthur G. (*M* 1922; *A* 1918) Chief Engr., Ilg Electric Ventilating Co., 2850 N. Crawford Ave., Chicago, and (for mail) 432 S. Delphia Ave., Park Ridge, Ill.

SUTFIN, George V. (*A* 1937) Sales Engr. (for mail) American Blower Corp., 1005-6 American Bldg., and 3270 Hildreth Ave., Cincinnati, O.

SUTHERLAND, David L. (*A* 1934) Pres. (for mail) Sutherland Air Conditioning Corp., 632 Builders Exchange, and 1815 S. Colfax Ave., Minneapolis, Minn.
SUTTER, Edgar E. (*A* 1936) Sales Engr., Mueller Brass Co., Port Huron, Mich., and (for mail) 6705 Sixth St. N. W., Washington, D. C.
SWAIN, William Leonard (*M* 1939) Dir. (for mail) Young Austen & Young, Ltd., 35 Uphill Rd., Mill Hill, London, N. W. 7, and St. Catherine's, Sandy Lodge Lane, Moor Park, Hertfordshire, England.
SWANEY, Carroll R. (*M* 1929; *J* 1921) Co-Partner (for mail) Gilbert Howe Gleason & Co., 28 St. Botolph St., Boston, and 43 Clyde St., Newtonville, Mass.
SWANSON, Donald F. (*J* 1938) Test Engr., Seeger Refrigerator Co., 850 Arcade St., St. Paul, and (for mail) 4316 Bloomington Ave., Minneapolis, Minn.
SWANSON, Earl C. (*A* 1935) Vice-Pres., Andersen Corporation, Bayport, Minn.
SWANSON, Nils W. (*A* 1936) Sales, McDonnell & Miller, 400 N. Michigan Ave., and (for mail) 2746 Morse Ave., Chicago, Ill.
SWEENEY, Robert H. (*A* 1939) Sales Engr., Minneapolis-Honeywell Regulator Co., 4740 Baum Blvd., and (for mail) 1201 Hillsdale Ave., Pittsburgh, Pa.
SWENEHART, Delmer W. (*A* 1940) Educational Director, Air Conditioning Training Corp., 789 Wick Ave., Youngstown, and (for mail) Cortland, O.
SWENSON, J. E. (*A* 1930) Dept. Mgr., Heating Div. (for mail) Minneapolis Gas Light Co., 739 Marquette Ave., and 4853 S. 14th Ave., Minneapolis, Minn.
SWINGLE, Wayne T. (*A* 1938) Pres., Hastings Air Conditioning Co., Inc., and (for mail) 918 W. Seventh St., Hastings, Nebr.
SWISHER, Stephen G., Jr. (*M* 1936; *A* 1934) Mgr. (for mail) The Trane Co., 1835 N. Third St., and 1711 E. Dean Rd., Milwaukee, Wis.
SYDOW, Louis J. (*M* 1936) Vice-Pres., Grossenbacher Furnace Co., Inc., 9416 W. Milton St., St. Louis, and (for mail) 9456 Midland Ave., Overland, Mo.
SYMONDS, Edward S. (*M* 1939) Dir., Abair Engineering Limited, 1 Devonshire Sq., London, E. C. 2, England.
SYSKA, Adolph G. (*M* 1933) Consulting Engr. (for mail) Syska & Hennessy, 144 East 39th St., New York, and 3 Alden Pl., Bronxville, N. Y.
SZEKELY, Ernest (*M* 1920) Pres. & Gen. Mgr. (for mail) Bayley Blower Co., 1817 South 66th St., Milwaukee, and 6026 W. Washington Blvd., Wauwatosa, Wis.
SZOMBATHY, L. R. (*A* 1930) Pres. (for mail) Ferguson Sheet Metal Works, Inc., 34 N. Florissant Blvd., and 3125 Hawthorne Blvd., St. Louis, Mo.

T

TAGGART, R. C.* (*M* 1912) Division of Architecture, New York State Dept. of Public Works, Albany, and (for mail) 14 Lyon Ave., Menands, Albany, N. Y.
TAHRY, Mahmoud El (*M* 1939) Engr. (for mail) Carrier Egypt, S. A. E., 37 Sharia Kasr El Nil, Cairo, and 30, Hishmat Pacha St., Zamalek, Cairo, Egypt.
TALIAFERRO, Robert R.* (*M* 1919) Service Engr., Carrier Corp., 300 S. Geddes St., and (for mail) c/o H. H. Marshall, 1307 E. Adams St., Syracuse, N. Y.
TALLIANOS, Peter C. (*A* 1938) Mgr., The Egyptian Wireless Co., 36 Nebi Daniel St., Alexandria, Egypt.
TALLMADGE, Webster (*M* 1924) Pres. (for mail) Webster Tallmadge & Co., Inc., 364 Glenwood Ave., East Orange, and 7 Claremont Pl., Montclair, N. J.
TANGER, Othon C. F. (*A* 1937) Dir. N. V. Technische Handelsmaatschappij "Renova", Rembrandtlaan 34, Arnhem, Netherlands.
TARR, Harold M. (*M* 1931) Htg., Vtg. and Air Cond. Engr., 21 Montague St., Arlington Heights, Mass.
TASKER, C.* (*M* 1935) Sr. Research Fellow (for mail) Ontario Research Foundation, 43 Queen's Park, and 737 Avenue Rd., Toronto, Ont., Canada.
TAVERNA, Fred F. (*M* 1928; *A* 1927; *J* 1924) Raisler Corp., 129 Amsterdam Ave., New York, N. Y., and (for mail) 1011 Palisade Ave., Union City, N. J.
TAYLOR, Edward M. (*A* 1934) Tech. Mgr. (for mail), Taylors, Ltd., 32A Lichfield St., and 51 Totara Rd., Christchurch, New Zealand.
TAYLOR, Fielding, Jr. (*J* 1938) Sales Engr. (for mail) American Machine & Metals, Inc., De-Bothezat Ventilating Div., 100 Sixth Ave., New York, and 83-02 Cornish Ave., Elmhurst, L. I., N. Y.
TAYLOR, Harold J. (*M* 1937) Owner, Harold J. Taylor, Htg. & Vtg., 17514 Greenlawn Ave., Detroit, Mich.
TAYLOR, R. F. (*M* 1915) Consulting Engr. (for mail) 910 Bankers Mortgage Bldg., and 2332 Watts Rd., Houston, Tex.
TAYLOR, Robert B. (*J* 1938) Office Mgr. (for mail) Buffalo Forge Co., 1305 Standard Bldg., Albany, and R. F. D., New Scotland, N. Y.
TAYLOR, Thomas E. (*J* 1937) Consulting Engr. (for mail) 307 Postal Bldg., and 7307 N. Wall, Portland, Ore.
TAZE, D. L. (*M* 1931) Mgr. (for mail) American Blower Corp., 1302 Swetland Bldg., Cleveland, and 19412 Winslow Rd., Shaker Heights, O.
TAZE, Edwin H. (*M* 1937) Br. Mgr. (for mail) American Blower Corp., 620 Court Square Bldg., Baltimore, and 28 Normal Terrace, Towson, Md.
TEASDALE, Lawrence A. (*M* 1926) Engr. (for mail) Yale University, University Service Bureaus, 20 Ashmun St., and 262 W. Rock Ave., New Haven, Conn.
TEELING, George A. (*M* 1930) Consulting Engr. (for mail) 1 Columbia Pl., Albany, and Box 81, Clarksville, N. Y.
TEMPLE, W. J. (*M* 1931) Engr., J. A. Temple Co., 108 Parkway, and (for mail) 1215 Reed St., Kalamazoo, Mich.
TEMPLIN, Charles L. (*M* 1921) Pres. (for mail) Carrier Atlanta Corp., 348 Peachtree St., and 781 Sherwood Rd. N. E., Atlanta, Ga.
TENKONOHY, Rudolph J. (*M* 1923) 3650 Shaw Blvd., St. Louis, Mo.
TENNANT, Raymond J. J. (*A* 1929) Engr. (for mail) Pittsburgh Business Properties, Inc., 2237 Oliver Bldg., and 752 N. Meadowcroft Ave., Mt. Lebanon, Pittsburgh, Pa.
TENNEY, Dwight (*M* 1932) Pres. and Chief Engr. (for mail) Tenney Engineering, Inc., 46 Farrand St., Bloomfield, and 33 Summit Rd., Verona, N. J.
TERHUNE, Ralph D. (*A* 1936) Sales Engr., American Radiator & Standard Sanitary Corp., 1747 Connecticut Ave. N. W., Washington, D. C., and (for mail) 504 Maple Ridge Rd., Bethesda, Md.
TERRY, Matson C. (*M* 1936) Pres. (for mail) Certified Products Co., 2014 North 14th St., and 1429 Potomac Dr., Toledo, O.
terWEEME, Albert (*A* 1938) Sales Engr., N. V. Radiatoren, Singel 206-208, Amsterdam C, and (for mail) Archimedesweg 13, Amsterdam O, Holland.
THEISS, Ernest S. (*A* 1941; *J* 1940) Instructor in Mech. Engrg. (for mail) Duke University, P. O. Box 264, College Station, Durham, N. C.
THEOBALD, Art (*A* 1937) Research Engr. (for mail) Payne Furnace & Supply Co., Inc., 336 N. Foothill Rd., Beverly Hills, and 116½ S. Kings Rd., Los Angeles, Calif.

THEORELL, Axel T. (*M* 1939) Civil Engr., Theorells Ingeniorsbyra, Skoldungagatan 4, and (for mail) Lokattsvagen 41, Appelviken, Stockholm, Sweden.

THEORELL, Hugo G. T.* (*Life Member*; *M* 1902) Consulting Engr., Hugo Theorells Ingeniorsbyra, Skoldungagatan 4, Stockholm, Sweden.

THINN, C. A.* (*M* 1921) C. A. Dunham Co., 450 E. Ohio St., Chicago, Ill.

THOM, Arthur J. (*J* 1939) Htg. & Vtg. Engr., Johnson Service Co., 507 E. Michigan St., and (for mail) 3037 South 39th St., Milwaukee, Wis.

THOM, George B. (*M* 1937) Asst. Prof. Mech. Engrg. (for mail) Swarthmore College, Swarthmore, Pa.

THOMAN, Estell O. (*A* 1938) Heat & Air Cond. Engr., Boot & Co., 115 Fulton St. W., and (for mail) 403 College St. S. E., Grand Rapids, Mich.

THOMAS, Arthur E. (*J* 1938) Contracts Mgr., Young, Austin & Young, Ltd., H35/36 Exchange Bldgs., and (for mail) "Arlen" 56, Thingwall Rd., Wavertree, Liverpool, 15, England.

THOMAS, Bernard A. (*A* 1937; *J* 1923) Mgr. Htg. Dept., Crane Co., and (for mail) 405 E. Idlewild Ave., Tampa, Fla.

THOMAS, Glegge (*M* 1923) Office Mgr. (for mail) Clarage Fan Co., 723 Albee Bldg., Washington, D. C., and 7 W. Leland St., Chevy Chase, Md.

THOMAS, L. G. Lee (*M* 1934) Vice-Pres., Economy Pumps, Inc., 1000 Weller Ave., Hamilton, O.

THOMAS, Melvern F. (*M* 1909) Consulting Engr. (for mail) 24 Bloor St. W., and 74 Rivercrest Rd., Toronto, Ont., Canada.

THOMAS, Norman A. (*M* 1928) Pres., Thomas Heating Co., 142 South 14th St., La Crosse, Wis.

THOMAS, Ralph C. (*A* 1938) Pres., Thomas Air Conditioning, Inc., 819 Westover Ave., Norfolk, Va.

THOMAS, Richard H. (*Life Member*; *M* 1920) Pres. (for mail) Economy Pumps, Inc., 2522 W. Congress St., Chicago, and 426 Forest Ave., Oak Park, Ill.

THOMPSON, Charles (*A* 1927) Owner, Builders Wholesale Supply Corp., Box 1669, Reno, Nev.

THOMPSON, Edward B. (*A* 1938) Supervisor, Htg. Engrg., Cincinnati Gas & Elec. Co., Fourth and Main Sts., and (for mail) 1198 Coronado Ave., Cincinnati, O.

THOMPSON, Frank (*M* 1935) Gen. Supt. (for mail) Vulcan Iron Works, Ltd., and 653 Kingsway Ave., Winnipeg, Man., Canada.

THOMPSON, Nelson S.* (*Life Member*; *M* 1917; *J* 1897) 1615 Hobart St. N. W., Washington, D. C.

THOMSEN, Nis B. (*M* 1938) 168 Fallingbrook Rd., Toronto, Ont., Canada.

THOMSON, Thomas N.* (*Life Member*; *M* 1899) Htg. & Plbg. Consultant, 37 Irwin Pl., Huntington, L. I., N. Y.

THORNBURG, Harold A. (*M* 1932; *J* 1929) Chief Engr., N. V. Gebr. van Swaay Mij, Soerabaja, Java, N. E. I., and (for mail) 1911 Dorchester Rd., Apt. 6 G, Brooklyn, N. Y.

THORNTON, Thaddeus L. (*M* 1937) Maintenance Engr., Prudential Insurance, 96 Barclay St., Newark, and (for mail) 37 Perry St., Belleville, N. J.

THORNTON, William Barry* (*M* 1931) Sales Engr. (for mail) Carrier Corp., Merchandise Mart, and 8314 Indiana Ave., Chicago, Ill.

THRUSH, Homer A. (*M* 1918) Pres., H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.

THULMAN, Robert Kelley* (*M* 1938) Mech. Engr., Federal Housing Administration, Vermont and K Sts. N. W., Washington, D. C., and (for mail) 6505 Ridgewood Ave., Chevy Chase, Md.

THUNEY, Francis M. (*A* 1939; *J* 1936) Application Engr. (for mail) Wm. E. Kingswell, Inc., 3707 Georgia Ave. N. W., Washington, D. C., and 202 Glenwood Rd., Bethesda, Md.

TIDMARSH, Patrick M. (*M* 1938) Vice-Pres. and Gen. Mgr. (for mail) Tidmarsh Engineering Co., P. O. Box 2425, and 2626 E. Fourth St., Tucson, Ariz.

TIERNAN, James P. (*A* 1940) Pres., Krauss Engineering Co., 120 Locust St., Roselle, and (for mail) 129 Union Rd., Roselle Park, N. J.

TILLER, Louin (*A* 1935; *S* 1933) Air Cond. Engr., Oklahoma Gas & Electric Co., 321 N. Harvey, and (for mail) 2712 Northwest 15th St., Oklahoma City, Okla.

TILLINGHAST, Harry S. (*J* 1941; *S* 1940) Chief Time Study (for mail) Detroit Extrusion Plant, Aluminum Co. of America, 3311 Dunn Rd., and 60 W. Euclid, Detroit, Mich.

TILTON, Neil K. (*A* 1939) Pres., Mayflower Air-Conditioners, Inc., Newcomb and East Seventh St., and (for mail) 876 Osceola Ave., St. Paul, Minn.

TILTZ, Bernard E. (*M* 1930) Pres. (for mail) Tiltz Air Conditioning Corp., 230 Park Ave., New York, and 25 Lookout Circle, Larchmont, N. Y.

TIMMIS, Pierce (*M* 1920) Service Equip. Engr. (for mail) United Engineers & Constructors, Inc., 1401 Arch St., Philadelphia, and 202 Midland Ave., Wayne, Pa.

TIMMIS, W. Walter (*M* 1933; *A* 1925) Vice-Pres. (for mail) Herske & Timmis, Inc., 33 West 60th St., New York, and Woodland Dr., Pleasantville, N. Y.

TJERSLAND, Alf (*M* 1916; *J* 1906) E. Sunde & Co., Ltd., Oslo, Norway.

TOBIN, John F. (*A* 1934) Sales (for mail) American Blower Corp., 228 N. LaSalle St., and 11256 S. Artesian Ave., Chicago, Ill.

TODD, Meryl L. (*M* 1940; *J* 1936) Mech. Engr. (for mail) 1111 Independence Ave., and 100 Highland Blvd., Waterloo, Ia.

TODD, Stanton W., Jr. (*A* 1939; *J* 1935) Repr., American Radiator & Standard Sanitary Corp., 8019 Joseph Campau St., Detroit, and (for mail) 309 Paris S. E., Grand Rapids, Mich.

TOENSFELDT, Ralf (*M* 1940) Consulting Engr. (for mail) 411 Security Bldg., and 6311 Waterman Ave., St. Louis, Mo.

TOLHURST, George C. (*M* 1936) Htg. Draftsman, Warden King, Ltd., Bennett Ave., Montreal, and (for mail) 142 Blvd. St., Germain, St. Laurent (near Montreal), Que., Canada.

TONE, Jay E., Jr. (*A* 1940) Pres. (for mail) Tone Air Conditioning Co., 533 Seventh St., and 651 36th St., Des Moines, Ia.

TONRY, Robert C. (*M* 1936) Mgr. (for mail) Wiedebusch Plumbing & Heating Co., 511 First St., and 217 Fairmont Ave., Fairmont, W. Va

TOONDER, Clarence L. (*M* 1933) Mech. Engr., Argonaut Div., General Motors Corp., General Motors Research Bldg., and (for mail) 13391 Marlowe Ave., Detroit, Mich.

TORNQUIST, Earl L. (*A* 1934) Test Engr. (for mail) Public Service Co. of Northern Illinois, 72 W. Adams St., Chicago, and 465 Parkside Ave., Elmhurst, Ill.

TOROK, Elmer (*M* 1936) Supt. of Power (for mail) North American Rayon Corp., and 203 West G St., Elizabethton, Tenn.

TORR, Thomas W. (*M* 1933) Chief Engr., The Rudy Furnace Co., and (for mail) 205 Green St., Dowagiac, Mich.

TOULOUKIAN, Yeram S. (*S* 1939) Research Asst. (for mail) Massachusetts Institute of Technology, M. I. T. Graduate House, Cambridge, Mass., and Istanbul, Turkey.

TOUTON, Rush D. (*M* 1933) Tech. Dir. (for mail) Bayuk Cigars, Inc., Ninth and Columbia Ave., Philadelphia, and 19 Lodges Lane, Bala-Cynwyd, Pa.

TOWER, Elwood S. (*M* 1930) Consulting Engr. (for mail) 213 Investment Bldg., and 5516 Woodmont St., Pittsburgh, Pa.

TOWLE, Philip Hamilton (*J* 1938) Air Cond. Engr. (for mail) General Air Conditioning Co., 1313 Jay St., and 942-42nd St., Sacramento, Calif.

TOWNE, Charles O. (*J* 1938) Engr. (for mail) Clowe & Cowan, Inc., 401 Harrison St., and 3020 W. Ninth Ave., Amarillo, Tex.

TRACY, William E. (*J* 1938) Sales Engr. (for mail) B. F. Sturtevant Co., 404 Wrigley Bldg., Chicago, and Glen Ellyn, Ill.

TRAMBAUER, Charles W. (*A* 1941; *J* 1936) Sales Engr., Hoffman Specialty Co., 500 Fifth Ave., New York, and (for mail) 145-50-18th Ave., Whitestone, L. I., N. Y.

TRANE, Reuben N.* (*M* 1915) Pres. (for mail) The Trane Co., and 126 South 15th St., LaCrosse, Wis.

TRAUGOTT, Mortimer (*A* 1930) Mgr. (for mail) Bryant Air Conditioning Corp., 915 N. Front St., Philadelphia, and 8208 Westminster Rd., Elkins Park, Pa.

TRAYNOR, Harry S. (*J* 1937) Engr., Carrier Corp., S. Geddes St., and (for mail), 594 Roberts Ave., Syracuse, N. Y.

TREADWAY, J. Quentin (*A* 1936; *J* 1932) Dist. Sales Mgr. (for mail) Clarage Fan Co., 210 Reynolds Arcade, and 826 Winona Blvd., Rochester, N. Y.

TRENNER, Kelvin (*A* 1938) Dist. Mgr., Steel Products Engineering Co., Springfield, and (for mail) 500 Parkview St., Mansfield, O.

TRIGGS, Fred E. (*M* 1938) Mfrs. Agt. (for mail) Box 1, H. P. Station, and 3901 Second St., Des Moines, Ia.

TROLESE, Louis G. (*A* 1940) Engr. (for mail) General Air Conditioning & Heating Co., 4001 Piedmont Ave., and 357 Palm Ave., Oakland, Calif.

TROSTEL, Otto A. (*M* 1935) Engr., Standard Distributing Co., 406 E. Wells St., Milwaukee, and (for mail) Route No. 2, Thiensville, Wis.

TROUP, John D. (*M* 1938) Managing Dir. (for mail) John D. Troup, Ltd., 90 High Holborn, London, W. C. 1, and 48 Plough Lane, Purley, Surrey, England.

TRUMBO, S. M. (*A* 1926) Sales (for mail) Buffalo Forge Co., 20 N. Wacker Dr., Chicago, and 921 Franklin St., Downers Grove, Ill.

TRZOS, Otto A. (*A* 1940; *J* 1938) Industrial Gas Engr. (for mail) Consumers Power Co., 2850 Chadwick, Pontiac, and Box 103, Keego Harbor, Mich.

TUCKER, Frank N. (*M* 1926) Field Engr., Ilg Electric Ventilating Co., 13 Park Row, Rm. 11, New York, and (for mail) 239 Whaley St., Freeport, L. I., N. Y.

TUCKER, Leonard A. (*M* 1935) Service Sales Mgr., J. J. Pocock, Inc., 31st and Jefferson Sts., Philadelphia, and (for mail) 518 Monroe Ave., Ardsley, Pa.

TUCKER, Ralph E. (*M* 1940) Chief Draftsman (for mail) York Ice Machinery Corp., 2201 Texas Ave., and 5220 Hillman St., Houston, Tex.

TUCKER, Thomas T. (*M* 1938; *A* 1936) Pres., Armor Insulating Co., 800 Forrest St. N. W., and (for mail) 3619 Ivy Rd., Atlanta, Ga.

TUCKERMAN, George E. (*M* 1932) Mgr. (for mail) Anderson Conditioning Co., 600 Schuylkill Ave., Philadelphia, and 502 Rodman Ave., Jenkintown, Pa.

TUMPANE, James P., Jr. (*S* 1939) Student (for mail) Carnegie Institute of Technology, 5029 Morewood Pl., Pittsburgh, Pa., and Big Flats, N. Y.

TURLAND, Charles H. (*M* 1934; *A* 1930) Sales Engr. (for mail) R. E. Johnston Co., Ltd., 1070 Homer St., and 4553 W. Third Ave., Vancouver, B. C., Canada.

TURNER, Edmond S. (*A* 1939) Sales Engr., Wm. S. Turner & Co., 311 Pacific Bldg., and (for mail) 3455 Northeast 36th Ave., Portland, Ore.

TURNER, George G. (*A* 1934) Western Repr., Heating & Ventilating (for mail) The Industrial Press, 228 N. LaSalle St., Chicago, and 827 Hinman Ave., Evanston, Ill.

TURNO, Walter G. (*M* 1917; *A* 1912) Secy., H. W. Porter & Co., Newark, and (for mail) 71 Lafayette Ave., East Orange, N. J.

TUSCH, Walter (*M* 1917) Secy., Tenney & Ohmes, Inc., 101 Park Ave., New York, and (for mail) 881 Sterling Pl., Brooklyn, N. Y.

TUTHILL, Arthur F.* (*J* 1940; *S* 1938) Instructor Mech. Engrg. (for mail) The Cooper Union, Cooper Square, New York, and Cutchogue, L. I., N. Y.

TUTSCH, Rodney J. (*J* 1939) Asst. Mgr. (for mail) Iron Fireman of Milwaukee, Inc., 4507 W. Wisconsin Ave., and 4455 N. Oakland Ave., Milwaukee, Wis.

TUTTLE, George H.* (*M* 1937; *A* 1936; *J* 1934) Htg. Engr. (for mail) The Detroit Edison Co., 2000 Second Ave., and 16714 Kentfield, Detroit, Mich.

TUTTLE, J. Frank (*M* 1913) Sales Agent (for mail) Warren Webster & Co., 127 Federal St., Boston, and 9 Lewis Rd., Winchester, Mass.

TUVE, George L.* (*M* 1932) (Council 1939-40) Prof. of Heat-Power Engrg. (for mail) Case School of Applied Science, Cleveland and 2510 Newbury Dr., Cleveland Heights, O.

TUXHORN, David B. (*M* 1936) Mgr., The York Heat & Fuel Co., 935 Shoreham Bldg., and (for mail) 4853 Sedgwick St. N. W., Washington, D.C.

TWIST, Charles F. (*M* 1921) Pres. (for mail) Ashwell-Twist Co., 967 Thomas St., Seattle, Wash.

TWIZELL, Edwin W. (*M* 1937) Partner (for mail) Connolly & Twizell Reg'd., 1405 Bishop St., and 5176 Westbury Ave., Montreal, Que., Canada.

TYLER, Roy D. (*M* 1928) Mgr. (for mail) Modine Mfg. Co., 101 Park Ave., Rm. 1734, New York, N. Y., and 5 Ferris Dr., Old Greenwich, Conn.

U

UHL, Edwin J. (*M* 1925) Partner (for mail) Uhl Co., 132 S. Tenth St., and 4830 Pleasant Ave. S., Minneapolis, Minn.

UHL, Willard F. (*M* 1918) Partner (for mail) Uhl Co., 132 S. Tenth St., and 4716 Lyndale Ave. S., Minneapolis, Minn.

UHLHORN, W. J. (*M* 1920) 733 S. Highland Ave., Oak Park, Ill.

ULLRICH, Anton B., Jr. (*J* 1937) Sales Engr. (for mail) Gilbert Engineering Co., 1305 Liberty Bank Bldg., and 6012 Lewis, Dallas, Tex.

UPSON, Walter L. (*M* 1938) Dir. of Research (for mail) Torrington Manufacturing Co., Torrington, and Litchfield, Conn.

URBAN, Frank F. (*A* 1939) Vice-Pres., Urban Plumbing & Heating Co., 1215 S. W. Fifth Ave., and (for mail) 6726 S. W. Burlingame Ave., Portland, Ore.

URDAHL, Thomas H. (*M* 1930) (Council 1940) Consulting Engr. (for mail) 726 Jackson Pl. N. W., and 1505-44th St. N. W., Washington, D. C.

V

VALE, Henry A. L. (*M* 1929) Managing Dir. (for mail) Vale Company, Ltd., 141-143 Armagh St., Christchurch, and 203 Llam Rd., Fendalton, Christchurch, New Zealand.

VAN ALSBURG, Jerold H.* (*M* 1931) Sales Engr. (for mail) Hart & Cooley Manufacturing Co., 61 W. Kinzie St., Chicago, Ill., and Westfield, N. Y.

VANCE, Louis G. (*M* 1919) Mfrs. Agent, 15 East 21st St., and (for mail) 4402 Maine Ave., Forest Park, Baltimore, Md.

VANDERHOOF, A. L. (*A* 1933) No. Ohio Repr. (for mail) Warren Webster & Co., 2341 Carnegie Ave., Cleveland, and 2762 Landon Rd., Shaker Heights, O.

VAN NOUHUYS, Herbert C. (*J* 1937) Engr., Detroit Southern Pipe Line Co., 9332 Buffalo Blvd., Hamtramck, and (for mail) 2007 Seward Ave., Detroit, Mich.

VAN WYNGARDEN, J. Edwin (*A* 1940) Gen. Mgr. (for mail) Day Heating Co., 339 N. Commercial St., and 945 N. Church St., Salem, Ore.

VAUGHAN, John G., Jr. (*J* 1935) Asst. Engr., U. S. Housing Authority, Region I, N. Interior Bldg., Washington, D. C., and (for mail) Franklin Park, East Falls Church, Va.

VAUGHAN, Lillian Lee (*M* 1938) Prof. of Mech. Engrg. and Head of Dept. (for mail) North Carolina State College, State College Station, and 11 Enterprise St., Raleigh, N. C.

VAUGHN, Frank R. (*M* 1938; *A* 1936) Vice-Pres. (for mail) Green Colonial Furnace Co., and 532 Polk Blvd., Des Moines, Ia.

VEGLERY, Alexander (*A* 1939) Engr. in Shipping Dept. (for mail) Socony-Vacuum Oil Co., Inc., P. O. Box 660, and Dirhem sokak, No. 11, Yenikoy, Bogazici, Istanbul, Turkey.

VERNON, J. Rexford (*M* 1928; *A* 1926) Advertising Mgr. (for mail) Johnson Service Co., 1355 Washington Blvd., Chicago, and 733 Brummel St., Evanston, Ill.

VERVOORT, Edward L. (*J* 1937; *S* 1936) Dist. Htg. Supvr. (for mail) Brooklyn Union Gas Co., 180 Remsen St., Brooklyn, and 31 Yale Pl., Rockville Centre, L. I., N. Y.

VETLESEN G. Unger (*M* 1930) Consulting Engr., 1 Beekman Pl., New York, N. Y.

VIDALE, Richard (*M* 1935) Mech. Engr. (for mail) Flesch & Schmitt, Inc., 118 Brown St., and 92 Harding Rd., Rochester, N. Y.

VINCENT, Paul J. (*M* 1931) Owner (for mail) Paul J. Vincent Co., 2208 Maryland Ave., and 3807 Beech Ave., Baltimore, Md.

VINSON, Neal L. (*J* 1936; *S* 1932) Engr., Estimator and Partner (for mail) L. W. Vinson & Son, Box 3007, Lowell, and Bisbee, Ariz.

VIRRILL, George A. (*A* 1940) Chief Engr., The University Club, 1-3 West 54th St., New York, and (for mail) 345 Washington Ave., New Rochelle, N. Y.

VISSAC, Gustave A. (*M* 1937) Consulting Engr., 1475 West 25th, Vancouver, B. C., Canada.

VIVARTTAS, E. Arnold (*Life Member*; *M* 1910) Mech. Engr., 154 Maine Ave., West New Brighton, S. I., N. Y.

VOISINET, Walter E. (*M* 1930) Air Conditioning Specialties (for mail) 250 Delaware Ave., Buffalo, and 151 Warren Ave., Kenmore, N. Y.

VOLK, George H. (*S* 1940) Student, University of Wisconsin, Madison, and (for mail) 2965 South 43rd St., Milwaukee, Wis.

VOLK, Joseph H. (*M* 1923) Pres. & Treas. (for mail) Thos. E. Hoye Heating Co., 1906 W. St. Paul Ave., and 2965 South 43rd St., Milwaukee, Wis.

VOLKHARDT, Aquila N. (*M* 1938) Owner (for mail) A. N. Volkhardt, 942 Bay St., Rosebank, and 104 Townsend Ave., Stapleton, S. I., N. Y.

VOLLMANN, Carl W. (*M* 1938) Pres. & Gen. Mgr. (for mail) Linde Canadian Refrigeration Co., Ltd., 355 St. Peter St., Montreal, and 517 Roslyn Ave., Westmount, Que., Canada.

vonCHRISTIERSON, Carl A. (*A* 1939; *J* 1937) Resident Engr., Carrier Engineering S. A., Ltd., P. O. Box 2421, Durban, South Africa.

vonROSENBERG, Paul C. (*J* 1939) Engr. (for mail) Illinois Engineering Co., 2035 S. Racine Ave., and 1221 Hood Ave., Chicago, Ill.

VOORHEES, G. A. (*M* 1922) Mgr. (for mail) Furblo Co., and P. O. Box 63, Hermansville, Mich.

VOSS, Walter W. (*A* 1938) Instruction Engr. (for mail) Utilities Engineering Institute, 404 N. Wells St., and 1347 N. Dearborn St., Chicago, Ill.

VROOME, Albert E. (*M* 1932) (for mail) Ebasco Services, Inc., 2 Rector St., New York, and 6218 Amboy Rd., Prince Bay, S. I., N. Y.

W

WACHS, Louis J. (*A* 1936; *J* 1930) Sales Engr., Carrier Corp., 405 Lexington Ave., New York, and (for mail) 1820 Cortelyou Rd., Brooklyn, N. Y.

WADSWORTH, Raymond H. (*J* 1937) Air Cond. Sales Engr. (for mail) Clarage Fan Co., 500 Fifth Ave., New York, N. Y., and 112 Summit St., East Orange, N. J.

WAECHTER, Herman P. (*A* 1930; *J* 1927) Air Cond. Engr., W. T. Grant Co., 1441 Broadway, and (for mail) 126 East 31st St., New York, N. Y.

WAGGONER, Jack H. (*M* 1937) Product Control Supvr., Owens-Corning Fiberglas Corp., and (for mail) 214 Rugg Ave., Newark, O.

WAGNER, Earle K. (*M* 1938) Sales Engr. (for mail) The Powers Regulator Co., 2240 N. Broad St., Philadelphia, and 312 Myrtle Ave., Cheltenham, Pa.

WAGNER, Edward A. (*M* 1937; *A* 1936) Pres., Wagner Engineering Corp., 22 Dunham St., and (for mail) 28 Waverly St., Pittsfield, Mass.

WAHRENBROCK, Orin K. (*A* 1939; *J* 1936) Lieut., U. S. N. R., Office of Inspector of Naval Materials, San Francisco, and (for mail) 410 Fairmount Ave., Oakland, Calif.

WAID, Glen H. (*A* 1930) Dist. Sales Mgr., Scott Valve Manufacturing Co., 3963 McKinley Ave., and (for mail) 2928 Northwestern Ave., Detroit, Mich.

WALDEN, H. Kenneth (*J* 1939) Secy. (for mail) Haydn Myer Co., Inc., 2224 Comer Bldg., and 512 Sixth St. S. W., Birmingham, Ala.

WALDON, Charles D. (*A* 1932) Consulting Engr., Spencer Foundry Co., Penetang, and (for mail) 32 Ferndale Ave., Toronto, Ont., Canada.

WALDREP, James E. (*J* 1939) Plant Engr. and Asst. Supt., Greater Greenville Sewer Commission, Box 1416, and (for mail) Mount Vista Ave., Rte. 4, Greenville, S. C.

WALFORD, L. C. A. (*M* 1938) 4264 Royal Ave., Notre Dame De Grace, Montreal, Que., Canada.

WALKER, Edmund R. (*M* 1934) Mgr., Air Cond. Div. (for mail) Fedders Manufacturing Co., Inc., 57 Tonawanda St., Buffalo, and 305 McKinley Ave., Kenmore, N. Y.

WALKER, J. Herbert* (*M* 1916) (Council 1938-40 Engr. Asst. to Gen. Mgr. (for mail) The Detroit Edison Co., 2000 Second Ave., Detroit, and 432 Arlington Rd., Birmingham, Mich.

WALLACE, George J. (*M* 1923) Principal Engr. and Contractor, 96-19-35th Ave., Corona, and (for mail) 27-36 Ericsson St., East Elmhurst, L. I., N. Y.

WALLACE, Harry P., Jr. (*A* 1936) Mgr., Sales Promotion, Crane Co., 400 Third Ave. N., and (for mail) 4909-34th Ave. S., Minneapolis, Minn.

WALLACE, William M., II (*M* 1929) Resident Partner (for mail) Syska & Hennessy, Cons. Engrs., 111 N. Corcoran St., and 2603 Highland Ave., Durham, N. C.

WALLIS, Walter M. (*A* 1940) 314-9 N., and (for mail) 11225-10 S. W., Seattle, Wash.

WALSH, Edward R., Jr. (*M* 1936; *A* 1935) Natl. Supvr., Htg. & Air Cond., York Ice Machinery Corp., and (for mail) Wyndham Hills, York, Pa.

WALSH, James A. (*A* 1932; *J* 1929) Pres. (for mail) Air Conditioning Co., 1017 Sampson St., and 513 Branard St., Houston, Tex.

WALTERS, Arthur L. (*M* 1926; *A* 1925; *J* 1924) Chief Engr. (for mail) Green Colonial Furnace Co., 322 S. W. Third St., and 900-29th St., Des Moines, Ia.

WALTERS, William T. (*M* 1917) Engr., Illinois Engineering Co., Cor. 21st St. and Racine Ave., and (for mail) 12747 Wallace St., Chicago, Ill.

WALTERTHUM, John J. (*A* 1922) Owner, John J. Walterthum, 212 East 58th St., New York, N. Y., and (for mail) 42-A Van Reipen Ave., Jersey City, N. J.

WALTON, Charles W., Jr. (*M* 1934) Mech. Engr. (for mail) Rockefeller Center, Inc., 50 Rockefeller Plaza, New York, N. Y., and 120 Monte Vista Ave., Ridgewood, N. J.

WALZ, Chester D. (*A* 1939) Mech. & Elec. Engr., 2835 Gilroy St., Los Angeles, and (for mail) 709 25th St., Santa Monica, Calif.

WALZ, George R. (*A* 1940; *J* 1937) Sales Engr. (for mail) Minneapolis-Honeywell Regulator Co., 1101 Vermont Ave. N. W., Washington, D. C., and 1808 Queen's Lane, Apt. 209, Arlington, Va.

WARD, Edward B. (*M* 1937) Pres. (for mail) Edward B. Ward & Co., 270 Fremont St., and 235 Lansdale Ave., San Francisco, Calif.

WARD, Frank J. (*M* 1935) Owner (for mail) Frank J. Ward Co., 237 W. Court St., Cincinnati, O., and Cold Spring, Ky.

WARD, Harry H. (*A* 1937) 1003 N. Mills St., Orlando, Fla.

WARD, Oscar G. (*M* 1919) Vice-Pres. (for mail) Johnson Service Co., 1355 Washington Blvd., Chicago, and 1345 Ashland Ave., Wilmette, Ill.

WARDELL, Arthur (*M* 1935) Asst. Prof. of Engrg. Drawing, University of Toronto, and (for mail) 124 Melrose Ave., Toronto, Ont., Canada.

WARE, John H., III (*M* 1937) Vice-Pres., Citizens Gas & Fuel Co., Pres., Oxford Co., Vice-Pres., Gas Oil Products, Inc. (for mail) 45 S. Third St., and "The Woods," Oxford, Pa.

WARING, J. M. S. (*M* 1932) Consulting Engr., 277 Park Ave., New York, N. Y.

WARREN, Francis C. (*M* 1934) Br. Mgr. (for mail) American Blower Corp., 200 Division Ave. N., and 329 Gladstone Ave. S. E., Grand Rapids, Mich.

WARREN, Hugh P. (*S* 1940) Lab. Asst. (for mail) Texas Engineering Experiment Station, and Box 712, College Station, Tex.

WARREN, Robert M., Jr. (*J* 1938) Chief Engr. (for mail) Page-Williamson, Inc., 228 W. First St., and 522 Fenton Pl., Charlotte, N. C.

WASHINGTON, Laurence W. (*M* 1929) Dist. Mgr. (for mail) The Powers Regulator Co., 702 American Bldg., and 1627 Northwood Dr., Cincinnati, O.

WASSER, Munny (*M* 1938) Htg., Vtg., & Air Cond. Engr., Vasile Lascar 8 et II, Bucharest, Roumania.

WASSON, Robert A. (*M* 1938) Eastern Dist. Mgr. (for mail) Clarage Fan Co., 500 Fifth Ave., New York, and 15 Willow St., Brooklyn, N. Y.

WATERMAN, John H. (*M* 1931) (for mail) Charles T. Main, Inc., 201 Devonshire St., Boston, Mass.

WATERS, George G. (*M* 1931; *A* 1926) Dist. Mgr. (for mail) American Blower Corp., 1433 Oliver Bldg., and 110 Longue Vue Dr. (16), Pittsburgh, Pa.

WATKINS, George B. (*A* 1936) Dir. of Research (for mail) Libbey-Owens Ford Glass Co., 1701 E. Broadway, and 4941 Rolandale Rd., Box 217-C. R. R. 8, Miner Park Subdivision, Toledo, O.

WATSON, Kenneth W. (*A* 1939) Mgr. of Htg. Dept., Crane Co., 710 Northwest 14th Ave., and (for mail) 11012 N. E. Prescott St., Portland, Ore.

WATT, R. M., Jr. (*M* 1939) Lt. Comdr., U. S. N., Officer in Charge Air Cond. Section, Bureau of Ships, Design Div. (for mail) Navy Dept., Rm. 2120, Washington, D. C., and 2328 S. Nash St., Arlington, Va.

WATT, Robert D. (*J* 1937) Engr., H. W. Beecher, Cons. Engr., Securities Bldg., Pres., Electrol Oil Burner Corp., 314 Stewart St., and (for mail) 3617-47th Ave. N. E., Seattle, Wash.

WATTS, Albert E. (*A* 1937) Mgr., A. E. Watts, 637 Craig St. W., and (for mail) 2347 Beaconsfield Ave., N. D. G., Montreal, Que., Canada.

WAUDBY, Walter (*M* 1938) Engr., American Radiator Co., 149 Blvd. Haussmann, Paris, and (for mail) 26 Rue de la Tourelle, Boulogne sur Seine, France.

WAYLAND, Clarke E. (*A* 1937) Vice-Pres. (for mail) Western Asbestos Co., 675 Townsend St., and 42 Allston Way, San Francisco, Calif.

WEATHERBY, Edward P., Jr. (*J* 1936; *S* 1935) 2223 Wilbargur St., Vernon, Tex.

WEAVER, John van O. (*M* 1940) Major, Post Exchange Officer, Air Corps, A. C. A. F. S., Kelly Field, Tex.

WEBB, Ernest C. (*M* 1935) Engrg. Service Mgr. (for mail) Iron Fireman Manufacturing Co., 3170 West 106th St., Cleveland, and 24721 Westlake Rd., Bay Village, O.

WEBB, John W. (*M* 1926) Managing Dir. (for mail) Webb Dust Removing & Drying Co., Vinery Works, Town Lane, Denton, N. Manchester, and "Ebor," Brinnington, Stockport, England.

WEBBER, Charles H. (*A* 1940) Sales Engr. (for mail) Pacific Scientific Co., 1206 Maple Ave., Los Angeles, and 1359 Topeka, Pasadena, Calif.

WEBER, Erwin L. (*M* 1921) Consulting Engr., 534 Medical Arts Bldg., Seattle, Wash.

WEBER, Eugene F., Jr. (*A* 1940; *J* 1937) Sales Engr., York Ice Machinery Corp., 117 South 11th St., and (for mail) 4405 W. Pine, St. Louis, Mo.

WEBSTER, Chester C. (*A* 1940) Pres. (for mail) John Hankin & Brother, 6 Church St., New York, and Piermont, N. Y.

WEBSTER, E. Kessler (*M* 1915) Secy. & Asst. Gen. Mgr. (for mail) Warren Webster & Co., 17th and Federal St., Camden, and First and Kings Highway, Haddon Heights, N. J.

WEBSTER, Warren, Jr. (*M* 1932; *J* 1927) Pres. (for mail) Warren Webster & Co., 17th and Federal St., Camden, and 200 Colonial Ridge Dr., Haddonfield, N. J.

WECHSBERG, Otto (*M* 1932) Pres. & Gen. Mgr. (for mail) Coppus Engineering Corp., 344 Park Ave., and 28 Lenox St., Worcester, Mass.

WEDDELL, George O. (*M* 1930) Br. Mgr., York Ice Machinery Corp., 2400 Carson St., and (for mail) 3114 Wainbell Ave., Dormont, Pittsburgh, Pa.

WEGMANN, Albert (*M* 1918) Owner, A. Wegmann Co., 2811-13 W. Fletcher St., and (for mail) 6206 North 17th St., Philadelphia, Pa.

WEID, Harry L. (*S* 1938) Test Engr., Hynes Electric Heating Co., 240 Cherry St., Philadelphia, Pa., and (for mail) Blackwood, N. J.

WEIL, F. H. Eugene (*A* 1938) Plbg., Htg. & Air Cond., The Maag Co., 831 N. Milwaukee St., and (for mail) 2515 North 59th St., Milwaukee, Wis.

WEIL, Leo S. (*M* 1940) Consulting Engr. (for mail) Leo S. Weil & Walter B. Moses, 425 S. Peters St., and 478 Broadway, New Orleans, La.

WEIL, Martin (*A* 1925) Vice-Pres. (for mail) Weil-McLain Co., 641 W. Lake St., and 4259 Hazel St., Chicago, Ill.

WEIL, Maurice I. (*A* 1928) Pres. (for mail) Chicago Pump Co., 2336 Wolfram St., and 3300 Lake Shore Dr., Chicago, Ill.

WEIMER, Fred G. (*A* 1919) Office Mgr. (for mail) Kewanee Boiler Corp., 312 E. Wisconsin Ave., and 3058 N. Stowell Ave., Milwaukee, Wis.

WEINERT, Fred C. (*A* 1937) Asst. Sales Mgr. (for mail) Chamberlin Metal Weather Strip Co., 1254 Labrosse St., Detroit, and Route No. 2, Plymouth, Mich.

WEINSHANK, Theodore* (*Life Member*; *M* 1906) (Board of Governors, 1913) Consulting Engr., Htg., Vtg. & Cooling, 3307 Belden Ave., Chicago, Ill.

WEISS, Arthur P. (*M* 1928) Burnham Boiler Corp., Irvington, and (for mail) 134 Farrington Ave., North Tarrytown, N. Y.

WEISS, Carl A. (*M* 1936; *A* 1924) Gen. Mgr. (for mail) Kornbrodt Kornice Co., 1811 Troost Ave., and 29 East 68th St., Kansas City, Mo.

WEISS, Walter G. (*J* 1940) Chief Engr. (for mail) St. Louis Furnace Manufacturing Co., 2901 ElliotAve., and 4571 Clarence Ave., St. Louis, Mo.

WEISSBLATT, Norman (*S* 1938) Student, New York Technical Institute, 108 Fifth Ave., New York, and (for mail) 1435–53rd St., Brooklyn, N. Y.

WEITZEL, Cameron B. (*M* 1936) Owner and Operator, Cameron B. Weitzel, 122 E. High St., Manheim, Pa.

WEITZEL, Paul H. (*J* 1936; *S* 1934) Jr. Engr., Cameron B. Weitzel, 122 E. High St., Manheim, Pa.

WELCH, Louis A., Jr. (*A* 1929) Owner (for mail) Welch Bros., 443 Second St., and 2001 Campbell Ave., Schenectady, N. Y.

WELDY, Lloyd O. (*M* 1930) Dist. Mgr. (for mail) The Powers Regulator Co., 2341 Carnegie Ave., Cleveland, and 19623 Laurel Ave., Rocky River, O.

WELLER, Albert K. (*A* 1939) (for mail) 1231 N. W. Hoyt St., and 3827 N. E. Davis St., Portland, Ore.

WELLS, Earl P.* (*M* 1938) Air Cond. Engr., Gay Engineering Corp., 2730 East 11th St., Los Angeles, and (for mail) 1133 Graynold Ave., Glendale, Calif.

WELLS, William Firth* (*M* 1939) Dir. Lab. for Study of Air-Borne Infection (for mail) University of Pennsylvania, School of Medicine, Philadelphia, and 112 Pine Ridge Rd., Media, Pa.

WENDT, Edgar F. (*M* 1918) Pres. (for mail) Buffalo Forge Co., 490 Broadway, and 120 Lincoln Parkway, Buffalo, N. Y.

WENDT, Edwin H. (*J* 1936) Engr. (for mail) O. A. Wendt Co., 2124 N. Southport Ave., and 4728 N. Lawndale Ave., Chicago, Ill.

WERKER, Herwart (*M* 1939) Engr., American Radiator & Standard Sanitary Corp., Institute of Thermal Research, 675 Bronx River Rd., and (for mail) 60 Ball Ave., Yonkers, N. Y.

WERNER, John G. (*M* 1937) Sales Engr., L. J. Mueller Furnace Co., Lafayette & Dickson Sts., Baltimore, Md., and (for mail) 4512–49th St. N. W., Washington, D. C.

WERNER, Philip H. (*A* 1941; *J* 1939) Br. Sales Mgr. (for mail) Barber-Colman Co., 914 N. Broadway, and 2967 North 78th St., Milwaukee, Wis.

WERNER, Richard K. (*M* 1936) Consulting Engr. (for mail) 316 W. T. Waggoner Bldg., Ft. Worth, and Jacksboro Highway, Lake Worth, Tex.

WESLEY, Ray O. (*A* 1937) Sales Engr. (for mail) U. S. Radiator Corp., 334 Boren Ave. N., Seattle, and Yarrow Point, Bellevue, Wash.

WEST, Perry* (*M* 1911) Prof. of Steam & Power Engrg., Head of Department of Mech. Engrg. (for mail) College of Engineering, University of Kentucky, and 185 E. Maxwell St., Lexington, Ky.

WESTENDARP, Francisco G. (*M* 1939) Engr. (for mail) Guinand Frères Sucs., Dept. de Refrigeracion, Carrier, Apartado 668, Caracas, Venezuela.

WESTOVER, Wendell (*M* 1936) Pres. (for mail) Westover-Wolfe, Inc., 170 Washington Ave., and 254 Lenox Ave., Albany, N. Y.

WESTPHAL, Norman E. (*J* 1940; *S* 1937) Engr., Northern Indiana Public Service Co., and (for mail) Long Beach, Michigan City, Ind.

WETHERED, Woodworth (*M* 1938) The Bohemian Club, San Francisco, Calif.

WETZELL, Horace E. (*M* 1934) Chief Engr. (for mail) The Smith & Oby Co., 6107 Carnegie Ave., Cleveland, and 21144 Aberdeen Rd., Rocky River, O.

WHEELER, Joe, Jr. (*M* 1938) Sales Repr. (for mail) Johnson Service Co., 28 East 29th St., New York, and 261 Dogwood Lane, Manhasset, L. I., N. Y.

WHELAN, William J. (*M* 1923) Purchasing and Estimating (for mail) Harrigan & Reid Co., 1365 Bagley Ave., and 3790 Seminole Ave., Detroit, Mich.

WHELLER, Harry S. (*M* 1916) Vice-Pres., L. J. Wing Manufacturing Co., 154 West 14th St., New York, and (for mail) 725 Union Ave., Elizabeth, N. J.

WHITE, Elmer D. (*A* 1939; *J* 1937) Wilbur Wright Field, Dayton, O.

WHITE, Elwood S. (*M* 1921) Pres. (for mail) U. S. Radiator Corp., 1056 National Bank Bldg., Detroit, Mich., and Meadowbank Rd., Old Greenwich, Conn.

WHITE, Eugene B. (*M* 1934) Arch. & Engrg. Bureau (for mail) 19 S. LaSalle St., Chicago, and 309 N. Taylor Ave., Oak Park, Ill.

WHITE, Everett G. (*A* 1938) Asst. Custodian Engr., Bronx Central Post Office, New York, and (for mail) 425 Rochelle Ter., Pelham Manor, N. Y.

WHITE, Harry S. (*A* 1936) Mgr. (for mail) Acme Sheet Metal Co., 2201 Broadway, and 20 W. Dartmouth Rd., Kansas City, Mo.

WHITE, John C. (*M* 1932) State Power Plant Engr. (for mail) State Bureau of Engineering, 624 E. Main St., and 622 E. Main St., Madison, Wis.

WHITE, Taylor G., Jr. (*A* 1938) Br. Mgr., U. S. Radiator Corp., 709 Ashland, Louisville, Ky.

WHITE, Thomas J. (*A* 1941; *J* 1938) Sales Engr. (for mail) American Blower Corp., 625 Market St., and 1700 N. Point St., San Francisco, Calif.

WHITE, William R. (*M* 1938; *A* 1936) Engr. (for mail) Nebraska Power Co., 718 Electric Bldg., and 4339 Larimore Ave., Omaha, Nebr.

WHITELAW, H. Leigh (*M* 1916) Jones & Laughlin Steel Corp., Third and Ross, and (for mail) Schenley Apts., Pittsburgh, Pa.

WHITELEY, Stockett M. (*M* 1933) Consulting Engr. (for mail) Baltimore Life Bldg., and 3931 Canterbury Rd., Baltimore, Md.

WHITMER, Robert P. (*M* 1935) Secy. (for mail) American Foundry & Furnace Co., and 1402 E. Washington St., Bloomington, Ill.

WHITNEY, C. W. (*M* 1935) Pres. (for mail) ABC Oil Burner & Engineering Co., Inc., 2012-14 Chestnut St., Philadelphia, and Sevilla Court, Apt. F-3, Bala-Cynwyd, Pa.

WHITT, Sidney A. (*A* 1938; *J* 1937) Engr., Air Cond. Div., Fedders Manufacturing Co., Inc., and (for mail) 12 Inwood Pl., Buffalo, N. Y.

WHITTAKER, Wayne K. (*A* 1935) Htg., Vtg. & Air Cond. Serviceman, Irving Trust Co. Bldg., 1 Wall St., New York, and (for mail) 119-23 226th St., St. Albans, L. I., N. Y.

WHITTEN, Horace E. (*M* 1924) Pres.-Treas., H. E. Whitten Co., 9 Federal Court, Boston, and (for mail) 56 Highland Rd., Somerville, Mass.

WHITTINGTON, James A. (*M* 1936) Utilization Testing Engr. (for mail) The Peoples Gas Light & Coke Co., 3921 S. Wabash Ave., Chicago, and 622 Sheridan Sq., Evanston, Ill.

WIDDOWFIELD, Arthur S. (*A* 1941; *J* 1937) Sales Engr., The Mercoid Corp., 4201 Belmont Ave., Chicago, Ill., and (for mail) 535 Hanna St., Birmingham, Mich.

WIDMER, Walter J. (*A* 1939) Secy.-Treas. (for mail) Widmer Plumbing & Heating Co., 34 N. E. Seventh Ave., and 1565 N. Shaver St., Portland, Ore.

WIEGNER, Henry B. (*M* 1919) Office Mgr., Johnson Service Co., 20 Winchester St., Boston, and (for mail) 143 Standish Rd., Watertown, Mass.

WIESNER, Blaine K. (*S* 1939) Student, Purdue University, and (for mail) 931 Sixth St., W. Lafayette, Ind.

WIGGS, G. Lorne (*M* 1936; *A* 1932; *J* 1924) (Council 1938-40) Consulting Engr. (for mail) 727 University Tower, and 4797 Grosvenor Ave., Montreal, Que., Canada.

WILCOX, Chester M. (*A* 1939) Engr., 3118 Mt. Pleasant St. N. W., Washington, D. C.

WILDE, Ray S. M. (*M* 1916) Consulting Engr., 194 Connecticut Ave., Highland Park, Mich.

WILDER, Edward L. (*M* 1915) Engr., Industrial Dept. (for mail) Rochester Gas & Electric Corp., 89 East Ave., and 369 Bonnie Brae Ave., Rochester, N. Y.

WILDER, Herbert P. (*M* 1938) Sales Engr. (for mail) Patterson Kelley Co., 101 Park Ave., New York, and Ardsley Rd., Scarsdale, N. Y.

WILDMAN, Eugene L. (*J* 1930) New York Repr., Stewart A. Jellett Co., 1328 Broadway, New York, N. Y., and (for mail) 400 Highland Terrace, Orange, N. J.

WILEY, Donald C.* (*A* 1939; *J* 1936) Engr. (for mail) John J. Nesbitt, Inc., State Rd. and Rhawn St., Philadelphia, and Kelvin Ave., Somerton. Pa.

WILHELM, Joseph E. (*J* 1936; *S* 1934) Purch. Agent, Avery Engineering Co., 1906 Euclid Ave., Cleveland, and (for mail) 294 East 195th St., Euclid, O.

WILKES, Gordon B.* (*M* 1937) Prof. of Heat Engrg. (for mail) Massachusetts Institute of Technology, Cambridge, and 51 Everett St., Newton Centre, Mass.

WILKINSON, Arthur (*A* 1936) (for mail) Wilkinson Engineering Agencies, 1253 McGill College Ave., Montreal, Que., Canada.

WILKINSON, Farley James (*M* 1933) Mgr., Central Engrg. Service, Montgomery Ward & Co., Chicago Ave. and Larabee St., Chicago, and (for mail) 18257 Martin Ave., Homewood, Ill.

WILLARD, Arthur Cutts* (*M* 1914) (*Presidential Member*) (Pres. 1928; 1st Vice-Pres. 1927; 2nd Vice-Pres. 1926; Council, 1925-1929) Pres. (for mail) University of Illinois, 355 Administration Bldg., and 711 Florida Ave., Urbana, Ill.

WILLER, Murray D. (*J* 1937; *S* 1936) 1600 Bathurst St., Toronto, Ont., Canada.

WILLEY, Earl C. (*M* 1934) Asst. Prof. of Mech. Engrg., Oregon State College, and (for mail) 121 North 29th St., Corvallis, Ore.

WILLIAMS, Allen W. (*Life Member*; *A* 1915) Secy.-Treas. (for mail) National Warm Air Register Manufacturers Institute, 5 E. Long St., Columbus, and 51 Meadow Park Ave., Bexley, O.

WILLIAMS, Chester D. (*M* 1938) Mgr. (for mail) General Air Conditioning & Heating Co., 4001 Piedmont Ave., Oakland, and 2709 College Ave., Berkeley, Calif.

WILLIAMS, Donald W. (*A* 1940; *J* 1938) Htg. Engr. (for mail) Iowa-Nebraska Light & Power Co., 1401-O St., and 2236-A St., Lincoln, Nebr.

WILLIAMS, Elwin C. (*A* 1939) Sales, Hoffman Specialty Co., 4028 Egbert Ave., Cincinnati, O.

WILLIAMS, Frank H. (*A* 1940; *J* 1934) Engr. (for mail) Frigidaire Div., General Motors Sales Corp., 4584 Waybury Grand Ave., Detroit, and 68 Amherst Rd., Pleasant Ridge, Mich.

WILLIAMS, Gordon S. (*A* 1941; *J* 1937; *S* 1936) Partner, The Augur-Williams Co., 15 Chapel St., and (for mail) P. O. Box 126, Woodmont, Conn.

WILLIAMS, H. Edmund (*J* 1939) Air Cond. Engr., John A. Connelly, Distributor, 1419 N. Broad St., Philadelphia, Pa., and (for mail) 14 West 103rd St., New York, N. Y.

WILLIAMS, J. Walter (*M* 1915) Pres. (for mail) Forest City Plumbing Co., 332 E. State St., and 923 E. State St., Ithaca, N. Y.

WILLIAMS, Lyle G. (*M* 1939) Htg. Engr., Sears-Roebuck & Co., Portland, and (for mail) Gladstone, Ore.

WILLIS, Leonard L. (*J* 1936; *S* 1935) Vice-Pres., H. Conrad Manufacturing Co., 509 First Ave. N. E., and (for mail) 5036 Lyndale S., Minneapolis, Minn.

WILLNER, Ira (*M* 1937) Pres. (for mail) Willner Heating Co., Inc., 415 Lexington Ave., and 308 East 79th St., New York, N. Y.

WILLS, Fred W. (*J* 1938) Sales Engr. (for mail) Tuttle & Bailey, Inc., 61 W. Kinzie St., and 2511 Leland Ave., Chicago, Ill.

WILMOT, Charles S. (*M* 1919) 436 Haverford Ave., Narberth, Pa.

WILSON, Alexander M. (*S* 1939) Engr., Andrew Wilson Co., 616 Essex St., and (for mail) 38 Olive Ave., Lawrence, Mass.

WILSON, Eric D. (*M* 1936) Gen. Mgr. in India (for mail) Carrier Engineering Co., Ltd., 24 Buckingham Gate, London, S. W. 1, and Ways End, St. Albans Rd., Redgate, England.

WILSON, George T. (*M* 1925) Sales Engr., Gurney Foundry Co., Ltd., 4 Junction Rd., Toronto, and (for mail) 25 Tyre Ave., Islington, Ont., Canada.

WILSON, Raymond W. (*M* 1934) Member of Firm, Wilson-Brinker Co., 412 Pythian Bldg., and (for mail) 429 Creston Ave., Kalamazoo, Mich.

WILSON, Robert A. (*M* 1936) Sales Engr., Minneapolis-Honeywell Regulator Co., 4501 Prospect Ave., Cleveland, and (for mail) Cor. Olive St. and Robens Ct., Chagrin Falls, O.

WILSON, Victor H. (*A* 1938) Contractor and Plibrico Refractory Distributor (for mail) 403 Hitchcock Bldg., Nashville, and "The Thistle-Patch," Donelson, Tenn.

WILSON, Westray E. (*A* 1939) Owner (for mail) Wilson Plumbing Co., 227 Haywood Rd., and 110 Salola St., Asheville, N. C.

WILSON, W. H. (*A* 1932) Chief Power Plant Engr., Pullman-Standard Car Manufacturing Co., 11001 Cottage Grove Ave., and (for mail) 22 West 110th Pl., Chicago, Ill.

WILTBERGER, Constant F. (*M* 1935) Partner, Pennell & Wiltberger, Cons. Engrs., Land Title Bldg., and (for mail) 2650 N. Ninth St., Philadelphia, Pa.

WINANS, Glen D. (*M* 1929) Engr. of Steam Distribution (for mail) The Detroit Edison Co., 2000 Second Ave., and 16183 Wisconsin Ave., Detroit, Mich.

WINER, Bernard B. (*S* 1940) Student (for mail) Carnegie Institute of Technology, 4921 Forbes St., and 3138 Avalon St., Pittsburgh, Pa.

WINKLER, Ralph A. (*A* 1940; *J* 1937) Sales Engr., Alfred C. Goethel Co., 2337 North 31st St., Milwaukee, and (for mail) P. O. Box 179, Elm Grove, Wis.

WINSLOW, C.-E. A.* (*M* 1932) (Council 1940) Prof. of Public Health, Yale School of Medicine and Dir., John B. Pierce Laboratory of Hygiene (for mail) 310 Cedar St., and 314 Prospect St., New Haven, Conn.

WINTERBOTTOM, Ralph F. (*M* 1923) Engr., Winterbottom Supply Co., and (for mail) 720 Moir, Waterloo, Ia.

WINTERER, Frank C. (*M* 1920) Br. Sales Mgr. (for mail) American Radiator & Standard Sanitary Corp., 300 Broadway, and 836 Juno Ave., St. Paul, Minn.

WISSING, Clement B. (*A* 1936) Secy. & Sales Mgr. (for mail) Ebner Ice & Cold Storage Co., Locust and Chestnut Sts., and 702 N. Sixth St., Vincennes, Ind.

WITHERIDGE, David E. (*J* 1936) Sales Engr., W. A. Witheridge Co., 746 S. Fourth Ave., Saginaw, Mich.

WITMER, Charles N. (*A* 1937; *J* 1930) Carrier Corp., 710 N. Harwood St., Dallas, Tex.

WITMER, Howard S. (*A* 1937) Engr. of Design (for mail) City of Bay City, City Hall, and 600 Elm St., Bay City, Mich.

WITTIG, Frederick E. (*J* 1939) Instructor, Pratt Institute, School of Science & Technology, Grand Ave., Brooklyn, and (for mail) Box 145, 19 Hillside Ave., Glenwood Landing, L. I., N. Y.

WOESE, Carl F. (*M* 1934) Consulting Engr. (for mail) Robson & Woese, Inc., 1001 Burnet Ave., and 256 Robineau Rd., Syracuse, N. Y.

WOLF, Philip (*M* 1935) Prop., City Contracting Co., 136 East 57th St., New York, N. Y.

WOLFF, Peter P. (*M* 1935) Engr., Bell & Gossett Co., 3000 Wallace St., and (for mail) 7509 Ridgeland Ave., Chicago, Ill.

WOLIN, Milton W. (*J* 1938; *S* 1937) Engr., Typhoon Air Conditioning Co., Inc., 252 West 26th St., New York, N. Y., and (for mail) R.F.D. No. 2, Box 73-D, New Brunswick, N. J.

WOLL, Willard M. (*M* 1938) Engr. (for mail) Commonwealth Edison Co., Rm. 1000, 72 W. Adams St., and 9320 S. Throop St., Chicago, Ill.

WOLLENBERGER, Louis (*M* 1938) Dist. Mgr. (for mail) Coast Counties Gas & Electric Co., 135 S. Sixth St., and 625 Sandalwood Ave., El Centro, Calif.

WONG, Wilfred S. B. (*M* 1938) (for mail) American Engineering Corp., 989 Bubbling Well Rd., and 669 Hart Rd., Shanghai, China.

WONSON, Arthur S., Jr. (*J* 1941; *S* 1938) Gauge Production Foreman, John D'Arcy, Inc., 184 Parkway, Chelsea, and (for mail) Walnut Park Ave., Essex, Mass.

WOOD, Alfred W. (*A* 1941; *J* 1938) Sales Engr., Clare Bros. & Co., Ltd., and (for mail) 451 Margaret St., Preston, Ont., Canada.

WOOD, Charles F. (*M* 1937) Air Cond. Mgr., Prod. Development & Application Dept., Frigidaire Div., General Motors Sales Corp., 300 Taylor St., and (for mail) 359 Aberdeen Ave., Dayton, O.

WOODBURY, Clyde D. (*M* 1938) Mech. Engr., Leland & Haley, Cons. Engrs., 58 Sutter St., and (for mail) 1321–37th Ave., San Francisco, Calif.

WOODGER, Herbert William (*M* 1939) Htg. & Vtg. Engr. (for mail) General Electric Co., 101 Woodlawn Ave., Pittsfield, and East St., Lenox, Mass.

WOODHOUSE, Graham D. (*A* 1938) General Supt., Dowagiac Steel Furnace Co., and 304 West St., Dowagiac, Mich.

WOODMAN, Lawrence E. (*M* 1934) Pres. (for mail) Woodman Engineering Corp., 203 E. Capitol Ave., and 925 Adams, Jefferson City, Mo.

WOODS, Baldwin M. (*M* 1937) Prof. Mech. Engrg. (for mail) University of California, and 249 The Uplands, Berkeley, Calif.

WOODS, Charles F. (*A* 1940) Sales Repr., Texas Southwestern Gas Co., and (for mail) P. O. Box 33, Bellville, Tex.

WOODS, Edward H. (*M* 1934) Engr. (for mail) Higgins & Zabriskie, 134-136 S. Aurora St., and Hook Pl., Ithaca, N. Y.

WOODWARD, Rothwell (*M* 1938) Air Cond. Sales Engr., Frigidaire Div., General Motors Sales Corp., 300 Taylor St., and (for mail) 1527 Benson Dr., Dayton, O.

WOOLCOCK, Edwin (*A* 1938) Mgr. (for mail) Woolcock Plumbing & Heating Co., 2217–15th St., and 410 Jefferson Apts., Niagara Falls, N. Y.

WOOLLARD, Mason S. (*M* 1934) Htg. Engr., H. H. Angus, Cons. Engr., 1221 Bay St., and (for mail) 31 Hillcrest Park Ave., Toronto, 5, Ont., Canada.

WOOLSTON, Alfred H. (*M* 1919) Chief Engr. (for mail) Woolston-Woods Co., 2132 Cherry St., and 4815 North 12th St., Philadelphia, Pa.

WOOTEN, M. Frank, Jr. (*M* 1941; *A* 1940) Consulting Engr. (for mail) 104 Latta Arcade, and 916 East Blvd., Charlotte, N. C.

WORMLEY, Robert F. (*A* 1938) Br. Mgr. (for mail) Grinnell Co. of Canada, Ltd., 700 Beaumont Ave., and 6092 Terrebonne Ave., Montreal, Que., Canada.

WORSHAM, Herman (*M* 1925; *J* 1918) (for mail) National Business Dept., Frigidaire Div. General Motors Sales Corp., 300 Taylor St., and 524 Daytona Pkwy., Dayton, O.

WORTHING, Stanley L. (*M* 1936) Spring Lake, Mich.

WORTHINGTON, Thomas H. (*M* 1937) Mgr., Eastern Htg. Sales, Standard Sanitary & Dominion Radiator, Ltd., and (for mail) 405 Beaubien St. W., Montreal, Que., Canada.

WORTON, William (*M* 1937) Br. Mgr. (for mail) C. A. Dunham Co., Ltd., 504 Scott Bldg., and 292 Lansdowne Ave., Winnipeg, Man., Canada.

WRIGHT, Clarence E. (*A* 1940; *J* 1935; *S* 1933) Mgr., Htg. & Vtg. Dept., Fairmont Wall Plaster Co., Tenth St., and (for mail) 908 Gaston Ave., Fairmont, W. Va.

WRIGHT, Daniel K., Jr.* (*J* 1938) Research Engr., Plymouth Cordage Co., North Plymouth, and (for mail) Obery St., Plymouth, Mass.

WRIGHT, Harris H. (*M* 1917) Prop. (for mail) H. H. Wright Co., 1322 Walnut St., and 808 Greenway Ter., Kansas City, Mo.

WRIGHT, John B. (*M* 1940) Sales Engr. (for mail) Nash Engineering Co., and 12 Fairfield Ave., South Norwalk, Conn.

WRIGHT, Kenneth A. (*M* 1921) Br. Mgr. (for mail) Johnson Service Co., 1113 Race St., Cincinnati, O., and 113 Orchard Rd., Ft. Mitchell, Ky.

WRIGHTSON, Wilbor T. (*M* 1937) Eastern Mgr. (for mail) Garden City Fan Co., 55 West 42nd St., New York, and 22 Sagamore Rd., Bronxville, N. Y.

WUNDERLICH, Milton S.* (*M* 1925) Asst. Resident Mgr., Minnesota & Ontario Paper Co., International Falls, and (for mail) 545 Mt. Curve Blvd., St. Paul, Minn.

WYATT, DeWitt H. (*M* 1936) Consulting Engr., Cooling & Htg. (for mail) Cooling & Heating Supply Co., 364 N. High St., and 226 N. Ridge Rd., Columbus, O.

WYLD, Reginald G. (*M* 1937) Vice-Pres. (for mail) Chrysler Corp., Airtemp Div., 1119 Leo St., Dayton, O.

WYLIE, Howard M. (*M* 1925; *J* 1917) Vice-Pres., in charge of Sales, The Nash Engineering Co., and (for mail) 51 Elmwood Ave., South Norwalk, Conn.

Y

YAGER, John J. (*M* 1921) Pres., Goergen-Mackwirth Co., Inc., 817 Sycamore St., and (for mail) 425 Woodbridge Ave., Buffalo, N. Y.

YAGLOU, Constantin P.* (*M* 1923) Assoc. Prof., Industrial Hygiene (for mail) Harvard School of Public Health, 55 Shattuck St., Boston, and 10 Vernon Rd., Belmont, Mass.

YATES, James E. (*M* 1934) Mgr. (for mail) Yates, Neale & Co., 231 Tenth St., and 431–16th St., Brandon, Man., Canada.

YATES, Joseph E. (*M* 1939) Asst. Engr. (for mail) Pacific Power & Light Co., 405 Public Service Bldg., and 1820 Northeast 57th Ave., Portland, Ore.

YATES, Robert A. (*J* 1939) Steamfitter (for mail) Yates Neale & Co., 231 Tenth St., and 431–16th St., Brandon, Man., Canada.

YATES, Walter (*Life Member*; *M* 1902) Governing Dir. (for mail) Matthews & Yates, Ltd., Cyclone Works, Swinton, and 4 Egerton Park, Worsley, Manchester, England.

YOUNG, Emil O. (*A* 1935) Owner (for mail) Young Regulator Co., 4500 Euclid Ave., and 2040 East 83rd St., Cleveland, O.

YOUNG, Forest H., Jr. (*A* 1936) Mgr. & Secy.-Treas., Young Heat Engineering Co., Inc., Montana Ave. and St. Johns St., Box 1583, Billings, Mont.

YOUNG, Harold J. (*M* 1937) Sales Engr., Young Radiator Co., Occidental Hotel Bldg., and (for mail) 1364 Lakeshore Dr., Muskegon, Mich.

YOUNG, J. T., Jr. (*A* 1936) Mgr. (for mail) Crane Co., Box 907, and 508 Ogden Canyon, Ogden, Utah.

YOUNG, Robert W. (*A* 1941) Dist. Sales Repr., Mueller Brass Co., and (for mail) 700 West 47th St., Kansas City, Mo.

SUMMARY OF MEMBERSHIP

Honorary Member	1	Associate Members	895
Presidential Members	20	Junior Members	357
Life Members	56	Student Members	59
Members	1644	Total	3032

UNITED STATES AND POSSESSIONS

Alabama	11
Arizona	4
Arkansas	3
California	103
Colorado	11
Connecticut	40
Delaware	12
District of Columbia	88
Florida	15
Georgia	32
Illinois	259
Indiana	37
Iowa	36
Kansas	14
Kentucky	12
Louisiana	40
Maine	4
Maryland	45
Massachusetts	96
Michigan	178
Minnesota	114
Mississippi	2
Missouri	123
Montana	4
Nebraska	20
Nevada	2
New Jersey	108
New Mexico	1
New York	417
North Carolina	37
Ohio	181
Oklahoma	19
Oregon	48
Pennsylvania	271
Rhode Island	8
South Carolina	10
South Dakota	2
Tennessee	14
Texas	89
Utah	2
Vermont	2
Virginia	35
Washington	34
West Virginia	11
Wisconsin	89
Alaska	1
Panama	1
Philippine Islands	3
	2688

DOMINION OF CANADA	215

FOREIGN COUNTRIES

Australia	11
Belgium	1
Brazil	2
Chile	1
China	10
Cuba	1
Denmark	2
Egypt	3
England	36
France	7
French North Africa	1
Germany	3
India	6
Ireland	2
Italy	7
Japan	4
Manchoukuo	1
Mexico	3
Netherlands	2
New Zealand	3
Norway	2
Roumania	1
South Africa	5
Spain	1
Straits Settlements	2
Sweden	8
Turkey	2
Venezuela	2
	129

TOTAL MEMBERSHIP	3032

LIST OF MEMBERS

(Geographically Arranged)

UNITED STATES *and* POSESSIONS

ALABAMA

Birmingham—
Cone, W. E.
Drum, L. J., Jr.
Gause, H. C.
Hardy, F. L.
Lichty, C. P.
Mabley, L. C.
Myer, H.
Richards, G. H.
Walden, H. K.

Montgomery—
Dowdy, R. B.

Tuscaloosa—
Murphree, R. L.

ARIZONA

Lowell—
Vinson, N. L.

Phoenix—
Genre, E. J.
Hummel, G. W.

Tucson—
Tidmarsh, P. M.

ARKANSAS

Little Rock—
Cumnock, H.
Kellogg, W. T.
McCoy, C. E.

CALIFORNIA

Albany—
Kaup, E. O.

Bakersfield—
Baker, H. S.

Berkeley—
Bentley, C. E.
Brokaw, G. K.
Peterson, C. L.
Raber, B. F.
Woods, B. M.

Beverly Hills—
Theobald, A.

Burlingame—
Gee, W. W.
Hill, J. A.

El Centro—
Wollenberger, L.

Fresno—
Newman, H. E.

Fullerton—
Miles, C. N.

Glendale—
Eggleston, H. L.
Wells, E. P.

Hollywood—
Billingsley, O. F., II

Los Angeles—
Anderson, C. S.
Blumenthal, M. I.
Bullock, H. H.
Dabbs, J. T.
Douglas, H. H.
Downes, A. H.
Ellingwood, E. L.
English, H.
Fabling, W. D.
Hazlehurst, H. D.
Hendrickson, H. M.
Hess, A. J.
Hill, F. M.
Hogue, W. M.
Hungerford, L.
Kennedy, M.
Kilpatrick, W. S.
Lauer, H. B.
Lowe, R. A.
McKenzie, M. C., Jr.
Medow, J.
Miller, G.
Moriarty, J. M.
Nelson, E. L.
Ness, W. H. C.
Ott, O. W.
Park, J. F.
Phillips, R. E.
Phillips, R. H.
Scofield, P. C.
Stanley, R. L.
Stewart, W. O.
Storms, R. M.
Webber, C. H.

Oakland—
Cummings, G. J.
Emanuels, M.
Harrison, G. G.
Trolese, L. G.
Wahrenbrock, O. K.
Williams, C. D.

Pacific Palisades—
Finney, B.

Palo Alto—
Hall, R. A.
Johnson, O. W.

Pasadena—
Gifford, R. L.

Piedmont—
Gayner, J.

Redwood City—
Hudson, R. A.

Riverside—
Owen, J. D.

Sacramento—
Porter, N. E.
Towle, P. H.

San Diego—
Sadler, C. B.

San Francisco—
Bouey, A. J.
Cochran, L. H.
Cockins, W. W.
Cooley, E. C.
Corrao, J.
Cushing, R. C.
Folsom, R. A.
Kolb, F. W.
Kooistra, J. F.
Krueger, J. I.
Haley, H. S.
Hickman, H. V.
Hill, E., Jr.
Holland, R. B.
Hook, F. W.
Leland, W. E.
Marshall, T. A.
Martin, J. O.
Molfino, P.
Parker, R. A.
Peterson, N. H.
Plosky, E. J.
Reed, V. C.
Rosen, E. J.
Scott, W. P., Jr.
Simons, E. W.
Simonson, G. M.
Ward, E. B.
Wayland, C. E.
Wethered, W.
White, T. J.
Woodbury, C. D.

San Gabriel—
Griffith, J. B.

San Jose—
Knudsen, W. R.

Santa Monica—
Coghlan, S. F.
Walz, C. D.

Sausalito—
Howe, W. W.

COLORADO

Colorado Springs—
Jardine, D. C.

Denver—
Adams, F. L.
Conrad, R.
Davis, A. F.
Goll, W. A.
Keithley, F. R.
McNevin, J. E.
McQuaid, D. J.
O'Rear, L. R.
Skelley, J. H.

La Junta—
Curtice, J. M.

CONNECTICUT

Bridgeport—
Earle, F. E.
Ostdahl, H. E.
Smak, J. R.

Cos Cob—
Roy, A. C.

Danbury—
Moore, M.
Orgelman, G. H.

Fairfield—
Osborn, W. J.

Greenwich—
Jones, A. L.

Hartford—
Peterson, H. P.

New Britain—
Hart, S.
Hart, T. S.

New Haven—
Converse, T. J.
Rodee, E. J.
Seeley, L. E.
Teasdale, L. A.
Winslow, C.-E. A.

New London—
Chapin, C. G.
Forsberg, W.
Hopson, W. T.

Riverside—
Murphy, J. R.

South Norwalk—
Adams, H. E.
Jennings, I. C.
Lyons, C. J.
Mead, E. A.
Wright, J. B.
Wylie, H. M.

Stamford—
Bowles, P.
Jehle, F.
Jessup, B. H.
Sanbern, E. N.
Scott, A. F. H.
Zuhlke, W. R.

Torrington—
Doster, A.
Upson, W. L.

Wallingford—
Burns, J. R.

Waterbury—
Simpson, W. K.
Stein, J.
Stewart, C. W.

West Hartford—
Hoyt, L. W.

Woodmont—
Williams, G. S.

DELAWARE

Claymont—
Hinnant, C. H., Jr.

Milford—
Downing, C. B.

Wilmington—
Burke, J. J.
Gawthrop, F. H.
Granke, A. A.
Hayman, A. E., Jr.
Kershaw, M. G.
Lownsbery, B. F.
Parvis, R. S.
Robinson, G. L.
Schoenijahn, R. P.
Steel, R. J.

DISTRICT OF COLUMBIA

Washington—
Addington, H. M.
Ay, E. L.
Baker, T.
Bennett, C. A.
Bensinger, M.
Bornstein, W.
Brown, L. S., Jr.
Brunner, E. G.
Burkhart, E. M.
Campbell, G. W.
Crawford, A. C.
Cullen, A. G.
Day, I. M.
Devore, A. B.
DeWitt, E. S.
Douglas, D. C.
Downes, H. H.
Eagleton, S. P.
Erisman, P. H., Jr.
Espenschied, F. F.
Febrey, E. J.
Feltwell, R. H.
Fife, G. D.
Fisher, J. T.
Fluckey, K. N.
Fogg, J. H.
Frankel, G. S.
Frederick, W. L.
Geiger, R. L.
Goldmann, P.
Graves, V.
Gregg, S. L.
Gritzan, L. L.
Hall, M. S.
Hanlein, J. H.
Heagerty, W. H.
Hill, W. W.
Holmes, P. B.
Holt, W. H.
Hoppe, M. F.
Hoover, W. L.
Humphrey, L. G., Jr.
Inman, C. M.
Iverson, H. R.
Karsunky, W. K.
Kiczales, M. D.
King, R. W.
Kingswell, W. E.
Koster, H. H.
Kugel, H. K.
Latterner, H., Jr.
Leser, F. A.
Ley, R. B.
Littleford, W. H.
Lloyd, E. H.
Lockhart, W. R.
Loughran, P. H., Jr.
Loving, W. H.
Malin, B. S.
McCusker, J. P.
McEntee, F. M.
Mergardt, A. P.
Miller, G. F.
Mitchell, A. E.
Nelson, H. M.
Nolan, J. J., Jr.
Norair, H.
Nordine, L. F.
Ourusoff, L.
Peacock, H.
Phillips, W. L.
Robinson, D. M.
Roper, R. F.
Sale, F. B.
Stewart, J. N.
Stock, C. S.
Stokes, A.
Stokes, A. D.
Sutter, E. E.
Thomas, G.
Thompson, N. S.
Thuney, F. M.
Tuxhorn, D. B.
Urdahl, T. H.
Walz, G. R.
Watt, R. M., Jr.
Werner, J. G.
Wilcox, C. M.

FLORIDA

Daytona Beach—
Odum, R. A.

Fort Lauderdale—
Barth, J. W.

Jacksonville—
Allen, W. W.
Beckwith, F. J.
Blackman, A. O.
Cameron, R. T.
Campbell, R. E.
Edge, A. J.
Pastor, J. C.

Miami—
Rock, G. A.

Miami Beach—
Friedman, D. H., Jr.

Orlando—
Porter, C. W.
Ward, H. H.

Tampa—
Patterson, G. P.
Thomas, B. A.

GEORGIA

Atlanta—
Baird, F. E.
Baker, C. T.
Barnes, L. L.
Boyd, S. W.
Brockinton, C. E.
Brodnax, G. H., Jr.
Clare, F. W.
Cole, C. B.
Como, J. A.
Crout, M. M.
DuChateau, M. F.
Foss, E. R.
Hahn, R. F.
Johnson, C. E.
Kent, L. F.
Klein, E. W.
Koch, A. H.
Laseter, F. L.
Lawrence, L. F., Jr.
McCain, H. K.
McKinney, W. J.
Sterner, D. S.
Stotz, R. B.
Sudderth, L., Jr.
Templin, C. L.
Tucker, T. T.

Augusta—
Akerman, J. R.
Arndt, H. W.

Brunswick—
Gilmore, J. L.

College Park—
Blackshaw, J. L.

Savannah—
Croley, J. G.
Hamlin, J. B., Jr.

ILLINOIS

Bloomington—
MaGirl, W. J.
Nesmith, O. E.
Soper, H. A.
Whitmer, R. P.

Chicago—
Adams, B. P.
Aeberly, J. J.
Ammerman, A. S., Jr.
Anoff, S. M.
Arenberg, M. K.
Aronson, H. H.
Banach, C. J.
Barnes, N. W.
Bates, J. H.
Baumgardner, C. M.
Becker, W. A.
Beery, C. E.
Bevington, C. H.
Bishop, M. W.
Black, F. C.

Blaker, A. H.
Blayney, W. R.
Borling, J. R.
Bowles, E. N.
Boyle, J. R.
Bracken, J. H.
Braun, L. T.
Brigham, C. M.
Brocha, J. F.
Brooke, I. E.
Broom, B. A.
Brown, A. P.
Buckley, M. L.
Burnam, C. M., Jr.
Casey, B. L.
Chapin, H. G.
Chase, L. R.
Christman, W. F.
Christophersen, A. E.
Clo, H. E.
Close, P. D.
Connors, E. C.
Cook, H. D.
Crone, C. E.
Crump, A. L.
Cunningham, T. M.
Dasing, E.
Dauber, O. W.
DeLand, C. W.
Ellis, G. W.
Emmert, L. D.
Ericsson, E. B.
Fatz, J. L.
Fenner, N. P.
Fleak, W. D.
Floreth, J. J.
Forbes, H. B., Jr.
Frank, J. M.
Funck, E. H.
Funk, D. S.
Gardner, W.
Gaylord, F. H.
Getschow, R. M.
Goelz, A. H.
Gossett, E. J.
Gothard, W. W.
Gotschall, H. C.
Grabman, H. B.
Graves, W. B.
Griffin, C. J.
Gritschke, E. R.
Gustafson, C. A.
Haas, S. L.
Haines, J. J.
Hale, J. F.
Hanley, T. F., Jr.
Hart, H. M.
Hattis, R. E.
Hayes, J. J.
Hendrickson, R. L.
Herlihy, J. J.
Hill, E. V.
Hines, J. C.
Howard, F. L.
Howatt, J.
Howell, L.
Hubbard, G. W.
Hustoel, A. M.
Isett, W. M.
Johns, H. B.
Johnson, C. W.
Keating, A. J.
Keeney, F. P.
Kehm, H. S.
King, A. C.
Krez, L.
Kuechenberg, W. A.
Kummer, C. J.
Lagodzinski, H. J.
LaRoi, G. H.
Lauterbach, H., Jr.
Lenone, J. M.
Leuthesser, F. W., Jr.
Lewis, S. R.
Maier, H. F.
Malone, D. G.
Malvin, R. C.
Manny, J. H.
Martin, A. B.
Matchett, J. C.
Mathis, E.
Mathis, H.
Mathis, J. W.
Mattingly, M. F.
May, A. O.
McCarthy, T. F.
McCauley, J. H.
McClellan, J. E.
McDonnell, E. N.
McDonnell, J. E.
McRae, M. W.
Merens, S. H.
Mertz, W. A.
Miller, F. A.
Miller, J. E.
Miller, R. T.
Milliken, J. H.
Mueller, H. C.
Muessig, J. W.
Mullen, T. J., Jr.
Narowetz, L. L., Jr.
Neiler, S. G.
Nelson, R. O.
Newport, C. F.
Nightingale, G. F.
Offen, B.
Olsen, C. F.
Olson, B.
Oosten, L. S.
Peck, H. E.
Peller, L.
Philippi, J. J.
Piatnitza, J., Jr.
Pitcher, L. J.
Pope, S. A.
Porter, K. C.
Powers, F. W.
Prebensen, H. J.
Prentice, O. J.
Price, C. E.
Raymond, F. I.
Reger, H. P.
Rietz, E. W.
Rottmayer, S. I.
Ruggles, R. F.
Russell, E. A.
Ryerson, H. E.
Scheidecker, D. B.
Schroeder, W. R.
Schuetz, C. C.
Schuler, W. B.
Seelig, L.
Shultz, E.
Sipp, E. F.
Solstad, L. L.
Sommerfield, S. S.
Spielmann, G. P.
Stermer, C. J.
Stevenson, M. J.
Sunderland, R. P.
Sutch, H. C.
Swanson, N. W.
Thinn, C. A.
Thomas, R. H.
Thornton, W. B.
Tobin, J. F.
Tornquist, E. L.
Tracy, W. E.
Trumbo, S. M.
Turner, G. G.
Van Alsburg, J. H.
Vernon, J. R.
vonRosenberg, P. C.
Voss, W. W.
Walters, W. T.
Ward, O. G.
Weil, M.
Weil, M. I.
Weinshank, T.
Wendt, E. H.
White, E. B.
Whittington, J. A.
Wills, F. W.
Wilson, W. H.
Wolff, P. P.
Woll, W. M.
Zack, H. J.
Zimmerman, A. H.

Des Plaines—
Lockhart, H. A.

East St. Louis—
Cover, E. B.
Lang, J. C.

Edwardsville—
Blackmore, J. J.
Mehmken, H. O.

Evanston—
Kearney, J. S.
Maccubbin, H. A.

Glencoe—
Dunham, C. A.
Hornung, J. C.

Glen Ellyn—
Parsons, L. D., Jr.
Sherman, V. L.

Hinsdale—
Gregerson, G.

Homewood—
Wilkinson, F. J.

Kewanee—
Bronson, C. E.
Charlet, L. W.
Dickson, R. B.
Hartman, J. M.

Kenilworth—
Storch, C. A.

La Grange Park—
Estep, L. G.

Moline—
Beling, E. H.
Nelson, H. W.
Nelson, R. H.

Mount Prospect—
Stacy, L. D.

Mt. Vernon—
Benoist, L. L.
Benoist, R. E.

North Chicago—
Stahl, W. A.

Oak Park—
Ash, R. S.
Fitzgerald, M. J.
May, E. M.
Pullum, C. E.
Uhlhorn, W. J.

Park Ridge—
Heckel, E. P.
Locke, J. S.
Moore, R. E.
Sutcliffe, A. G.

Peoria—
Hauer, F.
Meyer, F. L.

River Forest—
Bichowsky, F. R.

Rochelle—
Caron, H.

Rockford—
Berzelius, C. E.
Braatz, C. J.
Dewey, R. P.
Stewart, D. J.

Rock Island—
Kimble, C. W.

Springfield—
Sharp, J. E.

Urbana—
Broderick, E. L.
Engdahl, R. B.
Fahnestock, M. K.
Hershey, A. E.
Konzo, S.
Kratz, A. P.
Markland, C. E.
Sachs, S.
Severns, W. H.
Willard, A. C.

Villa Park—
Armspach, O. W.

Waukegan—
Cary, E. B.

Western Springs—
Morgan, R. W.

Wilmette—
DeBerard, P. E.
Marschall, P. J.

Winnetka—
Killian, V. J.
Mittendorff, E. M.

INDIANA

East Chicago—
Boyar, S. L.

Evansville—
Becker, R. K.
Grossman, F. A.
Koenig, A. C.

Fairland—
Garber, W. E., Jr.

Fort Wayne—
Abramson, R. J.

Gary—
Kirtland, E. M.

Goshen—
Shaw, B. E.

Huntington—
Redrup, W. D.
Smith, G. W.

Indianapolis—
Ammerman, C. R.
Clark, L. W.
Cross, R. C.
Fenstermaker, S. E.
Hagedon, C. H.
Hayes, J. G.
Hildreth, E. S.
Niesse, J. H.
Paetz, G. A.
Poehner, R. E.
Supple, G. B.

Lafayette—
Clemens, J. D.

La Porte—
Meyer, K. A.
Shrock, J. H.

Lawrenceburg—
Bechtol, J. J.

Michigan City—
Stockwell, W. R.
Westphal, N. E.

Muncie—
Pfriem, P. G.
Price, C. F.

Peru—
Thrush, H. A.

South Bend—
Rossiter, I. J.

Vincennes—
Wissing, C. B.

Wabash—
Shivers, P. F.

West Lafayette—
Miller, W. T.
Ruppert, C. F.
Wiesner, B. K.
Zaki, H. M.

IOWA

Ackley—
Nelson, G. O.

Amana—
Foerstner, G. C.
Zuber, O. C.

Ames—
Norman, R. A.
Sandfort, J. F.
Stiles, G. S.

Des Moines—
Bartels, E. M.
Borg, E. H.
Campbell, B.
Delavan, N. B.
Drain, H. E.
Duitch, P. R.
Fergestad, M. L.
Friedline, J. M.
Helstrom, C. W.
Hennessy, W. J.
Johnson, T. R.
Johnston, M. T.
Klein, F. R.
Landes, B. E.
LaRue, P.
Murphy, D. C.
Schnell, R. H.
Stuart, W. W.
Tone, J. E., Jr.
Triggs, F. E.
Vaughn, F. R.
Walters, A. L.

Sioux City—
Hagan, W. V.
Raven, A. H.

Waterloo—
Hedeen, L. E.
Herbert, R. M.
Knox, J. C.
Mitchell, J. A.
Todd, M. L.
Winterbottom, R. F.

KANSAS

Coffeyville—
Cheeseman, E. W.

Great Bend—
Morrison, W. L.

Hutchinson—
Stevens, H. L.

Junction City—
Froelich, H. A.

Kansas City—
Allen, D. M.
Angus, F. M.
Jolley, E. M.
Kelly, H. J.
Kerr, G. C.
May, G. E.
Moses, W. B., Jr.
Nelson, L. K.
Oesterle, A. L.
Oster, W. P.
Parkerson, W.
Purinton, I. H.
Salzer, A. R., Jr.
Shepperd, P. D.
Stone, F. M.
Story, J. W.
Weil, L. S.

Lawrence—
Machin, D. W.
Robb, J. E.
Sluss, A. H.

Leavenworth—
Foley, D. F.

Salina—
Bachofer, H. A., Jr.
Ryan, W. F.

Wichita—
Buckley, D. J.

KENTUCKY

Lexington—
May, J. W.
O'Bannon, L. S.
West, P.

Louisville—
Donelson, W. N.
Groot, H. W.
Hellstrom, J.
Hubbuch, N. J., Jr.
Murphy, H. C.
Nutting, A.
White, T. G., Jr.

South Fort Mitchell—
Kennedy, O. A.

Vine Grove—
Cropper, R. O.

LOUISIANA

Bogalusa—
Gray, W. C.

Gretna—
Hero, G. A., Jr.

New Orleans—
Adair, J. S.
Burns, F. G.
Crary, J. O.
Cressy, L. V.
DeLaureal, W. D.
Devlin, J.
Dudley, W. H., Jr.
Dunlap, A. L.
Elizardi, R.
Fischer, F. P.
Friedler, J. J., Jr.
Gamble, C. B.
Gammill, O. E., Jr.
Graham, F. D.
Grant, W. H., Jr.
Guest, R. B.
Gutknecht, F.
Healy, C. T.
Helwick, N. J.
Holzer, R. J., Jr.
Jordy, J. J.
Kelley, F. J.

Shreveport—
Fitz Gerald, W. E.
Zwally, A. L.

MAINE

Bangor—
Prince, R. F.

Lewiston—
Fowles, H. H.

Portland—
Fels, A. B.
Merrill, C. J.

MARYLAND

Baltimore—
Collier, W. I.
Crosby, E. L.
Dressell, R. E.
Gjertsen, G.
Hunt, M.
Leilich, R. L.
Levitt, L. L.
McCormack, D.
McCrea, L. W.
Nest, R. E.
North, W. R.
Posey, J.
Seiter, J. E.
Shepard, J. deB.
Sklarevski, R.
Smoot, T. H.
Taze, E. H.
Vance, L. G.
Vincent, P. J.
Whiteley, S. M.

Bethesda—
Brandt, A. D.
Goodwin, E. W.
Schlichter, C. F.
Smith, S.
Terhune, R. D.

Chevy Chase—
Beitzell, A. E.
Thulman, R. K.

Cumberland—
Griffith, C. A.

Elkton—
McCloskey, J. H.

Glen Burnie—
Rodgers, J. S.

Hyattsville—
Baldwin, K. F., Jr.

Kensington—
Markert, J. W.
Spurney, F. E.

Laurel—
Kluckhuhn, F. H.

Rockville—
Brunett, A. L.

Roland Park—
Dorsey, F. C.

Silver Spring—
Croney, P. A.
Dill, R. S.
Fineran, E. V.
Hannigan, W.
Kajuk, A. E.
Shuman, L.
Stack, A. E.

Takoma Park—
Borkat, P.

Yorktowne Village—
Burns, H. J.

MASSACHUSETTS

Arlington—
Shaw, N. J. H.

Arlington Heights—
Tarr, H. M.

Auburndale—
Ahearn, W. J.

Belmont—
Kellogg, A. S.
Spence, R. A.

Boston—
Archer, D. M.
Baker, R. H.
Bartlett, A. C.
Blakeley, H. J.
Brinton, J. W.
Brissette, L. A.
Bryant, A. G.
Colby, J. H.
Cummings, C. H.
Donohoe, J. B.
Drinker, P.
Edwards, D. J.
Foulds, P. A. L.
Franklin, R. S.
Gleason, G. H.
Hashagen, J. B.
Jennings, W. G.
Kelley, J. J.
Kimball, C. W.
Lair, P. H.
Licandro, J. P.
Lincoln, R. L.
Mayette, C. E.
McCoy, T. F.
Merrill, F. A.
Nee, R. M.
Pohle, K. F.
Stetson, L. R.
Swaney, C. R.
Tuttle, J. F.
Waterman, J. H.
Yaglou, C. P.

Braintree—
Chenoweth, D. M.

Bridgewater—
Beaulieu, A. A.

Cambridge—
Flint, C. T.
Holt, J.
Moore, H. C.
Peterson, C. M. F.
Saurwein, G. K.
Staszesky, F. M.
Touloukian, Y. S.
Wilkes, G. B.

Chelmsford—
Ridley, W. H.

Chelsea—
Hochman, E.

Dorchester—
Goodrich, C. F.
Hosterman, C. O.
Shaer, I. E.

East Lynn—
Lauckner, C. G., III

Essex—
Wonson, A. S., Jr.

Fitchburg—
Illig, E. E.
Illig, W. R.
Karlson, A. F.
McKittrick, P. A.

Harwich Port—
Maxwell, G. W.

Hyde Park—
Ellis, F. R.

Lawrence—
Bride, W. T.
Wilson, A. M.

Leominster—
Kern, R. T.

Lynn—
Farrow, H. L.
Feehan, J. B.

Malden—
Denham, H. S.
Richards, L. V.

Melrose—
Gerrish, G. B.

Newton Centre—
Murray, J. J.

Newtonville—
Emerson, R. R.
Jones, W. T.

Pittsfield—
Nicholls, J. M.
Wagner, E. A.
Woodger, H. W.

Plymouth—
Wright, D. K., Jr.

Reading—
Ingalls, F. D. B.

Revere—
Brayman, A. I.

Roslindale—
Larson, C. W.

Saugus—
Blair, D. W.

Shirley—
Boyden, D. S.

Somerville—
Scalingi, C. R.
Whitten, H. E.

South Hamilton—
Mandell, T. P.

Springfield—
Cooper, W. B.
Cross, R. E.
Holmes, R. E.
Huggins, L. G.
Leland, W. B.
Murphy, W. W.

Watertown—
Wiegner, H. B.

Wellesley Hills—
Barnes, W. E.

West Roxbury—
McPherson, W. A.

Weymouth—
McCafferty, J. E.

Winchester—
Carrier, E. G.

Worcester—
Kolb, R. P.
Wechsberg, O.

MICHIGAN

Albion—
Gable, H. R.

Ann Arbor—
Backus, T. H. L.
Kessler, C. F.
Marin, A.

Battle Creek—
Christenson, H.
Dempsey, S. J.
Dyer, W. S.

Bay City—
Gray, J. W.
Henry, E. C.
Witmer, H. S.

Birmingham—
Akers, G. W.
Hadjisky, J. N.
Hyde, E. F.
Root, E. B.
Widdowfield, A. S.

Detroit—
Adam, R. W.
Anderson, E. J.
Baldwin, W. H.
Barth, H. E.
Barton, J.
Bassett, J. W.
Bay, C. H.
Beattie, J.
Berryman, R. H.
Biggers, R. H.
Bishop, F. R.
Blackmore, F. H.
Boales, W. G.
Bottum, E. W.
Busse, H.
Champlin, R. C.
Clark, E. H.
Connell, R. F.
Coon, T. E.
Cummins, G. H.
Darlington, A. P.
Dauch, E. O.
Deppmann, R. L.
Dickenson, F. R.
Dubry, E. E.
Elliott, N. B.
Falk, D. S.
Feinberg, E.
Gair, K. B.
Giguere, G. H.
Goss, M. H.
Harrigan, E. M.
Harrigan, E. R.
Hesselschwerdt, A. L.

Heydon, C. G.
Hogan, E. L.
Hubbard, N. B.
Hughson, H. H.
Hutzel, H. F.
Johnson, F. W.
Kasier, F.
Kaufman, H. J.
Kilner, J. S.
Kirkpatrick, A. H.
Knapp, A. E.
Knibb, A. E.
Lewis, K. C.
Linebaugh, J. E.
Linsenmeyer, F. J.
Livermore, J. N.
Luty, D. J.
Mabley, T. H.
MacMillan, A. R.
Mally, C. F.
Marzolf, F. X.
McCrea, J. B.
McGeorge, R. H.
McIntire, J. F.
McLean, D.
Metcalfe, C.
Middleton, D. K.
Milward, R. K.
Morse, C. T.
Morse, L. S., Jr.
Nutting, H. G. D.
Oberschulte, R. H.
Old, W. H.
O'Gorman, J. S., Jr.
O'Rourke, H. D., Jr.
Paetz, H. E.
Parrott, L. G.
Partlan, R. L.
Pavey, C. A.
Purcell, F. C.
Randall, R. D.
Randall, W. C.
Reader, J. T.
Sanford, S. S.
Schechter, J. P.
Schmidt, K., Jr.
Shea, M. B.
Sheley, E. D.
Shields, C. D.
Smith, W. O.
Snyder, J. W.
Soeters, M.
Spitzley, R. L.
Spurgeon, J. H.
Stites, R.
Strand, C. A.
Taylor, H. J.
Tillinghast, H. S.
Toonder, C. L.
Tuttle, G. H.
Van Nouhuys, H. C.
Waid, G. H.
Walker, J. H.
Weinert, F. C.
Whelan, W. J.
White, E. S.
Williams, F. H.
Winans, G. D.

Dowagiac—
Cunningham, J. S.
Deming, R. E.
Harden, J. C.
Snyder, E. F., Jr.
Torr, T. W.
Woodhouse, G. D.

East Lansing—
Ely, R. S.
Kelly, O. A.
Merz, R. A.
Miller, L. G.
Pesterfield, C. H.

Flint—
Hendriksen, L.

Grand Rapids—
Boot, A.
Bradfield, W. W.
Bratt, H. D.
Hall, T.
Marshall, O. D.
Morton, C. H.
Stafford, T. D.
Thoman, E. O.
Todd, S. W., Jr.
Warren, F. C.
Ziesse, K. L.

Grosse Pointe Park—
Buckeridge, V. L.
Davis, G. L., Jr.
Feely, F. J.
McConachie, L. L.

Hermansville—
Voorhees, G. A.

Highland Park—
Harrower, W. C.
Wilde, R. S. M.

Hillsdale—
Oberlin, J. A.

Holland—
DeRoo, W. C.

Houghton—
Seeber, R. R.

Kalamazoo—
Brinker, H. A.
Brundage, F. W.
Downs, S. H.
McConner, C. R.
Metzger, H. J.
Schlichting, W. G.
Temple, W. J.
Wilson, R. W.

Lansing—
Distel, R. E.
Hill, V. H.
McLouth, B. F.
Parsons, R. A.

Lawrence—
Morton, P. S.

Marquette—
Bernhard, G.

Mt. Clemens—
Bailey, E. P.

Muskegon—
Young, H. J.

Muskegon Heights—
Reid, H. F.

Okemos—
Foote, J. H., Jr.

Pleasant Ridge—
Burch, L. A.

Plymouth—
Lee, J. A.

Pontiac—
Trzos, O. A.

Rochester—
Clar, R., Jr.

Royal Oak—
Helmrich, G. B.
Keyser, H. M.

Saginaw—
Witheridge, D. E.

Spring Lake—
Worthing, S. L.

Three Rivers—
Hall, C. H.

Wayland—
Snook, A. H.

MINNESOTA

Bayport—
Swanson, E. C.

Cloquet—
Pauley, R. D.

Duluth—
Foster, C.
Prosser, R. G.

Hibbing—
Bispala, J. T.

Mankato—
Forderbruggen, K. J.

Mendota—
Estes, E. C.

Minneapolis—
Algren, A. B.
Bell, E. F.
Bensen, C. L.
Benson, M. L.
Betts, H. M.
Bjerken, M. H.
Bredesen, B. P.
Burns, E. J.
Burritt, C. G.
Caple, I.
Carlson, C. O.
Chalmers, C. H.
Copperud, E. R.
Cottrell, W. H.
Craig, J. A.
Cumming, F. J.
Dahlstrom, G. A.
Davidson, J. C.
Dever, H. F.
Eklund, G. P.
Fedders, M. P.
Forfar, D. M.
Gausewitz, W. H.
Gerrish, H. E.
Gordon, E. B., Jr.
Gorgen, R. E.
Gross, L. C.
Haines, J. E.
Haley, R. T.
Hanson, L. C.
Hanson, L. P.
Harris, J. B.
Helstrom, H. G.
Hitchcock, P. C.
Huch, A. J.
Jordan, R. C.
Killeen, E. F.
Knapp, D. S.
Knowles, E. L.
Lange, F. F.
Larson, C. P.
Lawrence, C. T.
Legler, F. W.
Lilja, O. L.
Lund, C. E.
Lunn, R. J.
Marshall, S. C.
McDonald, T.
Mills, H. C.
Morgan, G. C.
Morton, H. S.
Newton, A. B.
Nicols, J. A.
Orr, G. M.
Petersen, B. W.
Petersen, C. P.
Petersen, R. H.
Priester, G. B.
Proebstle, L.
Reisberg, L. K.
Resch, R. J.
Roberts, H. P.
Rowley, F B.
Schad, C. A.
Schultz, A. W.
Seelert, E. H.
Shipley, S. C.
Stafford, J. F.
Stiller, F. W.
Sutherland, D. L.
Swanson, D. F.
Swenson, J. E.
Uhl, W. F.
Uhl, E. J.
Wallace, H. P., Jr.
Willis, L. L.

Nicollet Island—
Schernbeck, F. H.

Owatonna—
Anderson, G. A. M.

Robbinsdale—
Hyde, L. L.

Rochester—
Adams, N. D.

St. Paul—
Anderson, D. B.
Backstrom, R. E.
Bauer, A. E.
Bean, G. S.
Colman, R. C.
Diamond, D. D.
Evans, R. W.
Gausman, C. E.
Hickey, D. W.
Hyde, E. V.
Jones, E. F.
Lynn, R. G.
McNamara, W.
Mitchell, J. G.
Oberg, H. C.
Paine, H. A.
Persson, N. B.
Peterson, J. R.
Ruff, D. C.
Sanford, A. L.
Schlick, P. F.
Sturm, W.
Tilton, N. K.
Winterer, F. C.
Wunderlich, M. S.

Spring Park—
Streater, E. C.

Wayzata—
Hughes, S.

MISSISSIPPI

Jackson—
Light, J. C.
Ludlow, H. M.

MISSOURI

Clayton—
DuBois, L. J.

Des Peres—
Lautz, F. A.

Hickman Mills—
Case, D. V.

Independence—
Cook, B. F.

Jefferson City—
Woodman, L. E.

Joplin—
Jones, J. T.
McMullen, E. W.
Satterlee, H. A.

Kansas City—
Arthur, J. M., Jr.
Ball, W.
Banner, F. L. D.
Barnes, A. R.
Betz, H. D.
Caleb, D.
Campbell, E. K.
Campbell, E. K., Jr.
Campbell, R. P.
Cassell, W. L.
Clegg, C.
Dawson, T. L.
Dean, F. J., Jr.
Dean, M. H.
Dodds, F. F.
Downes, N. W.
Farber, L. M.
Fehlig, J. B.
Fehlig, J. B., Jr.
Flarsheim, C. A.
Forslund, O. A.
Gillham, W. E.
Harbordt, O. E.
Holuba, H. J.
Kitchen, J. H.
Leffel, P. C.
Mahon, C. A.
Manning, C. E.
Marchio, E., Jr.
Marston, A. D.
Matthews, J. E.
Millis, L. W.
Nottberg, G.
Nottberg, H.
Nottberg, H., Jr.
O'Dower, H. J.
Pellmounter, T.
Pexton, F. S.
Rivard, M. M.
Russell, W. A.
Ryan, J. B.
Selig, E. T., Jr.
Sheppard, F. A.
Stephenson, L. A.
Stevens, K. M.
Weiss, C. A.
White, H. S.
Wright, H. H.
Young, R. W.
Zink, D. D.

Kirkwood—
Hartwein, C. E.
Schwartz, N. E.

Maplewood—
Curry, R. F.

Normandy—
Dulle, W. L.

Overland—
Cain, W. J.
Sydow, L. J.

Richmond Heights—
Siegel, D. E.
Spelbrink, R. G.

St. Joseph—
Harton, A. J.
Zurow, W.

St. Louis—
Ahrens, C. F.
Bayse, H. V.
Boester, C. F.
Carlson, E. E.
Carter, J. H.
Clarkson, J. R.
Cooper, J. W.
Corrigan, J. A.
Davis, C. R.
Dreher, L. F.
Driemeyer, R. C.
Evans, B. L.
Fagin, D. J.
Falvey, J. D.
Foster, J. M.
Gilmore, L. A.
Grossenbacher, H. E.
Grossmann, H. A.
Haller, A. L.
Hamig, L. L.
Hester, T. J.
Horch, G. E.
Kella, W. B.
Kuntz, E. C.
Laskaris, N. G.
Laufketter, F. C.
Malone, J. S.
McLarney, H. W.
McMahon, T. W.
Moon, L. W.
Nelson, C. L.
Norris, W. P.
Oonk, W. J.
Pellegrini, L. C.
Rodenheiser, G. B.
Rosebrough, J. S.
Sampson, W. D.
Sanders, C. M., Jr.
Scherrer, L. B.
Simons, B. C.
Sodemann, P.
Sodemann, W. C. B.
Stammer, E. L.
Szombathy, L. R.
Tenkonohy, R. J.
Toensfeldt, R.
Weber, E. F., Jr.
Weiss, W. G.

University City—
Bradley, E. P.
Carlock, M. F.
Sharp, H. C.
Smith, R. C.

Webster Groves—
Myers, G. W. F.
Ronsick, E. H.
Rosebrough, R. M.

MONTANA

Billings—
Cchagen, C. C.
Young, F. H., Jr.

Butte—
Sullivan, T. J.

Great Falls—
Ginn, T. M.

NEBRASKA

Hastings—
Swingle, W. T.

Lincoln—
Green, E. W.
Hellmers, C. C., Jr.
Lehman, M. G.
Ress, O. J.
Stanton, H. W.
Williams, D. W.

Omaha—
Eaton, B. K.
Kleinkauf, H.
Knepper, H. H.
Mathis, J.
Merwin, G. E.
Olson, M. J.
Organ, F.
Peiser, M. B.
Reifschneider, J.
White, W. R.

Schuyler—
Matousek, A. G.

Scottsbluff—
Davis, O. E.
Prawl, F. E.

NEVADA

Reno—
Mast, C. M.
Thompson, C.

NEW JERSEY

Asbury Park—
Strevell, R. P.

Atlantic City—
Strouse, S. B.

Bayonne—
Schwartz, J.

Belleville—
Thornton, T. L.

Blackwood—
Weid, H. L.

Bloomfield—
Austin, W. H.
Berry, R. U.
Burges, J. H. M.
Faust, F. H.
Harrington, E.
Jackes, H. D.
Kelsey, H. D.
McLenegan, D. W.
Tenney, D.

Bogota—
Griess, P. G.

Caldwell—
Albright, C. B.

Camden—
Brown, W. M.
Coward, C. W.
Kappel, G. W. A.
Plum, L. H.
Webster, E. K.
Webster, W., Jr.

Chatham—
Maddux, O. L.

Cliffside Park—
Butler, P. D.

Clifton—
Hilder, F. L.

Closter—
Steinke, B. J.

Collingswood—
Mohrfeld, H. H.

East Orange—
Ekings, R. M., Jr.
Ferguson, R. R.
Gombers, H. B.
Reilly, J. H.
Schroth, A. H.
Stacey, A. E., Jr.
Tallmadge, W.
Turno, W. G.

Elizabeth—
Anderson, J. W.
Cornwall, G. I.
Wheller, H. S.

Englewood—
Hendrickson, W. B.

Essex Fells—
Soule, L. C.

Fort Dix—
Sharp, J. R.

Freehold—
Buck, D. T.

Glen Ridge—
Stark, C. E.

Haddonfield—
Buzzard, F. H.

Hasbrouck Heights—
Goodwin, S. L.

Hoboken—
Dick, H.

Irvington—
Feldermann, W.
Reinke, A. G.
Stengel, F. J.

Jersey City—
Borak, E.
Jones, H. L.
Kelly, C. J.
Kunzog, T. W.
Ritchie, W.
Waltherthum, J. J.

Kearny—
Shaffer, C. E.

Leonia—
Close, R.

Lyndhurst—
Ehrlich, M. W.

Manville—
Parkinson, J. S.

Merchantville—
Binder, C. G.

Montclair—
Bentz, H.
Broome, J. H.

Mount Holly—
King, J. S.

Newark—
Bryant, P. J.
Carey, P. C.
Haitmanek, L. M.
Kruse, W. C., Jr.
Kucinski, W. V.
Leinroth, J. P.
Morehouse, H. P.
Ray, L. B.
Raymer, W. F., Jr.
Steinmetz, C. W. A.
Zemelman, I. M.

New Brunswick—
Wolin, M. W.

North Arlington—
Bermal, A. H.

North Plainfield—
Seal, A. T.

Nutley—
Morris, C. R.

Orange—
Crawford, J. H., Jr.
Settelmeyer, J. T.
Wildman, E. L.

Passaic—
Blazer, B. V.

Paterson—
Cox, H. F.
Frank, O. E.
McBride, J. N.

Perth Amboy—
Simkin, M.

Plainfield—
Pond, W. H.

Red Bank—
Isaacs, H. A., Jr.
Miner, H. H.

Ridgefield Park—
Davis, A. C.

Ridgewood—
Fitts, J. C.
Miller, A. T.

Riverside—
Graham, J. J.

Roselle Park—
Tiernan, J. P.

South Orange—
Browne, A. L.

Summit—
Oaks, O. O.

Teaneck—
Heebner, W. M.

Trenton—
Bieringer, F. A.

Union—
Lyman, S. E.

Union City—
Matheka, C. R.
Taverna, F. F.

Verona—
Boxall, F.

Westfield—
Hach, E. C.
Hansler, J. E.

Westmont—
Robinson, E. R.

West Orange—
Adlam, T. N.
Snavely, E. R.

Woodbury—
Schmieler, J. B.

NEW MEXICO

Albuquerque—
McKinley, C. B.

NEW YORK

Albany—
Bond, H. A.
Dick, A. V.
Lewis, H. F.
Murray, T. F.
Mytinger, K. L.
Taggart, R. C.
Taylor, R. B.
Teeling, G. A.
Westover, W.

Bronxville—
Bishop, C. R.
Dornheim, G. A.
Groves, S. A.

Brooklyn—
Balsam, C. P.
Bigolet, L.
Birkett, H. S.
Brex, I. E.
Derer, B.
DeSomma, A. E.
Dutcher, H. S.
Dwyer, T. F.
Fidelius, W. R.
Goldberg, M.
Gornston, M. H.
Hollister, N. A.
Jacobi, B. A.
Janet, H. L.
Josephson, S.
Levine, C.
Levine, L. J.
Mollander, E. D.
Phillips, F. W.
Potter, J. R.
Ritchie, E. J.
Rodman, R. W.
Rose, J. C.
Thornburg, H. A.
Tusch, W.
Vervoort, E. L.
Wachs, L. J.
Weissblatt, N.

Buffalo—
Adema, G. E.
Beman, M. C.
Berringer, S. H.
Booth, C. A.
Cherry, L. A.
Cheyney, C. C.
Constant, E. S.
Davis, J.
Day, H. C.
Drake, G. M.
Farnham, R.
Farrar, C. W.
Harding, L. A.
Hart, F. D.
Hawk, J. K.
Heath, W. R.
Hedley, P. S.
Hirschman, W. F.
Hurwich, S. B.
Jackson, M. S.
Kamman, A. R.
Landers, J. J.
Lenihan, W. O.
Lighthart, C. H.
Long, H. P.
Love, C. H.
Madison, R. D.
Mahoney, D. J.
Moesel, F. A.
Mosher, C. H.
Osborne, S. R.
Peterson, D. J.
Pleuthner, R. L.
Quackenbush, S. M.
Reif, A. F.
Reif, C. A.
Roebuck, W., Jr.
Schafer, H. C.
Schmidt, H.
Seelbach, H., Jr.
Seyfang, W. G.
Siegel, R. C.
Spencer, W. E.
Voisinet, W. E.
Walker, E. R.
Wendt, E. F.
Whitt, S. A.
Yager, J. J.
Zolitsch, H. G.

Waechter, H. P.
Walton, C. W., Jr.
Waring, J. M. S.
Wasson, R. A.
Webster, C. C.
Wheeler, J., Jr.
Wilder, H. P.
Williams, H. E.
Willner, I.
Wolf, P.
Wrightson, W. T.

Niagara Falls—
Woolcock, E.

North Tarrytown—
Weiss, A. P.

North Tonawanda—
Conaty, B. M.

Oswego—
Lyke, H. W.

Pelham Manor—
White, E. G.

Potsdam—
Armstrong, E. T.

Poughkeepsie—
Bertolette, C., Jr.

Rochester—
Baker, H. L., Jr.
Betlem, H. T.
Cook, R. P.
Dickason, G. D.
Hakes, L. M.
Hutchins, W. H.
Kimmell, P. M.
King, R. L.
Leonhard, L. W.
Lewis, C. E.
Stacy, S. C.
Treadway, J. Q.
Vidale, R.
Wilder, E. L.

Rome—
Lynch, W. L.

Scarsdale—
Cumming, R. W.

Schenectady—
Hunziker, C. E.
Spitzley, J. H.
Welch, L. A., Jr.

Snyder—
Gifford, C. A.

Staten Island—
Brown, H. J. (Clifton)
Callahan, P. J. (Great Kills)
Eschenbach, S. P. (Fort Totten)
Frimet, M. (West Brighton)
Johnson, E. B. (Port Richmond)
Pfuhler, J. L. (W. New Brighton)
Vivarttas, E. A. (W. New Brighton)
Volkhardt, A. N. (Rosebank)

Suffern—
Barnum, M. C.

Syracuse—
Acheson, A. R.
Ashley, C. M.
Avery, L.
Carrier, W. H.
Cherne, R. E.
Cox, T. M., Jr.
Day, V. S.
DesReis, J. F.
Driscoll, W. H.
Dunne, R. VanD.
Evans, E. C.
French, D.
Graham, W. D.
Henszey, W. P.
Hockensmith, F. E.
Ingels, M.
Kubasta, R. W.
Lewis, L. L.
Lyle, J. I.
Mohsin, S.
Murphy, E. T.
Noble, M.
Schoeffter, H. M.
Sheldon, N. E.
Silvera, A.
Taliaferro, R. R.
Traynor, H. S.
Woese, C. F.

Tonawanda—
Karlsteen, G. H.

Tuckahoe—
Goulding, W.

Utica—
Randolph, H. F.
Steinhorst, T. F.

Valhalla—
Mehne, C. A.

Watervliet—
Schubert, A. G.

White Plains—
Belsky, G. A.
Zibold, C. E.

Yonkers—
Flink, C. H.
Goerg, B.
Harmonay, W. L.
Ormiston, J. B.
Rainger, W. F.
Werker, H.

NORTH CAROLINA

Asheville—
Wilson, W. E.

Charlotte—
Baber, J. E.
Bailey, J. L.
Barnes, H. S.
Brandt, E. H., Jr.
Hill, H. H.
Hodge, W. B.
Muirheid, J. G.
Petty, C. E.
Warren, R. M., Jr.
Wooten, M. F., Jr.

Durham—
Altemueller, G. F.
Cooke, T. C.
Daddario, F. T.
Reed, F. J.
Skinner, A., Jr.
Theiss, E. S.
Wallace, W. M., II

Fort Bragg—
Neal, J. P.

Greensboro—
Adams, C. Z.
Burr, G. C.
Guest, P. L., Jr.
Harding, E. R.
Hoffman, H.
Klages, F. E. P.
Oertel, F. H. E.

High Point—
Gray, H. E.
Gray, W. E.
Lewis, W. W.

Raleigh—
Lee, R. T.
Rice, R. B.
Vaughan, L. L.

Winston-Salem—
Bahnson, F. F.
Brown, M. D.
Cornwall, C. C.
Marshall, J.
Page, A.

OHIO

Amsterdam—
Allensworth, J. E.

Ashland—
Rybolt, A. L.

Chagrin Falls—
Guilbert, S. R.
Wilson, R. A.

Cincinnati—
Bird, C.
Black, J. M.
Blum, R. J., Jr.
Boyd, T. D.
Buenger, A.
Coombe, J.
Cott, W. B.
Crombie, J.
Edwards, A. W.
Ellis, G. P.
Gannon, R. R.
Green, W. C.
Hard, A. L.
Helburn, I. B.
Houlis, L. D.
Houliston, G. B.
Hudepohl, L. F.
Hust, C. E.
Hutchinson, B. L., Jr.
Juergens, W. A.
Junker, W. H.
Kiefer, C. J.
Killian, W. J.
Kinney, A. M.
Kramig, R. E., Jr.
Leupold, G. L.
Lewis, H. E.
Mathewson, M. E.
McNamee, E. W.
Moore, H. W.
Motz, O. W.
Mursinna, G. P.
Pillen, H. A.
Pistler, W. C.
Richard, E. J.
Royer, E. B.
Ruff, A. G.
Sigmund, R. W.
Silberstein, B. G.
Sproull, H. E.
Stevens, W. R.
Sutfin, G. V.
Thompson, E. B.
Ward, F. J.
Washington, L. W.
Williams, E. C.
Wright, K. A.

Cleveland—
Auer, G. G.
Avery, L. T.
Baggaley, W.
Beach, W. R.
Berger, J. L.
Brown, T.
Cohen, P.
Conner, R. M.
Cummings, R. J.
Curtis, H. F.
Cushing, C. F.
Downe, E. R.
Evans, W. A.
Eveleth, C. F.
Friedman, A.
Gayman, P. D.
Geltz, R. W.
Gottwald, C.
Gray, E. W.
Hall, N. H.
Harvey, L. C.
Heisterkamp, H. W.
Jones, J. P.
Kaercher, C. M. H.
Kitchen, F. A.
Klie, W.
Levy, M. I.
Machen, J. T.
McKeeman, C. A.
Moore, W. R.
Nachman, G. P.
Nessell, C. W.
Nobis, H. M.
Platz, J. F.

Pogalles, L. H.
Powers, L. G.
Quick, B. A.
Repp, H. L.
Rhoton, W. R.
Rowe, W. M.
Slawson, L. E.
Smith, W. D.
Southmayd, R. T.
Taze, D. L.
Tuve, G. L.
Vanderhoof, A. L.
Webb, E. C.
Weldy, L. O.
Wetzell, H. E.
Young, E. O.

Cleveland Heights—
Cady, E. F.
Clark, R. L.
Davis, R. G.

Columbus—
Breneman, R. B.
Brown, A. I.
Conklin, R. M.
Gowdy, A. C.
Leiby, R. S.
Myler, W. M., Jr.
Noble, J. P.
Oelgoetz, J. F.
Ralph, D. S.
Sherman, R. A.
Slemmons, J. D.
Williams, A. W.
Wyatt, DeW. H.

Cortland—
Swenehart, D. W.

Cuyahoga Falls—
Humphrey, D. E.
Ray, G. E.

Dayton—
Baker, I. C.
Brown, J. S.
Chapman, W. A., Jr.
Everetts, J., Jr.
Fuller, E. W.
Gibbons, M. J.
Godfrey, J. E.
Gonzalez, R. A.
Hull, H. B.
Kincaide, M. C.
Kucher, A. A.
Kuempel, L. L.
LaSalvia, J. J.
Lindsay, G. W., Jr.
Livar, A. P.
Smith, N. J.
White, E. D.
Wood, C. F.
Woodward, R.
Worsham, H.
Wyld, R. G.

East Cleveland—
Stark, W. E.
Steffner, E. F.

Elyria—
Kalinsky, A. G.
Maynard, J. E.

Euclid—
Seigel, L. J.
Wilhelm, J. E.

Hamilton—
Thomas, L. G. L.

Kent—
Saginor, S. V.

Lakewood—
Longcoy, G. B.
Norrington, W. L.
Schurman, J. A.

Lima—
Hawisher, H. H.

Lorain—
Jackson, W. F.

Mansfield—
Trenner, K.

Middletown—
Byrd, T. I.
Essex, J. L.
Somers, W. S.

Newark—
Waggoner, J. H.

Norwood—
Braun, J. J.

Oberlin—
Ries, L. S.
Sable, E. J.

Painesville—
Hobbs, J. C.

Piqua—
Lange, R. T.

Powell—
Allonier, H. R.

Shaker Heights—
Foley, J. L.

South Park—
Barney, W. E.

Springfield—
Hauck, E. L.

Toledo—
Bergan, J. R.
Graham, J. M.
Hall, J. R.
Jones, S.
McKitrick, W. D.
Terry, M. C.
Watkins, G. B.

University Heights—
Mannen, D. E., Jr.

Waverly—
Armbruster, F. T. W.

Wilmington—
Marconett, V. G.
Sapp, C. L.

Youngstown—
Montgomery, J. R.
Sgambati, A. P.

OKLAHOMA

Norman—
Dawson, E. F.

Oklahoma City—
Braniff, P. R.
Campbell, A. O.
Carnahan, J. H.
Carroll, W. M.
Dolan, R. G.
Gray, E. W.
Kidd, C. R.
Loeffler, F. X.
Mideke, J. M.
Morin, A. R.
Rolland, S. L.
Tiller, L.

Tulsa—
Dean, C. H.
Holmes, A. D.
Jones, E.
Meinholtz, H. W.
Mumford, W. W.
Pauling, R. E.

OREGON

Corvallis—
Willey, E. C.

Eugene—
Chase, P. S.

Gladstone—
Williams, L. G.

Medford—
Hoey, J. K.

Portland—
Armstrong, C. E.
Banta, G. L.
Brissenden, C. W.
Burton, W. R.
Byrne, J. J.
Campau, W. R.
Carroll, E. E.
Clark, A. C.
Davis, R. J.
Enders, C. E.
Farnes, B. W.
Finnigan, W. T.
Fox, W. K.
Freeman, J. A.
Gehrs, W.
Goehler, E. E.
Hanthorn, W.
Heinkel, C. E.
Kollas, W. J.
Kroeker, J. D.
Lynch, J. R.
MacGregor, C. M.
McClung, T. H.
McDermott, J. P.
McIndoe, J. F.
Mead, H. K.
Michaels, M. A.
Moore, B. W.
Morrison, W. B.
Munro, D. R.
Murhard, E. A.
Neubauer, E. W.
Nielson, H. B.
Ponder, E. A.
Richardson, L. S.
Taylor, T. E.
Turner, E. S.
Urban, F. F.
Watson, K. W.
Weller, A. K.
Widmer, W. J.
Yates, J. E.

Salem—
Cooper, D. E.
VanWyngarden, J. E.

PENNSYLVANIA

Abington—
Bigelow, E. S.
Park, N. W.

Aldan—
Mulcey, P. A.

Allentown—
Goundie, J. K.
Hersh, F. C.
Korn, C. B.

Ambler—
McElgin, J. W.

Ardmore—
Haynes, C. V.

Ardsley—
Tucker, L. A.

Avondale—
Battan, S. W.

Beechwood—
Kipe, J. M.

Bethlehem—
Murnin, E. A., Jr.
Stuart, M. C.

Blairsville—
Mabon, J. E.

Bradford—
Paterson, F. C., Jr.

Brookline Park—
Call, J.

Drexel Hill—
Matz, G. N.

Dunbar—
Sherwood, L. T.

East Pittsburgh—
Hazlett, T. L.
Penney, G. W.

Edgewood—
Strauch, P. C.

Elizabethtown—
Dibble, S. E.

Erie—
Sahlmann, F. L.

Glenmoore—
Gant, H. P.

Harrisburg—
Eicher, H. C.
Geiger, I. H.
Herre, H. A.

Hershey—
Snavely, A. B.

Johnstown—
Hunter, L. N.
Knowles, F. R.
Novotney, T. A.

Kingston—
MacDonald, D. B.
McGown, F. H., Jr.
Stewart, J. P.

Lancaster—
Jones, A.
Lloyd, E. C.
Shorb, W. A.

Lansdowne—
James, H. R.
Seltzer, P. A.

Manheim—
Weitzel, C. B.
Weitzel, P. H.

McKeespor.—
Dugan, T. M.

Middletown—
Locke, R. A.

Milton—
Mohn, H. L.

Mount Lebanon—
Lige, W. W.

Narberth—
Grant, W. A.
Searle, W. J., Jr.
Wilmot, C. S.

Newtown—
Lewis, T.

Oakmont—
Edwards, J. D.
Lieberman, M. S.

Oxford—
Ware, J. H., III

Penn Valley—
Smith, W. F.

Philadelphia—
Ahlff, A. A.
Arnold, R. S.
Bachman, F.
Ballman, W. H.
Barnard, M. E.
Bartlett, C. E.
Belford, L. duB.
Black, E. N., III
Black, H. G.
Blankin, M. F.
Bogaty, H. S.
Bornemann, W. A.
Caldwell, A. C.
Carney, E. J.
Cassell, J. D.
Childs, L. A.
Cody, H. C.
Culbert, W. P.
Dafter, E. H.
D'Ambly, A. E.
Davidson, L. C.
Davidson, P. L.
Dietz, C. F.
Dome, A. G.
Donovan, W. J.
Eastman, C. B.
Elliot, E.
Erickson, H. H.
Faltenbacher, H. J.
Fameletti, A. R.
Farrington, S. E.
Galligan, A. B.
Gillett, M. C.
Gilman, F. W.
Goff, J. A.
Hajek, W. J.
Hedges, H. B.
Hibbs, F. C.
Hucker, J. H.
Hunger, R. F.
Hutchison, J. E.
Hynes, L. P.
Ickeringill, J. C.
Jacobsen, K. C. S.
Jakoby, A. C.
Jardine, W. H., Jr.
Keeling, F. V.
Kelble, F. R.
Killough, R. E.
Kirkbride, J. O.
Kohler, J. C.
Kriebel, A. E.
Ladd, D.
Landau, M.
Lauer, R. F.
Leopold, C. S.
Lyon, P. S.
Mack, L.
Macrow, L.
Makin, H. T., Jr.
Martocello, J. A.
Mather, H. H.
McCaffrey, C. E.
McCullough, H. G.
McIlvaine, J. H.
Mellon, J. T. J.
Meloney, E. J.
Mensing, F. D.
Millham, F. B.
Moody, L. E.
Morgan, R. C.
Murdoch, J. P.
Nesbitt, A. J.
Nesbitt, J. J.
Nusbaum, L.
Nusbaum, S. R.
Parent, H. M.
Pfeiffer, F. F.
Plewes, S. E.
Powell, G. W., Jr.
Powers, Earle C.
Powers, Edgar C.
Prewitt, H. B.
Pryibil, P. L.
Redstone, A. L.
Rettew, H. F.
Roberts, H. L.
Rugart, K.
Sabin, E. R.
Schneider, C. H.
Schulz, E. L.
Shanklin, A. P.
Shapiro, C. A.
Sheffler, M.
Shore, D.
Speckman, C. H.
Timmis, P.
Touton, R. D.
Traugott, M.
Tuckerman, G. E.
Wagner, E. K.
Wegmann, A.
Wells, W. F.
Whitney, C. W.
Wiley, D. C.
Wiltberger, C. F.
Woolston, A. H.

Pittsburgh—
Beighel, H. A.
Blackmore, G. C.
Borton, A. R.
Brauer, R.
Braun, C. R., Jr.
Breyer, F.
Bushnell, C. D.
Carr, M. L.
Collins, J. F. S., Jr.
Comstock, G. M.
Cooperman, E.
Cost, G. W.
Cox, S. F.
Dickinson, R. P., Jr.
Dickson, R. W., Jr.
Dorfan, M. I.
Edwards, P. A.
Ferderber, M. B.
Gallagher, F. H.
Good, C. S.
Greiner, G. E., Jr.
Griest, K. C.
Hample, H. J.
Hebley, H. F. J.
Hecht, F. H.
Heilman, R. H
Houghten, F. C.
Humphreys, C. M.
Hutcheon, C. R.
Hyde, E. H.
Jones, H. S.
Jones, L. K.
Kenney, T. W.
Kirkendall, H. J.
Landes, B. D.
Lieblich, M.
Lifshitz, H.
Long, E. J.
Loucks, D. W.
Lowe, W.
Maehling, L. S.
Mahon, F. B.
Maier, G. M.
Marshall, A. W.
McCullough, J. L.
McGonagle, A.
McIntosh, F. C
Metzger, A. F.
Miller, R. A.
Moore, H. L.
Mueller, J. E.
Nass, A. F.
Nicholls, P.
Park, H. E.
Parks, C. E.
Peacock, G. S.
Proie, J.
Reed, Van A., Jr.
Reed, W. H., III
Riesmeyer, E. H., Jr.
Rockwell, T. F.
Rose, H. J.
Rosenberg, I.
Scanlon, E. L.
Small, B. R.
Smith, R. L.
Smyers, E. C.
Speller, F. N.
Stanger, R. B.
Stauffer, J. E.
Steggall, H. B.
Stevenson, W. W.
Sweeney, R. H.
Tennant, R. J. J.
Tower, E. S.
Tumpane, J. P., Jr.
Waters, G. G.
Weddell, G. O.
Whitelaw, H. L.
Winer, B. B.

Pottstown—
Harberger, G. L.

Pottsville—
Marty, E. O.

Primos—
Johnson, A. J.

Reading—
Luck, A. W.

Ridley Park—
Mawby, P.

Scranton—
Mahon, B. B.

Springdale—
Lynn, F. E.

State College—
Queer, E. R.

Stroudsburg—
Kiefer, E. J.

Swarthmore—
Hobbs, W. S.
Robinson, A. S.
Thom, G. B.

Tarentum—
Orr, L.

Uniontown—
Marks, A. A.

Upper Darby—
Bertrand, G. F.
Morehouse, J. S.

Villa Nova—
Barr, G. W.

Washington—
Frazier, J. E.

West View—
Reilly, B. B.

Wilkinsburg—
Biber, H. A.
Campbell, T. F.
Graham, J. B.

Williamsport—
Axeman, J. E.

Wyncote—
Buck, L.

York—
Barnum, W. E., Jr.
Hertzler, J. R.
Kartorie, V. T.
Mirabile, J. J.
Nicoll, S. F.
Walsh, E. R., Jr.
Zieber, W. E.

RHODE ISLAND

Pawtucket—
Kramer, C.

Providence—
Blanding, R. L.
Coleman, J. B.
Gibbs, E. W.
Hartwell, J. C.
McCarthy, J. J.
McLaughlin, J. D.
Moulder, A. W.

SOUTH CAROLINA

Charleston—
Bailey, F. A., Jr.
Herty, F. B.

Clemson—
Shenk, D. H.

Columbia—
Hartin, W. R.
Kerr, W. E.
McDowell, H. L.
Mercer, C. F.
Sherman, W. P.

Greenville—
Ramseur, V. D., Jr.
Waldrep, J. E.

SOUTH DAKOTA

Lead—
Pullen, R. R.

Sioux Falls—
Monick, F. R.

TENNESSEE

Elizabethton—
Torok, E.

Kingsport—
Herbert, J. S.

Knoxville—
Cross, F. G.
Hawes, H. D.
Oakley, L. W.

Memphis—
Hoshall, R. H.
Shafer, W. P., Jr.

Nashville—
Armistead, W. C.
Baker, W. C.
Brown, F.
Campbell, A. Q., Jr.
Campbell, G. S.
Crane, R. S.
Wilson, V. H.

TEXAS

Amarillo—
Burnett, E. S.
Towne, C. O.

Austin—
Degler, H. E.

Beaumont—
Shell, J.

Bellville—
Woods, C. F.

Brownwood—
Feldstein, H.
Rhine, G. R.

College Station—
Giesecke, F. E.
Hines, G. M.
Hopper, J. S.
Long, W. E.
Smith, E. G.
Warren, H. P.

Corpus Christi—
Beals, D. E.
Blum, H., Jr.
Holsworth, R. C.

Dallas—
Anspacher, T. H.
Bishop, J. A.
Brown, M. L.
Brown, M. W.
Cox, V. G.
DeVilbiss, P. T.
Disney, M. A.
Eutsler, E. E.
Farrow, E. E.
Gardner, C. R.
Gilbert, L. S.
Green, S. H.
Jelinek, F. R.
Kribs, C. L., Jr.
Landauer, L. L.
Linskie, G. A.
Lyford, R. G.
Martyn, H. J.
McClanahan, L. C.
Mehl, O. H.
Perkins, R. C.
Pfeiffer, D. C.
Pines, S.
Renouf, E. P.
Rodgers, F. A.
Rogers, R. C.
Ullrich, A. B., Jr.
Witmer, C. N.
Zumwalt, R.

Fort Worth—
Harris, A. M.
MacEachin, G. C.
Skinner, H. W.
Sprekelmeyer, J. M.
Werner, R. K.

Galveston—
Ruemmele, A. M.

Houston—
Badgett, W. H.
Banowsky, A. B.
Barnes, A. F.
Barnes, W. L.
Beaird, B. J.
Bible, H. U.
Chase, A. M., Jr.
Cooper, D. S.
Drescher, F. E.
Earl, W.
Johnson, R. B.
Keeland, B. W.
Kurtz, R. W.
Maves, G. D.
McKinney, C. A.
Mills, D. M.
Mitchell, A. J.
Morrow, J. DeW.
Naman, I. A.
Olson, G. E.
Owings, H. L.
Pettit, E. N., Jr.
Salinger, R. J.
Spencer, R. M.
Taylor, R. F.
Tucker, R. E.
Walsh, J. A.

Kelly Field—
Weaver, J. VanO.

Lubbock—
Ainsworth, S. E.

San Antonio—
Barnes, R. W.
Diver, M. L.
Ebert, W. A.
Miller, B. R.
Pawkett, L. S.
Rummel, A. J.

San Jacinto—
Dillender, E. A.

Vernon—
Weatherby, E. P., Jr.

Waco—
Benham, F. C., Jr.

UTAH

Ogden—
Young, J. T., Jr.

Salt Lake City—
Richardson, H. G.

VERMONT

Burlington—
Lanou, J. E.
Raine, J. J.

VIRGINIA

Alexandria—
Goergens, A. G.
Millard, J. W.
Skagerberg, R.

Arlington—
Bozeman, R.
Cover, R. R.
Ferrarini, J.
Grimes, F. M.
Hackett, F. C.
Horne, H. F.
Marshall, W. D.
Nye, L. B., Jr.

Blacksburg—
Johnston, R. M.

East Falls Church—
Vaughan, J. G., Jr.

Falls Church—
Rogers, C. S.

Fort Belvoir—
Gault, G. W.

Fredericksburg—
Delany, J. V.

Front Royal—
Hartsook, G. S., Jr.

Lynchburg—
Doering, F. L.
Franklin, S. H., Jr.

Norfolk—
Huybert, L. E., Jr.
Nowitzky, H. S.
Thomas, R. C.

Petersburg—
Ibison. J. L.

Portsmouth—
Stubbs, W. C.

Richmond—
Bahlmann, W. F.
Bernard, E. L.
Carle, W. E.
Johnston, J. A.
Peebles, J. K., Jr.
Schulz, H. I.

Roanoke—
Bailey, A. E., Jr.
Bernert, L. A.
Nininger, C. H.

Williamsburg—
McGinnis, F. L.

Windsor—
Bailey, C. F.

WASHINGTON

Bremerton—
Bysom, L. L.

Kent—
Boyker, R. O.

Port Orchard—
Pratt, F. J.

Seattle—
Beggs, W. E.
Bouillon, L.
Clausen, A. H.
Cox, W. W.
Eastwood, E. O.
Granston, R. O.
Griffith, H. T.
Hauan, M. J.
Mallis, W.
May, C. W.
Morse, R. D.
Musgrave, M. N.
O'Connell, P. M.
Peterson, S. D.
Pollard, A. L.
Sparks, J. D.
Twist, C. F.
Wallis, W. M.
Watt, R. D.
Weber, E. L.
Wesley, R. O.
Zokelt, C. G.

Spokane—
Brown, S. D.
Russell, W. B.

Tacoma—
Chase, R. E.
Foote, E. E.
Norby, K. H.
Spofforth, W.

Vancouver—
Alben, E. A.

Yakima—
Leichnitz, R. W.
McCune, B. V.

WEST VIRGINIA

Charleston—
Pugh, D. C.
Rosenblatt, A. M.
Rothmann, S. C.
Shanklin, J. A.
Shearer, W. A., Jr.

Fairmont—
Tonry, R. C.
Wright, C. E.

Huntington—
Johnson, L. O.

Largent—
Donnelly, J. A.

Martinsburg—
Caskey, L. H., Jr.

Wheeling—
Hitt, J. C.

WISCONSIN

Appleton—
Eisele, D. E.

Clintonville—
Quall, C. O.

Elm Grove—
Winkler, R. A.

Kohler—
Hvosleff, F. W.
Kohler, W. J., Jr.

La Crosse—
Anderegg, R. H.
Atherton, G. R.
Bowen, J. C.
Pellmounter, T. V.
Rowe, W. A.
Sloane, D.
Thomas, N. A.
Trane, R. N.

Madison—
Dean, C. L.
Feirn, W. H.
Hall, G.
Kliefoth, M. H.
Larson, G. L.
Nelson, D. W.
Seymour, J. E.
White, J. C.

Milwaukee—
Alfery, H. F.
Allan, W.
Banks, J. B.
Becker, C. S.
Boden, W. F.
Bowers, A. F.
Brown, W. H.
Buller, C. R.
Cummiskey, J. F.
Cutler, J. A.
Davis, D. W., Jr.
Davis, K. T.
Ellis, H. W.
Frentzel, H. C.
Gerstenberger, E. J.
Goldsmith, F. W.
Griewisch, A. H.
Haerle, R. A.
Hamacher, K. F.
Hamilton, H. S.
Hanley, E. V.
Haus, I. J.
Hessler, L. W.
Hoffmann, A.
Holland, W. T.
Hughey, T. M.
Jackson, C. H.
Jones, E. A.
Jung, J. S.
Ketter, J. W.
Kluge, B. M.
Koch, R. G.
Krenz, A. S.
Lavorgna, M. L.
Leitgabel, K. A.
Lingen, R. A.
Mack, E. H.
McKee, J. W.
Miller, C. W.
Miller, L. B.
Mueller, H. P.
Noll, W. F.
Page, H. W.
Podolske, A. R.
Randolph, C. H.
Reinke, L. F.
Schreiber, H. W.
Spence, M. R.
Stevens, W. H.
Swisher, S. G., Jr.
Szekely, E.
Thom, A. J.
Tutsch, R. J.
Volk, G. H.
Volk, J. H.
Weil, F. H. E.
Weimer, F. G.
Werner, P. H.

Neenah—
Angermeyer, A. H.
Eiss, R. M.
Harvey, A. D.

Racine—
Dixon, A. G.
May, M. F.
Minkler, W. A.

South Milwaukee—
Ouweneel, W. A.

Thiensville—
Trostel, O. A.

Wauwatosa—
Gifford, E. W.

Whitefish Bay—
Runge, C. H.

ALASKA

Fairbanks—
Menden, P. J.

PANAMA

Canal Zone—
Danielson, W. A.

PHILIPPINE ISLANDS

Manila—
Grandia, W. M.
Hausman, L. M.
Macrae, R. B.

DOMINION OF CANADA

Brandon, Man.—
Yates, J. E.
Yates, R. A.

Brantford, Ont.—
Armour, E. G.

Calgary, Alta.—
Deeves, E. W.
Jenkins, S. D.

Edmonton, Alta.—
Mould, D. E.

Flin Flon, Man.—
Foster, P. H.

Freeman, Ont.—
Goodram, W. E.

Galt, Ont.—
Libby, R. S.
Sheldon, W. D., Jr.

Halifax, Nova Scotia-
Goodman, C. E.
Meagher, A. T.

Hamilton, Ont.—
Dickenson, M. E.
Moffat, O. G.

Hampstead, P. Q.—
Montgomery, E. G.

Islington, Ont.—
Wilson, G. T.

Kirkland, Ont.—
Arkley, L. M.

Kirkland Lake, Ont.-
Calver, R. W.

Kitchener, Ont.—
Beavers, G. R.
Pollack, C. A.

Leaside, Ont.—
Norton, J. A.

London, Ont.—
Gilbert, T.
O'Flaherty, J. G.
Rogers, T. L. C.

Montreal, P. Q.—
Armstrong, W. J.
Ballantyne, G. L.
Barnsley, F. R.
Baxter, W. E.
Becker, S. J.
Berridge, W. W.
Boland, R. O.
Chenevert, J. G.
Colle, S. S.
Darling, A. B.
Dufault, F. H.
Dykes, J. B.
Ewens, F. G.
Flanagan, J. B.
Forrester, N. J.
Freeman, E. M.
Friedman, F. J.
Garneau, L.
Hamlet, F. A.
Hamlet, T. F.
Hughes, W. U.
Johnson, C. W.
Keith, J. P.
Kitchen, W. H. J.
LaMontagne, A. F.
LaRue, J. A. D.
Linton, J. P.
Madely, F. J.
Martin, R.
Milne, A. H.
Morris, J. A.
Nickle, A. J.
Noyes, R. R.
Peart, A. M.
Perras, G. E.
Phipps, F. G.
Plant, E. B.
Robertson, J. A. M.
Roche, I. F.
Ross, J. D.
Sampson, E. T.
Shaw, J. A.
Ste. Marie, G. P.
Twizell, E. W.
Vollmann, C. W.
Walford, L. C. A.
Watts, A. E.
Wiggs, G. L.
Wilkinson, A.
Wormley, R. F.
Worthington, T. H.

Moosomin, Sask.—
Barton, E. H.

Noranda, P. Q.—
MacLean, H. A.

Oakville, Ont.—
Stott, F. W.

Ottawa, Ont.—
Allen, A. W.
Colclough, O. T.
Gray, G. A.
Johns, C. F.
McGrail, T. E.
Pennock, W. B.

Outremont, P. Q.—
Gittleson, H.
Osborne, G. H.

Preston, Ont.—
Everest, R. H.
Wood, A. W.

Quebec, P. Q.—
Paquet, J. M.
Roy, L.

Sackville, N. B.—
Rand, F. R.

St. Lambert, P. Q.—
Lefebvre, E. J.

St. Laurent, Que.—
Standring, R. A.
Tolhurst, G. C.

Sherbrooke, P. Q.—
Archambault, J. A.
Labonne, H.

Thorold, Ont.—
Calnan, E. J.

Three Rivers, P. Q.—
Germain, O.

Timmins, Ont.—
Smith, R. J.

Toronto, Ont.—
Abbott, T. J.
Alexander, S. W.
Allcut, E. A.
Allsop, R. P.
Angus, H. H.
Anthes, L. L.
Arrowsmith, J. O.
Baker, G. R.
Bayles, R. W.
Bishop, J. W.
Blackhall, W. R.
Blizzard, B. C.
Bowerman, E. L.
Bowes, W. H.
Brittain, A., Jr.
Carter, A. W.
Chambers, F. W.
Church, H. J.
Clifton, J. A.
Cole, G. E.
Davis, E. J.
Daynes, J. H.
Dickey, A. J.
Dickson, G. P.
Dion, A. M.
Dowler, E. A.
Duncan, W. A.
Eaton, W. G. M.
Ellis, F. E.
Fear, S. L.
Fitzsimons, J. P.
Foley, J. J.
Fox, E.
Fox, J. H.
Gauley, E. R.
Givin, A. W.
Gordon, C. W.
Gurney, E. H.
Gurney, E. R.
Henion, H. D.
Hill, H. G.
Hills, A. H.
Hopper, G. H.
Hughes, L. K.
Jenkinson, V. J.
Jenney, H. B.
Jennings, S. A.
Jones, A. T.
Kelly, W. C.
Lawlor, J. J.
Ledgett, F. D.
Leitch, A. S.
Lock, R. H.
Macdonald, D. J.
Marriner, J. M. S.
Mathison, R. S.
Maxwell, R. S.
McKerlie, J.
McLaren, T. H.
Moore, F. C.
Moore, H. S.
Morgan, A. S.
Murray, H. G. S.
Nearingburg, A.
Oke, W. C.
O'Neill, J. W.
Paul, D. I.
Philip, W.
Playfair, G. A.
Price, D. O.
Ritchie, A. G.
Roth, H. R.
Shears, M. W.
Smith, G. E.
Smith, W. H.
Spall, E. G.
Stencel, R. A.
Stott, D. A.
Sturdy, O. C.
Tasker, C.
Thomas, M. F.
Thomsen, N. B.
Waldon, C. D.
Wardell, A.
Willer, M. D.
Woollard, M. S.

Vancouver, B. C.—
Hale, F. J.
Johnston, R. E.
Leek, C. W.
Leek, W.
Turland, C. H.
Vissac, G. A.

Victoria, B. C.—
Sheret, A.

Westmount, P. Q.—
Colford, J.

Winnipeg, Man.—
Anderson, E.
Argue, E. J.
Ball, F. T.
Charles, P. L.
Chester, F. L.
Dahlgren, G. E.
Davis, G. C.
Glass, W.
Hepburn, E. M.
Jones, B. G.
Kent, R. L.
Kipp, T.
McDonald, I.
Michie, D. F.
Miller, E. R.
Munn, E. F.
Price, E. H.
Steele, J. B.
Stephenson, J. R.
Thompson, F.
Worton, W.

Woodstock, Ont.—
Karges, A.

AUSTRALIA

Kew—
Davies, R. H.

Melbourne—
Atherton, A. E.
Bell, S. R.
Ross, R.

Rose Bay—
Robinson, J. A.

Sydney—
Davey, G. I.
Henderson, A. S.
Moloney, R. R.
Roseby, T. A.
Sands, C. C.

Watsons Bay—
Picot, J. W.

BELGIUM

Brussels—
Lebrun, P.

BRAZIL

Rio de Janeiro—
Botelho, N. J.
De Sales, M., Jr.

CHILE

Santiago—
Demeter, J.

CHINA

Shanghai—
Bradford, G. G.
Doughty, C. J.
Gange, F. B.
Hart-Baker, H. W.
Hutchinson, F. A.
Kwan, I. K.
Loh, N. S.
Loo, P. Y.
Muckle, J.
Wong, W. S. B.

CUBA

Havana—
Colmenares, G. V.

DENMARK

Copenhagen—
Reck, W. E.
Schulein, E. H.

EGYPT

Alexandria—
Tallianos, P. C.

Cairo—
Ezz-El-Din, K.
Tahry, M. E.

ENGLAND

Birmingham—
Richardson, R. D.

Kent—
Figgis, T. G.
Lipscombe, H. W. J.

Lancaster—
Bartley, H. E.
Carter, D.

Leeds—
Jennins, H. H.

Liverpool—
Thomas, A. E.

London—
Bailey, W. M.
Benham, C. S. K.
Bird, G. L. H.
Butt, R. E. W.
Daniel, W. E.
Greenland, S. F.
Haden, G. N.
Herring, E.
Jackson, G. R.
Morrison, C. B.
Nobbs, W. W.
Russell, J. N.
Swain, W. L.
Symonds, E. S.
Troup, J. D.
Wilson, E. D.

Manchester—
Pougher, B. R. E.
Pougher, E. W.
Webb, J. W.
Yates, W.

Middlesex—
Case, W. G.
Fraser, J. J.
MacLachlan, V. D.

Nottingham—
Norfolk, L. W

Surrey—
Casperd, H. W. H.
Faber, O.

Trowbridge—
Haden, W. N.

Warwickshire—
Mann, W. N.

Westminster—
Kraminsky, V.

FRANCE

Boulogne—
Waudby, W.

Lyon—
Goenaga, R. C.

Paris—
Bodmer, E.
Modiano, R. N.
Nessi, A.

Sucy en Brie—
Beaurrienne, A.

Vanves—
Ghilardi, F.

FRENCH NORTH AFRICA

Algiers—
Macherel, F.

GERMANY

Berlin-Charlottenburg—
Brust, O.

Berlin-Wilmersdorf—
Schmidt, E. G.

Stuttgart—
Klein, A. R.

INDIA

Bombay—
Badhe, J. M.
Duplant, J. L.

Calcutta—
Ghose, K. N.
Ghosh, B. B.
Rachal, J. M.

New Delhi—
Baker, D. L.

IRELAND

Cork—
Barry, P. I.

Dublin—
Leonard, L. C. G.

ITALY

Florence—
Scotti, F. D.

Milan—
Dell'Orto, L.
Gini, A.
Hauss, C. F.
Marzorati, G.
Parrilli, R.

Torino—
Baldi, G.

JAPAN

Osaka—
Fukui, K.

Tokyo—
Kitaura, S.
Saito, S.
Sekido, K.

MANCHOUKUO

Yamatoku—
Kawase, S.

MEXICO

Mexico, D. F.—
Gilfrin, G. F.
Huber, E.
Martinez, J. J.

NETHERLANDS

Amsterdam—
ter Weeme, A.

Arnhem—
Tanger, O. C. F

NEW ZEALAND

Christchurch—
Taylor, E. M.
Vale, H. A. L.

Dunedin—
Davies, G. W.

NORWAY

Oslo—
Tjersland, A.

Stabekk—
Alfsen, N.

ROUMANIA

Bucharest—
Wasser, M.

SOUTH AFRICA

Cape Town—
Ehlers, J.

Johannesburg—
Daitsh, A.
Overton, S. H.
Rabe, A. E.
vonChristierson, C. A.

SPAIN

Madrid—
Alfageme, B.

STRAITS SETTLEMENTS

Singapore—
Faxon, H. C.
Hill, C. F.

SWEDEN

Nockeby—
Erikson, H. A.

South Angby—
Rosell, A. F.

Stockholm—
Eklund, K. G.
Gille, H. B.
Ostrom, E. W.
Stromgren, S. G.
Theorell, A. T.
Theorell, H. G. T.

TURKEY

Istanbul—
Karakash, T. J.
Veglery, A.

VENEZUELA

Caracas—
Blas, R. J.
Westendarp, F. G.

PAST OFFICERS

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1894

President....Edward P. Bates
1st Vice-President....Wm. M. Mackay
2nd Vice-President....Wiltsie F. Wolfe
3rd Vice-President....Chas. S. Onderdonk
Treasurer....Judson A. Goodrich
Secretary....L. H. Hart

Board of Managers

Chairman, Fred P. Smith

Henry Adams — A. A. Cary
Hugh J. Barron — James A. Harding
Edward P. Bates, *Pres.* — L. H. Hart, *Secy.*

Council

Chairman, R. C. Carpenter

Albert A. Cryer — Chas. W. Newton
F. W. Foster — Ulysses G. Scollay, *Secy.*

1895

President....Stewart A. Jellett
1st Vice-President....Wm. M. Mackay
2nd Vice-President....Chas. S. Onderdonk
3rd Vice-President....D. M. Quay
Treasurer....Judson A. Goodrich
Secretary....L. H. Hart

Board of Managers

Chairman, James A. Harding

Geo. B. Cobb — Ulysses G. Scollay
Wm. McMannis — B. F. Stangland
Stewart A. Jellett, *Pres.* — L. H. Hart, *Secy.*

Council

Chairman, R. C. Carpenter

Henry Adams — T. J. Waters
Edward P. Bates — Albert A. Cryer, *Secy.*

1896

President....R. C. Carpenter
1st Vice-President....D. M. Quay
2nd Vice-President....Edward P. Bates
3rd Vice-President....F. W. Foster
Treasurer....Judson A. Goodrich
Secretary....L. H. Hart

Board of Managers

Chairman, Wm. M. Mackay

Hugh J. Barron — Stewart A. Jellett
W. S. Hadaway, Jr. — Wiltsie F. Wolfe
R. C. Carpenter, *Pres.* — L. H. Hart, *Secy.*

Council

Chairman, A. A. Cary

Albert A. Cryer — B. F. Stangland
Wm. McMannis — J. J. Blackmore, *Secy.*

1897

President....Wm. M. Mackay
1st Vice-President....H. D. Crane
2nd Vice-President....Henry Adams
3rd Vice-President....A. E. Kenrick
Treasurer....Judson A. Goodrich
Secretary....H. M. Swetland

Board of Managers

Chairman, R. C. Carpenter

Edward P. Bates — Stewart A. Jellett
W. S. Hadaway, Jr. — Wiltsie F. Wolfe
Wm. M. Mackay, *Pres.* — H. M. Swetland, *Secy.*

Council

Chairman, Albert A. Cryer

John A. Fish — James Mackay
Wm. McMannis — B. F. Stangland

1898

President....Wiltsie F. Wolfe
1st Vice-President....J. H. Kinealy
2nd Vice-President....A. E. Kenrick
3rd Vice-President....John A. Fish
Treasurer....Judson A. Goodrich
Secretary....Stewart A. Jellett

Board of Managers

Chairman, Wm. M. Mackay

Thomas Barwick — A. C. Mott
John A. Connolly — Francis A. Williams
Wiltsie F. Wolfe, *Pres.* — Stewart A. Jellett, *Secy.*

Council

Chairman, R. C. Carpenter

Henry Adams — W. S. Hadaway, Jr.
Albert A. Cryer — Wm. McMannis
Wiltsie F. Wolfe, *Pres.* — Stewart A. Jellett, *Secy.*

1899

President....Henry Adams
1st Vice-President....D. M. Quay
2nd Vice-President....A. E. Kenrick
3rd Vice-President....Francis A. Williams
Treasurer....Judson A. Goodrich
Secretary....Wm. M. Mackay

Board of Managers

Chairman, Stewart A. Jellett

B. H. Carpenter — Wm. Kent
A. A. Cary — Wiltsie F. Wolfe
Henry Adams, *Pres.* — Wm. M. Mackay, *Secy.*

Council

Chairman, R. C. Carpenter

John Gormly — Wm. McMannis
W. S. Hadaway, Jr. — B. F. Stangland
Henry Adams, *Pres.* — Wm. M. Mackay, *Secy.*

1900

President....D. M. Quay
1st Vice-President....A. E. Kenrick
2nd Vice-President....Francis A. Williams
Treasurer....Judson A. Goodrich
Secretary....Wm. M. Mackay

Board of Governors

Chairman, D. M. Quay

Wm. Kent, *Vice-Chm.* — D. M. Nesbit
R. C. Carpenter — C. B. J. Snyder
John Gormly — Wm. M. Mackay, *Secy.*

1901

President....J. H. Kinealy
1st Vice-President....A. E. Kenrick
2nd Vice-President....Andrew Harvey
Treasurer....Judson A. Goodrich
Secretary....Wm. M. Mackay

Board of Governors

Chairman, J. H. Kinealy

Wm. Kent, *Vice-Chm.* — John Gormly
R. C. Carpenter — C. B. J. Snyder
R. P. Bolton — Wm. M. Mackay, *Secy.*

1912

President............John R. Allen
1st Vice-President............John F. Hale
2nd Vice-President............Edmund F. Capron
Treasurer............James A. Donnelly
Secretary............Wm. W. Macon

Board of Governors

Chairman, John R. Allen

John F. Hale, *Vice-Chm.*
Edmund F. Capron
R. P. Bolton
Jas. D. Hoffman
Dwight D. Kimball
Samuel R. Lewis
Wm. M. Mackay
Wm. W. Macon, *Secy.*

1913

President............John F. Hale
1st Vice-President............A. B. Franklin
2nd Vice-President............Edmund F. Capron
Treasurer............James A. Donnelly
Secretary............Edwin A. Scott

Board of Governors

Chairman, John F. Hale

A. B. Franklin, *Vice-Chm.*
John R. Allen
Edmund F. Capron
R. P. Bolton
Frank T. Chapman
Ralph Collamore
James A. Donnelly
Dwight D. Kimball
Wm. W. Macon
James M. Stannard
Theodore Weinshank
Edwin A. Scott, *Secy.*

1914

President............Samuel R. Lewis
1st Vice-President............Edmund F. Capron
2nd Vice-President............Dwight D. Kimball
Treasurer............James A. Donnelly
Secretary............J. J. Blackmore

Council

Chairman, Samuel R. Lewis

E. F. Capron, *Vice-Chm.*
Dwight D. Kimball
John R. Allen
Frank T. Chapman
Frank I. Cooper
James A. Donnelly
John F. Hale
Harry M. Hart
Frank G. McCann
Wm. W. Macon
James M. Stannard
J. J. Blackmore, *Secy.*

1915

President............Dwight D. Kimball
1st Vice-President............Harry M. Hart
2nd Vice-President............Frank T. Chapman
Treasurer............Homer Addams
Secretary............J. J. Blackmore

Council

Chairman, Dwight D. Kimball

Harry M. Hart, *Vice-Chm.*
Homer Addams
Frank T. Chapman
Frank I. Cooper
E. Vernon Hill
Wm. M. Kingsbury
Samuel R. Lewis
Frank G. McCann
J. T. J. Mellon
Henry C. Meyer, Jr.
Arthur K. Ohmes
J. J. Blackmore, *Secy.*

1916

President............Harry M. Hart
1st Vice-President............Frank T. Chapman
2nd Vice-President............Arthur K. Ohmes
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Harry M. Hart

F. T. Chapman, *Vice-Chm.*
Homer Addams
Charles R. Bishop
Frank I. Cooper
Milton W. Franklin
E. Vernon Hill
Dwight D. Kimball
Henry C. Meyer, Jr.
Arthur K. Ohmes
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1917

President............J. Irvine Lyle
1st Vice-President............Arthur K. Ohmes
2nd Vice-President............Fred R. Still
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, J. Irvine Lyle

A. K. Ohmes, *Vice-Chm.*
Homer Addams
Davis S. Boyden
Bert C. Davis
Milton W. Franklin
Charles A. Fuller
Harry M. Hart
E. Vernon Hill
James M. Stannard
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1918

President............Fred R. Still
1st Vice-President............Walter S. Timmis
2nd Vice-President............E. Vernon Hill
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Fred R. Still

W. S. Timmis, *Vice-Chm.*
Homer Addams
William H. Driscoll
Howard H. Fielding
H. P. Gant
C. W. Kimball
J. Irvine Lyle
E. Vernon Hill
Frank G. Phegley
Fred. W. Powers
Champlain L. Riley
Casin W. Obert, *Secy.*

1919

President............Walter S. Timmis
1st Vice-President............E. Vernon Hill
2nd Vice-President............Milton W. Franklin
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Walter S. Timmis

E. Vernon Hill, *Vice-Chm.*
Homer Addams
Howard H. Fielding
Milton W. Franklin
Harry E. Gerrish
George B. Nichols
Frank G. Phegley
Fred. W. Powers
Robt. W. Pryor, Jr.
Champlain L. Riley
Fred R. Still
Casin W. Obert, *Secy.*

1920

President............E. Vernon Hill
1st Vice-President............Champlain L. Riley
2nd Vice-President............Jay R. McColl
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, E. Vernon Hill

C. L. Riley, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Wm. H. Driscoll
A. C. Edgar
Alfred Kellogg
Jay R. McColl
George B. Nichols
Robt. W. Pryor, Jr.
W. S. Timmis
Perry West
Casin W. Obert, *Secy.*

1921

President............Champlain L. Riley
1st Vice-President............Jay R. McColl
2nd Vice-President............H. P. Gant
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Champlain L. Riley

Jay R. McColl, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm. H. Driscoll
H. P. Gant
E. S. Hallett
E. Vernon Hill
Alfred Kellogg
E. E. McNair
Perry West
Casin W. Obert, *Secy.*

1922

President....Jay R. McColl
1st Vice-President....H. P. Gant
2nd Vice-President....Samuel E. Dibble
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, Jay R. McColl

H. P. Gant, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm. H. Driscoll
E. S. Hallett
L. A. Harding
E. E. McNair
H. J. Meyer
C. L. Riley
Perry West
Casin W. Obert, *Secy.*

1923

President....H. P. Gant
1st Vice-President....Homer Addams
2nd Vice-President....E. E. McNair
Treasurer....Wm. H. Driscoll
Secretary....Casin W. Obert

Council

Chairman, H. P. Gant

Homer Addams, *Vice-Chm.*
W. H. Carrier
J. A. Cutler
S. E. Dibble
Wm. H. Driscoll
E. S. Hallett
Alfred Kellogg
Thornton Lewis
E. E. McNair
Perry West
Casin W. Obert, *Secy.*

1924

President....Homer Addams
1st Vice-President....S. E. Dibble
2nd Vice-President....William H. Driscoll
Treasurer....Perry West
Secretary....F. C. Houghten

Council

Chairman, Homer Addams

S. E. Dibble, *Vice-Chm.*
F. Paul Anderson
W. H. Carrier
J. A. Cutler
William H. Driscoll
H. P. Gant
W. E. Gillham
L. A. Harding
Alfred Kellogg
Thornton Lewis
Perry West
F. C. Houghten, *Secy.*

1925

President....S. E. Dibble
1st Vice-President....Wm. H. Driscoll
2nd Vice-President....F. Paul Anderson
Treasurer....Perry West
Secretary....F. C. Houghten

Council

Chairman, S. E. Dibble

Wm. H. Driscoll, *Vice-Chm.*
Homer Addams
F. Paul Anderson
W. H. Carrier
J. A. Cutler
W. E. Gillham
W. T. Jones
Thornton Lewis
J. H. Walker
Perry West
A. C. Willard
F. C. Houghten, *Secy.*

1926

President....W. H. Driscoll
1st Vice-President....F. Paul Anderson
2nd Vice-President....A. C. Willard
Treasurer....W. E. Gillham
Secretary....A. V. Hutchinson

Council

Chairman, W. H. Driscoll

F. Paul Anderson, *Vice-Chm.*
W. H. Carrier
J. A. Cutler
S. E. Dibble
W. E. Gillham
C. V. Haynes
W. T. Jones
E. B. Langenberg
Thornton Lewis
J. F. McIntire
A. C. Willard

1927

President....F. Paul Anderson
1st Vice-President....A. C. Willard
2nd Vice-President....Thornton Lewis
Treasurer....W. E. Gillham
Secretary....A. V. Hutchinson

Council

Chairman, F. Paul Anderson

A. C. Willard, *Vice-Chm.*
H. H. Angus
W. H. Carrier
W. H. Driscoll
Roswell Farnham
H. H. Fielding
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
Thornton Lewis
J. F. McIntire
H. Lee Moore
F. B. Rowley

1928

President....A. C. Willard
1st Vice-President....Thornton Lewis
2nd Vice-President....L. A. Harding
Treasurer....W. E. Gillham
Secretary....A. V. Hutchinson

Council

Chairman, A. C. Willard

Thornton Lewis, *Vice-Chm.*
F. Paul Anderson
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
J. F. McIntire
H. Lee Moore
F. B. Rowley

1929

President....Thornton Lewis
1st Vice-President....L. A. Harding
2nd Vice-President....W. H. Carrier
Treasurer....W. E. Gillham
Secretary....A. V. Hutchinson
Technical Secretary....P. D. Close

Council

Chairman, Thornton Lewis

L. A. Harding, *Vice-Chm.*
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
W. A. Rowe
F. B. Rowley
A. C. Willard

1930

President....L. A. Harding
1st Vice-President....W. H. Carrier
2nd Vice-President....F. B. Rowley
Treasurer....C. W. Farrar
Secretary....A. V. Hutchinson
Technical Secretary....P. D. Close

Council

Chairman, L. A. Harding

W. H. Carrier, *Vice-Chm.*
H. H. Angus
D. S. Boyden
R. H. Carpenter
J. D. Cassell
N. W. Downes
Roswell Farnham
C. W. Farrar
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
Thornton Lewis
F. C. McIntosh
W. A. Rowe
F. B. Rowley

1931

President......W. H. Carrier
1st Vice-President......F. B. Rowley
2nd Vice-President......W. T. Jones
Treasurer......F. D. Mensing
Secretary......A. V. Hutchinson
Technical Secretary......P. D. Close

Council

Chairman, W. H. Carrier

F. B. Rowley, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
Roswell Farnham
E. H. Gurney
L. A. Harding
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
F. D. Mensing
W. A. Rowe

1932

President......F. B. Rowley
1st Vice-President......W. T. Jones
2nd Vice-President......C. V. Haynes
Treasurer......F. D. Mensing
Secretary......A. V. Hutchinson
Technical Secretary......P. D. Close

Council

Chairman, F. B. Rowley

W. T. Jones, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
W. H. Carrier
John D. Cassell
E. O. Eastwood
Roswell Farnham
F. E. Giesecke
E. H. Gurney
C. V. Haynes
John Howatt
G. L. Larson
J. F. McIntire
F. D. Mensing
W. E. Stark

1933

President......W. T. Jones
1st Vice-President......C. V. Haynes
2nd Vice-President......John Howatt
Treasurer......D. S. Boyden
Secretary......A. V. Hutchinson

Council

Chairman, W. T. Jones

C. V. Haynes, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
R. Farnham
F. E. Giesecke
E. H. Gurney
John Howatt
G. L. Larson
J. F. McIntire
F. C. McIntosh
L. W. Moon
F. B. Rowley
W. E. Stark

1934

President......C. V. Haynes
1st Vice-President......John Howatt
2nd Vice-President......G. L. Larson
Treasurer......D. S. Boyden
Secretary......A. V. Hutchinson

Council

Chairman, C. V. Haynes

John Howatt, *Vice-Chm.*
M. C. Beman
D. S. Boyden
Albert Buenger
R. H. Carpenter
J. D. Cassell
F. E. Giesecke
E. H. Gurney
W. T. Jones
G. L. Larson
J. F. McIntire
F. C. McIntosh
L. Walter Moon
O. W. Ott
W. A. Russell
W. E. Stark

1935

President......John Howatt
1st Vice-President......G. L. Larson
2nd Vice-President......D. S. Boyden
Treasurer......A. J. Offner
Secretary......A. V. Hutchinson

Council

Chairman, John Howatt

G. L. Larson, *Vice-Chm.*
M. C. Beman
D. S. Boyden
Albert Buenger
R. H. Carpenter
J. D. Cassell
F. E. Giesecke
E. H. Gurney
C. V. Haynes
J. F. McIntire
F. C. McIntosh
L. Walter Moon
A. J. Offner
O. W. Ott
W. A. Russell
W. E. Stark

1936

President......G. L. Larson
1st Vice-President......D. S. Boyden
2nd Vice-President......E. H. Gurney
Treasurer......A. J. Offner
Secretary......A. V. Hutchinson

Council

Chairman, G. L. Larson

D. S. Boyden, *Vice-Chm.*
M. C. Beman
R. C. Bolsinger
Albert Buenger
S. H. Downs
W. L. Fleisher
F. E. Giesecke
E. H. Gurney
John Howatt
C. M. Humphreys
L. Walter Moon
J. F. McIntire
A. J. Offner
O. W. Ott
W. A. Russell
W. E. Stark

1937

President......D. S. Boyden
1st Vice-President......E. Holt Gurney
2nd Vice-President......J. F. McIntire
Treasurer......A. J. Offner
Secretary......A. V. Hutchinson

Council

Chairman, D. S. Boyden

E. H. Gurney, *Vice-Chm.*
J. J. Aeberly
M. C. Beman
R. C. Bolsinger
Albert Buenger
S. H. Downs
E. O. Eastwood
W. L. Fleisher
F. E. Giesecke
C. M. Humphreys
G. L. Larson
W A. Russell
W. E. Stark

1938

President......E. Holt Gurney
1st Vice-President......J. F. McIntire
2nd Vice-President......F. E. Giesecke
Treasurer......A. J. Offner
Secretary......A. V. Hutchinson
Technical Secretary......John James

Council

Chairman, E. Holt Gurney

J. F. McIntire, *Vice-Chm.*
N. D. Adams
J. J. Aeberly
M. C. Beman
R. C. Bolsinger
D. S. Boyden
S. H. Downs
E. O. Eastwood
W. L. Fleisher
F. E. Giesecke
C. M. Humphreys
A. P. Kratz
A. J. Offner
W. A. Russell
J. H. Walker
G. L. Wiggs

1939

President......J. F. McIntire
1st Vice-President......F. E. Giesecke
2nd Vice-President......W. L. Fleisher
Treasurer......M. F. Blankin
Secretary......A. V. Hutchinson
Technical Secretary......John James

Council

Chairman, J. F. McIntire

F. E. Giesecke, *Vice-Chm.*
N. D. Adams
J. J. Aeberly
M. C. Beman
M. F. Blankin
E. K. Campbell
S. H. Downs
E. O. Eastwood
W. L. Fleisher
E. H. Gurney
A. P. Kratz
A. J. Offner
W. A. Russell
G. L. Tuve
J. H. Walker
G. L. Wiggs

1940

President......F. E. Giesecke
1st Vice-President......W. L. Fleisher
2nd Vice-President......E. O. Eastwood
Treasurer......M. F. Blankin
Secretary......A. V. Hutchinson
Technical Secretary......John James

Council

Chairman, F. E. Giesecke

W. L. Fleisher, *Vice-Chm.*
N. D. Adams
E. K. Campbell
J. F. S. Collins, Jr.
S. H. Downs
A. P. Kratz
E. N. McDonnell
J. F. McIntire
A. J. Offner
G. L. Tuve
T. H. Urdahl
J. H. Walker
G. L. Wiggs
C.-E. A. Winslow